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TRANSPORTATION RESEARCH COMMAND

FORT EUSTIS, VIRGINIA

TCREC TECHNICAL REPORT 62-40

THE APPLICATION OF INFLATABLE STRUCTURES TO THE GROUND EFFECT MACHINE (GEM)

Task 9R99-01-005-06

Contract DA 44-177-TC-726

June 1962

prepared by:

BOOZ ALLEN APPLIED RESEARCH, INC.,
Bethesda, Maryland

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U. S. ARMY TRANSPORTATION RESEARCH COMMAND
Fort Eustis, Virginia

This report presents the results of an investigation conducted by Booz-Allen Applied Research, Incorporated, under Project 9R99-01-005-05, GEM Structures Study.

Studies and experimental investigations into the external loadings of air cushion vehicles indicated that the stresses were compatible with the properties of inflatable materials. Benefits which would accrue through use of inflatable components in the vehicles were the high potential of inflatables to absorb local impacts without rupture or permanent deformation and the capability of providing a practical planform area of minimum weight.

The report does not attempt to solve any particular development problems but rather to indicate where inflatable materials could be utilized advantageously, what influence natural and induced environmental conditions would have on design considerations, and what effect pneumatic structure would have on the weight, performance, and efficiency of military air cushion applications.

FOR THE COMMANDER:

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for
U. S. ARMY TRANSPORTATION RESEARCH COMMAND
FORT EUSTIS, VIRGINIA

PREFACE

The study presented in this report was undertaken by the Aerospace Engineering Group, Booz, Allen Applied Research, Inc., 4815 Rugby Avenue, Bethesda 14, Maryland and sponsored by the U. S. Army Transportation Research Command. This research program was carried out under the supervision of Mr. Peter G. Fielding. This study began in July 1961 and was completed March 1962. Mr. W. E. Sickles of the U. S. Army Transportation Research Command was the Project Officer.

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SUMMARY

This report is the result of a 9-month study covering the application of inflatable structures to the Ground Effect Machine (GEM) with particular reference to the 15-ton payload LOTS carrier mission described in QMR (reference 24). Consequently, the scope of study has been somewhat limited to the investigation of suitable inflation pressures, fabrics and bindings consistent with the structural and environmental criteria considered mandatory for this mission.

A review of the environmental conditions associated with the operation of Ground Effect Machines, followed by a discussion of the capabilities and construction of practical and projected GEM projects, provides the background material for the problem of inflatable structures. A review of the materials available for construction of inflatable structures indicates those that are of most value; a similar study of inflation systems indicates the advantages and disadvantages of several possible systems.

A simple flat cross-section with round ends is chosen as the example section for the analytical development of the required inflation pressure, during which the development of pressure, bending and torsion stresses in an inflated structure are discussed, and other aspects of the structure are reviewed, including deflection and collapse. In order to provide a realistic basis for evaluating an inflatable structure for GEMs, a preliminary design study is conducted for a 15-ton payload machine, resulting in critical bending moment values for the design. Using the geometrical characteristics of the design and the critical bending moments evaluated, the inflation pressure required to prevent structure wrinkling is determined. Changes in the structural geometry are investigated to determine quantitatively their effect on the required inflation pressure, and the utility of such structures for primary structure application in GEMs. Further analysis investigates their utility in other applications in GEMs.

As the final steps in the study, a parametric analysis is made of a range of GEM sizes and associated primary structures to show how change in GEM size and basic characteristics affects the required inflation pressure and structural depth. The potential of primary, secondary and auxiliary structures is reviewed, and an example auxiliary structure is developed. The inflation systems reviewed in the first part of the study are discussed further in the light of the several structural applications.

INTRODUCTION

One area of technology that has been used but not fully exploited in recent years is that of inflatable structures. Although much research and development needs to be carried out before inflatable structures for vehicles could be acceptable in military transportation, there are sufficient promising features in this type of structure to warrant further investigation into the feasibility and application to military ground effect machines (GEMs).

Among the characteristics of inflatable structures that are attractive to potential users and designers of transport vehicles are:

(1) Inherent transportability

(2) Easy repair

(3) Shock-absorbing capability

(4) Overload capabilities

(5) High strength-to-weight ratio, while inflated.

When compared with conventional structures there are, however, a number of potential disadvantages that must be examined before the use of inflated structures can be recommended for any application. Among these are:

(1) The effect of one or several punctures;

(2) The explosive failure of the structure while pressurized;

(3) The deterioration of the structural fibers and bonding agents under extreme service conditions;

(4) The strength of bonded joints in the structure;

(5) Tearing of the structural fabric;

(6) Abrasion of the structural fabric.

The characteristics of GEMs that have been proposed for use as transportation equipment in the near future give rise to problems in the development of such vehicles. Two such characteristics that affect the area of structural design are:

(1) Size

Such GEMs will be large in order to be able to operate with world-wide, year-round capability with acceptable economy.

(2) Operating Economy

The economics of such GEMs are particularly sensitive to the structure weight and power plant unit of the system, since power and associated ducting and controls are directly utilized in the production of lift, in addition to the structure utilized for the distribution of lift forces and the power required for forward motion. In other vehicles, lift is developed as a result of either forward motion or environment, and additional power and structure for lift is very small. The major problems that arise in consequence are concerned with:

(1) The transportability of these GEMs over intercontinental distances or to inland areas where operation is desired.

(2) The development of means for reducing structure weight and power plant weight to the minimum commensurate with the desired performance capability.

Solution of the second problem will provide the most satisfactory means for alleviating the first problem, since reduction in structural and power plant weight is a continuing effect, which ultimately will reduce vehicle size and gross weight for a given desired maximum payload, and ease the transportability problem.

Since the advantages basically inherent in inflatable structural technology evidently would be of particular benefit in GEMs, this study examines the specific advantages and disadvantages in the application of inflatable structures to the GEM; highlights those applications, methods, and materials that appear to have sufficient merit to warrant further research and development; and outlines the research and development required to permit operational use of inflatable structures. In particular, this study examines the following:

(1) Where and to what extent inflatable structures can be utilized in GEMs.

(2) What types of inflatable structure and material are most suitable for each application.

(3) What effect the use of inflatable structures has on GEM performance.

(4) What influence operating environment has on the use of inflatable structures.

(5) What effect variation in GEM size has on the utilization of inflatable structures, and what other applications there are in the ground effect field.

(6) What developments are required to improve the utilization of inflatable structures in relatively large GEMs.

In keeping with the contractual work statement, this study is limited to consideration of GEMs suitable for amphibious support and resupply operations in the range of 5- to 30-ton payloads, except where the study development indicates extension of this range to be useful. The major emphasis is given to a 15-ton payload machine, which has been suggested as being close to the size of GEM that will be most useful in such operations.

CHAPTER I

BACKGROUND DATA

1.1 INTRODUCTION

In common with many other technological areas, the development of inflatable structures over the past 150 years has been very gradual. The feasibility of utilizing a form of inflated structure for lifting men into the air was demonstrated as long ago as 1783, but serious application was not developed until the wars of the 19th century when balloons were used for aerial spotting. Up until the last ten years, serious development of this type of structure for vehicles has concentrated almost entirely on airship applications for military and civil use.

In the last ten years, the search for lightweight structures that could be packaged in small volume for applications in the aircraft and space fields has led to intensive investigation of more sophisticated forms of inflated structures. Several such structures have been built or proposed, ranging from complete lightweight aircraft, through a passive satellite, to proposals for complete space stations in orbit around the earth. Although the feasibility of using inflated structures has been demonstrated by these developments, the state of the art in structural design has not yet advanced sufficiently to make possible formulation of design criteria that would permit immediate application of such structures in military vehicles. The problems of environment, combat damage, maintenance, and repair are of such magnitude that development of requirements for the application of such structures in military situations is imperative, if such structures are to be successfully utilized

In order to develop a rational understanding of the problems inherent in inflatable structures as applied to GEMs, the environmental considerations of GEM operation, the current range of GEM sizes and performance of interest, the materials available for the fabrication of GEM inflatable structures, and the systems required to maintain inflation pressure under operational conditions must be critically reviewed.

This chapter, drawing heavily upon the work of reference 1, attempts to lay a solid quantitative foundation for the subsequent analyses reported in Chapters II, III and IV. The chapter is in five main sections covering environmental considerations, materials selection, review of current and projected GEMs with respect to their structural characteristics, inflatable materials and their characteristics, and inflation systems review.

1.2 ENVIRONMENTAL CONSIDERATIONS IN GEM DESIGN AND MATERIALS SELECTION

The design and operation of military Ground Effect Machines are significantly influenced by all the elements of the total environment surrounding the operations. Military GEMs are significantly influenced by the following three types of environment:

(1) Natural environment, such as climate, terrain, vegetation, sea states, beach slopes and condition, sea approaches to coast, and natural and man-made obstacles.

(2) Induced environment, such as component generated temperatures, noise, vibration, shock, explosive vapors, dust, spray, and exhaust gases.

(3) Combat environment, including detection, vulnerability, protection against damage, and field repairs.

An analysis of the total environmental picture surrounding the Ground Effect Machine on a world-wide operational basis is reported in reference 1. This section has been based on this work and relies on the data collected at that time.

Since the Ground Effect Machine has apparent capability for operations over water, overland, and across the land-water transition, the study referenced was divided into these three regimes of operation. At this time it is believed that the Army's interest is primarily in application of the Ground Effect Machine to the logistical over-the-shore cargo transfer operation. Greatest attention has therefore been given to the amphibious role and mission. Used in this context, the term "amphibious operations" covers land-water operations in the coastal areas of the world. Generally, the inland penetration is considered to be no more than 20 to 30 miles, but in areas of stream valley approaches this is not a strict limit. The over-water portion is limited to ranges less than 200 miles. This may include an unloading operation from a ship at sea, a traverse of a small body of water from another land area, or a land-based sea patrol. The total development of the environmental influences in amphibious GEM operations provides nearly the complete range of elements significant to Army application of GEMs.

In addition to the generalized world-wide operational environment of reference 1 which covers the Ground Effect Machine on a mobility basis, the GEM must also meet the Army's requirements for operation under extreme environmental conditions. These regulations are set down in reference 2, and have been incorporated in the report by reference. Significant elements are discussed in detail.

Other regulations and specifications of the Army and other services have been reviewed for applicable information. These include elements of environment in which Ground Effect Machine amphibious operations have analogs in other military systems including aircraft, ground support equipment, and tactical vehicles.

1.2.1 Operational Environment

For application to the LOTS operation, the development of parameters for amphibious GEMs is particularly appropriate. Such a development is summarized below:

(1) Vehicle Size

Limited by lateral clearance for operation in stream valleys and by widths of beaches, but not as severely as in general over-land operations:

30-foot width usable on 80-90 per cent of coastal stream valleys,

40-foot width usable on 70 per cent,

50-60-foot width usable on 50 per cent,

50-foot width (for end-loading GEMs) satisfactory for unloading above high water line on 70 per cent of beaches, 100-foot width satisfactory for unloading above high water line on 25 per cent of beaches,

20-foot width about maximum for operation in forest clearings other than waterways.

It is important to note that vehicle size may also be severely limited by transportability requirements which, although an operational consideration rather than environmental, must be considered.

(2) Operating Height

Determined by the necessity to traverse the surf zone and overland obstacles.

1. Normal Operating Height Overland

1 foot for operations to beaches only;

2 feet for limited inland penetration.

2. Jump Capability

4-foot minimum for crossing walls, dikes, embankments;

6 feet for wider area usability across walls, dikes, embankments;

5-foot vertical bank up and down for access from stream valleys.

3. Maximum Sustained Operating Height Over Water

2.5 feet for clearance of surf in 10 per cent of coastal areas;

4 feet for surf clearance in 70 per cent;

6 feet for surf clearance in 95 per cent.

(3) Slope Capability

10 per cent slope capability for operations over 80 per cent of beaches;

15 per cent slope capability for operations over 90 per cent of beaches, and for most near-coast terrains;

50-100 per cent slope on 10-foot banks for access from many stream valleys.

(4) Operating Speeds

Limited to about 30 knots for operations in tropical coastal swamps; also limited by obstacles in the overland phase; no specific numbers can be determined without further experimental work.

A suggested classification of amphibious GEMs based on these factors is given in Table I-1.

The vehicles classified in Table I-1 range in size from 20 x 40 feet to 75 x 150 feet. The normal operating heights given are the cruising heights overland; the maximum height is the operating height at about half normal cruise speed, and is intended primarily for operation through the surf zone. Jump capability represents the maximum attainable short duration height for clearance of obstacles. The slope capability listed corresponds to the slopes within the area having the given height ranges. The columns headed "Access to Beaches" and "Access Inland" represent the percentage of the world's coastline (excluding Antarctica, Greenland, and Arctic islands north of 77°) which are accessible to amphibious GEMs of each class. This accessibility index is intended to include operations throughout all seasons of the year, but not during stormy weather. On Arctic coasts the occurrence of hummocked ice may restrict operations during a large part of the year.

The most significant elements of induced and combat environments are summarized in Tables I-2 and I-3; an over-all summary of military GEM characteristics and typical equipment requirements is given in Tables I-4 and I-5.

TABLE I-1
AMPHIBIOUS GEM CLASSIFICATION

Class	Size (Feet)	Normal Height (Feet)	Max. Height (Feet)	Jump Capability (Feet)	Slope Capability (Per cent)	Access to Beaches (Per cent)	Access Inland Via Stream Valley (Per cent)
I.	20 x 40	1.5	2.5	3	10	10	10
II.	30 x 60	2	3	4	15	40	25
III.	40 x 80	3	5	6	15	70	40
IV.	50 x 100	3	5	6	15	70	35
V.	75 x 150	4	6	6	15	50	30

TABLE I-2
INDUCED ENVIRONMENTS
SUMMARY DATA

Noise

Maximum Over-All Noise Level, Relative to .0002 Millibar

Location	Acceptable (Air Force)	Experienced	Expected	Special Equipment Needed
Inside Vehicle	113	90 to 120	80 to 95	Ear protection for continuous use. Telephony for adequate operational functioning.
Close to Out-Side of Vehicle	113	90 to 120	90 to 120	
	See Combat Environment Summary on Noise (Table 19)			

Vibrations

Acceptable limits for Crew Stations

Frequency (cycles/ sec.)	Amplitude (in.)	Experience to Date
1	.02< h< 2	No indications that these values will be exceeded. No vibration problems so far.
10	.0002 < h <.02	
100	.00002< h <.0005	

TABLE I-2 (continued)

Temperatures (Maximum Values)

Engine	Combustion Chamber Area	Exhaust Area
Piston	240°C to 200°C	1000°C to 600°C
Jet	200°C to 250°C	800°C to 850°C

Exhaust Gases

Engine	Inert	Toxic	Water and Oxygen
Piston	77 per cent	6 to 18 per cent	17 to 5 per cent
Jet	77 per cent	1.5 to 4.6 per cent	21.5 to 18.4 per cent

Static Electricity

Typical Accumulated Charge (Volts)	Stored Energy Levels (Millijoules)	Energy Level for Spark Ignition of JP-4 (Millijoules)
300 to 400	2×10^{-4} to 2×10^{-2}	0.2

Effect on Personnel Bridging Gap

Initial current level -- lethal
Drops below lethal level within 2 millionths of a sec.

Solution to all Problems of Static Charge

A trailing ground-contacting conductor, and good vehicle bonding throughout.

TABLE I-2 (continued)

Visibility Reduction	
Due to:	**Comments**
Configuration	No worse than a ship, better than an aircraft
Night and Bad Weather	Usual solutions applicable
Surface Disturbance	
1. Forward Speed	No problem forward, upwards, or sideways. Some obstructions by surface material towards rear.
2. Hover	Severe loss of visibility. No major problem, as GEM basically stable and can move from hover location under full control, to regain visibility.

Acceleration and Shock — Accelerations at CG.	
Longitudinal	+ .2g, to -.5g
Lateral	$\pm$ .4g
Vertical (approx.)	$\pm$ .3g to $\pm$ .4g

TABLE I-2 (continued)

Shock	
Operating height	.02 (vehicle length)
Wave length	2.0 x (vehicle length)
"Sinusoidal" wave height	1.5 x (operating height)
Equivalent "Random Sea" wave height	3.0 x (operating height)

Operating Conditions	Normal Operating Speed	Twice Normal Operating Speed
At Bow - no immersion	9.0 to 12.0g	18 to 20g
with immersion	6.0 to 8.0g	12 to 15g
At Center of Gravity	3.0 to 6.0g	6 to 12g

TABLE I-2 (continued)

	Ingestion		
Material	Concentrations and Particle Sizes	Effect on Lift System & Propulsion	Engines
Dirt and Sand	0-80μ 40-105μ, 105-200μ. 0.01 gms/cu.ft.	Rotor blade and intake erosion, with loss in efficiency	Compressor blade erosion, up to 10 per cent power loss, reduction of surge margin, high turbine inlet temperatures
Large Particles & Objects	From 200μ through 1/4 diam. bolt to 4 lb. birds	Blade and intake damage, followed by blade failure	Nicked, dented, twisted compressor blades, damaged inlet screen facing. Stripped compressor stages, complete failure. Subsequent failure due to damage.
Ice	1/2" to 3" diam. hailstones, shaved ice	Intake damage, little or no blade damage	Damaged intake screens. No compressor damage, occasional flame-out
Salt Water	2 to 500μ > 10 c.c./min/engine	Salt incrustation with loss in power	Slight erosion, salt build-up, power loss, SFC rise
	"Solid" or "green"	Destruction of fan blading, bending of prop blading	Flame out, inlet damage, compressor damage

TABLE I-2 (continued)

Ingestion (cont.)

Methods for Combating Effects	GEM Industry Comments
Deflectors Coatings	
Debris Guards	No indication of major problems as yet
Debris Guards	
Deflectors, Coatings, Washdown, Inhibitors	Indications of appreciable salt build-up in lift system intake and ducting

Snow and Ice Accumulation

Type	Characteristic	Problem	Severity	Solution
1.	Accumulation on static vehicle in snow storm	Removal	As for existing transportation vehicles and aircraft	Standard de-icing equipment, snow removal equipment
2.	Icing at forward speeds in snow storms	Removal	As for existing transportation vehicles and aircraft	Standard de-icing equipment, snow removal equipment
3.	Operation over water in sub-freezing temperatures	Removal	More severe than (1) or (2)	More sophisticated equipment, or new approaches.

TABLE I-3
COMBAT ENVIRONMENTS
SUMMARY DATA

Noise

Major sound pressure levels at source	
Engines (turbine and piston)	$(12 \log_e(HP) + 95) \pm 10$ db
Fans 4 to 6 blade, 50 to 150 HP per fan. HP speed < 1100 fps	130 to 150 db
Over-all sound pressure levels, approximately 1 yard from complete vehicle	
In line with intakes (propulsion, lift, etc.)	110 to 130 db
In line with exhaust	110 to 130 db
Elsewhere	90 to 110 db
Approximate spherical spreading attenuation with distance from vehicle	
At 100 yds.	- 40 db
1000 yds.	- 60 db
10,000 yds.	- 80 db
Ambient noise levels	
Jungle	40 to 50 db
Coast	50 to 60 db
Quiet residential	70 to 80 db
Noisy commercial	85 to 95 db
Heavy traffic	90 to 100 db

TABLE I-3 (continued)

Noise

Approximate atmospheric attenuation with distance	Frequency (cycles per second) 10	100	1000	10,000
100 yds.	$.69 \times 10^{-3}$	3.78×10^{-3}	312.7×10^{-3}	31.2
1000 yds.	6.9×10^{-3}	37.8×10^{-3}	3127×10^{-3}	312
10,000 yds.	69×10^{-3}	378×10^{-3}	$31,270 \times 10^{-3}$	3120

Sand and Dust

Particle Size (Inches Diameter)	Lightly loaded GEM 30 lbs/sq. foot Max. Height	Max. Distance	Heavily loaded GEM 80 lbs/sq. foot Max. Height	Max. Distance
.005	300h	140h	600h	310h
.01	100h	70h	300h	150h
.050	25h	20-32h	60h	42-60h
.10	15h	27-32h	35h	34-56h
.50	4h	11-22h	15h	30-60h

Note: h = Vehicle operating height - feet.

Spray and Mist

Particle Size (Inches Diameter)	Lightly loaded GEM 30 lbs/sq. foot Max. Height	Max. Distance	Heavily loaded GEM 80 lbs/sq. foot Max. Height	Max. Distance
.005	500h	250h	1000h	600h
.010	250h	120h	550h	270h
.050	50h	30-38h	100h	66-80h
.10	25h	21-32h	60h	44-60h

Note: h = Vehicle operating height - feet.

TABLE I-3 (continued)

Radar Reflectivity

Likely Cross-Sections for CEMs

Vehicle Sizes	Radar Cross-Section (sq.ft.)
5 ton	5-10
100 ton	50-100

Requirements will be that vehicle radar cross-section be minimized. This is achieved by careful attention to configuration details, shaping, aspect; by shrouding propellers as much as possible; by utilizing coating media capable of high absorption at radar frequencies.

Infrared Emanation

<u>IR radiation</u> - Less than 10^{-9} watts/sq. cm. at 100 yds. in wavelength range 3 to 14 microns and less than 10^{-10} watts per sq. cm. at 100 yds. in wavelength range between 1.8 and 2.7 microns.

Typical requirements are those for Light Observation Aircraft. Maximum likely radiation intensity at vehicle is 60 watts/sq. cm. at exhaust or jet pipe - rapidly reduced and dispersed by insulation, structure and atmosphere.

Vulnerability (Relative)

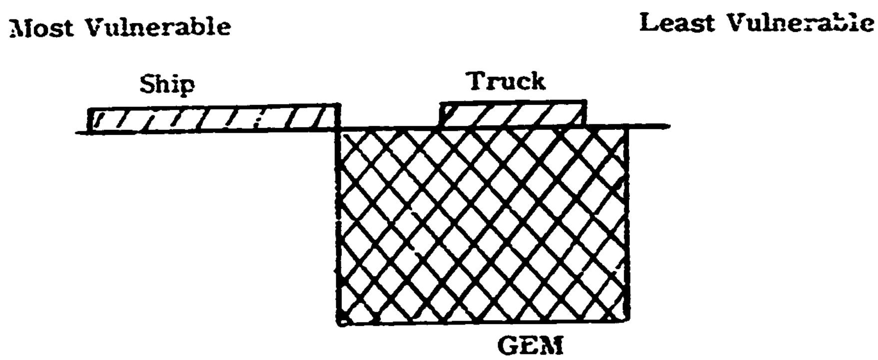


TABLE I-3 (continued)

Damage Protection

Normal (aircraft or any vehicle) plus additional for lift system.

Field Repairs

Same as for current Army vehicles, facilitated by modular construction.

Mine Field Operations

Mine Type	GEM Characteristics	Effect of Explosion on GEM
Pressure	to 30 lbs/sq. ft. 80 lbs/sq. ft.	If explosion is under center of vehicle - instantaneous increase of cushion pressure underside damaged by shrapnel - probably OK to continue. If explosion near edge - damage to annular jet ducting, reduction in efficient operation, damage to underside.
Acoustic	90 db → 100 db	
Magnetic	---	

TABLE I-3 (continued)

Nuclear Environment

Environment	Effect on Personnel*	Vehicle*
Radiation	In hover -	In hover -
	Intensified radioactive field--more protection required.	As for personnel--more careful choice of materials or more frequent inspections.
	At forward speed -	At forward speed -
	As for other vehicles of comparable speed.	As for personnel.
Blast pressure wave	Severe pressure fluctuations inside vehicle, unless adequate sealing and strength designed into cabin.	Uncontrollable vehicle motions if near explosion, may result in complete destruction of vehicle.
Temperature and flash	Skin burns, blindness, unless protected from high burst intensities.	Flash--little effect. Temperatures--engine stalling, fuel fires.

* These are predicated on vehicle being near enough to explosion for the quoted effects to occur.

Loadability at Sea

GEM limited to design operating heights of 4.5 to 6 feet, until safe unloading capabilities of cargo ships are extended to greater sea states.

Table 1-4
Summary of Military GEM Characteristics

Type	Class	Size: Length ft.	Size: Width ft.	Operating height required by surface: Normal	Operating height required by surface: Max.	% of year operation is feasible	% of land & water surface vehicle can be used	Slope capability required per cent	Surface treatment required before operating vehicle	Relative Vulnerability similar to: Ship	Relative Vulnerability similar to: Truck	Relative Vulnerability similar to: Air-craft	Special Equipment Required	Relative maintenance level: Truck	Relative maintenance level: Air-craft	Transportability possible by: Air-craft	Transportability possible by: Ship
Marine	I	40	20	1	1.5	Year round	Coastal waters and inland	5-10			x				x	x	x
	II	60	30	1.5	2.5	Year round	waterways	5-10	GEM		x		a, b*, c, d, e,		x		x
	III	100	50	2.5	4	61	80% of ocean	5-10	beaching		x		f, g, h, i, j, k,		x		x
	IV	150	75	4	6	70	80% of ocean	5-10	ramps		x		l, m, n.		x		x
	V	200	100.	6	10	84	83% of ocean	5-10			x				x		x
Amphibious	I	40	20	1.5	2.5	Year round	10% of beaches 10% inland	10	none		x				x	x	x
	II	60	30	2	3	Year round	40% of beaches 25% inland	15	none		x		a, b, c, d, e,		x		x
	III	80	40	3	5	Year round	70% of beaches 40% inland	15	none		x		f, g, h, i, j, k,		x		x
	IV	100	50	3	5	Year round	70% of beaches 35% inland	15	none		x		l, m, n.		x		x
	V	150	75	4	6	Year round	50% of beaches 30% inland	15	none		x				x		x
Overland	I	20	10	1	2	Year round	0% grassy plain only	15	fill in		x				x	x	x
	II	40	20	2	1	Year round	18% of land mass	20-30	ditches and bulldoze		x				x	x	x
	III	60	30	3	6	Year round	34% of land mass	30	path		x		a, b, c, g, o, p		x		x
	IV	80	40	3	6	Year round	32% of land mass	30	clear path		x				x		x
	V	100	50	3	6	Year round	28% of land mass	30			x				x		x

* See special equipment listing Table 1-5

TABLE I-5
SPECIAL FEATURES AND EQUIPMENT

Overland	Amphibious	Marine
a. Heating and air-conditioning for crew and cargo compartments.		
b. Shock and vibration isolation throughout the structure.		
c. Sand, dust and spray screens or deflectors over engine and lift system intakes.		
d. Engine boost capability for tropical conditions or jump capability.		
e. Obstructions radar or equivalent.		
f. De-icing for Arctic and Antarctic operations.		
g. Anti-corrosion treatment for tropical operations.		
	h. Cushion edges around vehicle perimeter, for structural protection.	
	i. Anti-corrosion treatment, to permit sustained sea operations.	
	j. Flotation capability.	
	k. Wheeled bottom for ground handling.	
	l. Strengthened or protected undersurface to resist impact.	
	m. Hydrophobic treatment on beaching surfaces, to reduce friction and wear.	
	n. Clear-vision windshield.	
o. Retractable support legs for static support above rough terrain.		
p. Erosion-resistant coatings in the lift and propulsion systems.		

1.2.2 Materials Environment

Further consideration is now given to the elements of the total environment as they affect the special materials used in the inflatable GEM. Indeed, these elements of environment must be considered for the system as a whole, as well as for its materials. Inasmuch as inflatable construction utilizes materials having properties differing from those of steel and aluminum--the normal military vehicle materials--environmental considerations which may affect these materials must be emphasized.

As discussed in reference 1, the total operational environment includes natural environments, induced environments, and combat environments. All three of these may be present during vehicle operations. In addition, natural environments also may have significant effects on the methods, procedures, and problems associated with the erection, storage and transportation of inflatable GEMs, both inflated and deflated, including packaging.

Again, reference is made to the generalized environmental analysis reported in reference 1. In particular, Chapter III of that study, covering induced and combat environments, should be reviewed. Other important source materials are reference 2, and reference 3 which provides further information in some of the areas defined in reference 2. Areas of environmental influences unique to the GEM or to inflatable structures require the application of some additional data, including special studies of surface conditions and materials, static electricity and downwash disturbances, and application of standards and specifications developed for the other services.

The order of presentation is as follows: first, the effects of natural environment and the probable extreme conditions which may be encountered in operational service and in storage and transportation; and secondly, a brief review of the pertinent elements of induced and combat environments emphasizing effects on inflatable structures and their materials. All of this material, including the extreme conditions mentioned, are applicable to the materials of construction, and to the assembled vehicle.

1.2.3 Natural Environment

The natural environment affecting inflatable GEMs is nearly the same as that for all military vehicles (more specifically, for amphibious military vehicles), and is primarily dependent on the geographical area of intended operations. Because of the world-wide commitments of the U. S. Army, all initial design parameters must be based on world-wide operations to the extent feasible, and this material is developed on that basis. The application of inflatable structures to a specialized GEM designed for a specific operational mission only would change the relative importance of some of the extreme conditions found in world-wide operations.

1.2.3.1 Important Environmental Factors

The following elements of natural environment are considered to be most significant in respect to the use of inflatable structures in GEMs:

- Temperature ranges and extremes.
- Absolute and relative humidity.
- Precipitation intensity and duration.
- Accumulated snow loads and icing rates.
- Wind speeds, including gusts.
- Atmospheric pressure.
- Solar radiation, including infrared, visible, and ultraviolet.
- Blowing snow.
- Blowing sand and dust, including particle sizes.
- Salt spray.
- Ozone concentrations.
- Fungus growth.
- Hailstorm duration and particle sizes.
- Abrasive qualities of surface materials.
- Abrasive qualities of vegetation.
- Other environmental considerations.

Temperature. The basic temperature range for world-wide operations is -25°F to + 115°F. For operations in hot desert regions the maximum temperature will be 125°F. For operations in arctic areas minimum temperatures will range down to -65°F (-75°F in Greenland and the Antarctic). The temperature range for shipboard operations is somewhat more limited, from -20°F to a maximum of about + 100°F. The temperature range chosen should correspond to all areas of potential operation. For a vehicle such as the LOTS carrier, the range of -25°F to + 115°F covers most of the areas of potential operation, inasmuch as the LOTS carrier will be able to operate only in areas open to ship navigation. Storage environments for all materials require consideration of high temperatures, as high as 155°F, for a period of up to four hours daily without solar radiation, and to 185°F with solar radiation of 360 BTUs per square foot per hour. Low temperature limits for storage should be -65°F for periods up to three days. These limits apply to packaged materials and components in transportation and long-term storage.

Humidity. Absolute humidity ranges for world-wide operation are from 0.1 grains per cubic foot at -25°F to 13 grains per cubic foot at 85°F where 7,000 grains equal one pound. This corresponds to relative humidities as low as 5% at 115°F and as high as 100% throughout the range -25°F to + 85°F. For operations in arctic areas absolute humidity will be very low (approximately 0.1 grains per cubic foot), although relative humidities to 100% are very common. For desert operations relative humidities will be as low as .5 grains per cubic foot at 125°F, yet still as high as 95% in the temperature range from 80° to 85°F.

Precipitation. Equipment must be designed to withstand the effects of two types of precipitation--both steady, wind-driven rain, and brief torrential downpours. Steady rain may have intensity of as much as 12 inches in 12 hours with a drop size of 2.25 milimeters predominant. These rains will be accompanied by winds up to 40 miles per hour. Torrential downpours such as are common in the tropics and in the temperate region thunderstorms range up to 7 inches in one hour with a drop size of approximately 3.2 millimeters. The peak rate is up to 2 inches in 5 minutes with a drop size of 4.0 millimeters. These downpours generally occur during periods of calm to light winds.

Snow Loads and Icing Rates. Heavy loadings of snow and ice on the vehicle will not only decrease performance but will also tend to clog engine and fan intakes, ducts, and windshields. A GEM in operational use should be considered equivalent to "portable equipment" by the

AR 705-15 definition. For this type of equipment the maximum snow load will be approximately 10 pounds per square foot with a density of about 6 pounds per cubic foot. GEMs which are stored in the open may receive snow loads up to 30 pounds per square foot--again at a density of 6 pounds per cubic foot. Inasmuch as most GEMs now being considered for vehicle application have large flat areas, the provision for snow loads may be a significant structural problem. Superstructure icing will occur in the arctic and sub-arctic (and antarctic) regions. World areas in which superstructure icing is prevalent are tabulated in the first chapter of reference 1. Maximum superstructure icing rates may be as high as 6 inches per hour.

Wind. For determination of the effects of wind, consideration must be given both to the maximum anticipated steady winds and to instantaneous gusts of higher velocity. Standard wind measurements taken at 10 feet above ground level appear to be directly applicable to a vehicle such as the GEM, without modification for height effects. Inasmuch as the GEM is significantly affected by wind loads, particular attention must be given to control system requirements for handling these loads. Normal operation conditions in land areas generally include maximum winds (highest 5-minute wind) of 40 miles per hour with gusts up to 60 miles per hour. In shipboard operations, wind speeds range up to 75 miles per hour with gusts of 100 miles per hour. It is presumed that during periods of such winds all operations in which GEMs could be involved will cease; therefore, these limits would apply to vehicles which are tied down aboard ship. In storage areas and depots, maximum winds may range up to 80 miles per hour--again assuming all material will be well secured.

Pressure. Atmospheric pressure ranges for GEM operations normally will be quite limited, except for air transportation. A range of 1,060 millibars to 887 millibars, corresponding to normal conditions from sea level to 3,700 feet altitude, is suitable for amphibious operations--in fact, for 90% of all overland operations in which GEMs could be used. For air transport, possible range of pressure is down to 116 millibars, corresponding to a flight altitude of 50,000 feet. The combined effects of low temperature and low pressure on a packaged inflatable GEM in unpressurized air transport may be severe.

Solar Radiation. Maximum solar radiation corresponds to temperatures of 125°F in the desert and approximately 115°F in the moist tropics. In the desert areas, solar radiation intensity will be as high as 105 watts per square foot (360 BTUs per square foot per hour).

This radiation is made up as follows: 50 per cent infrared, 44 per cent visible, and 6 per cent ultraviolet. For the moist tropics and shipboard operations, some of the radiation is absorbed by water vapor in the air so that a maximum of 90 watts per square foot may be anticipated. This includes 51 per cent infrared, 44.5 per cent visible, and 4.5 per cent ultraviolet. All materials will be subjected to long-wave radiation losses in extreme cold areas in which material surface may be cooled below the ambient air temperature. These radiation losses can be significant for long-term storage.

Blowing Snow. Blowing snow can clog intakes and electrical machinery as well as reduce visibility to zero-zero conditions. Even in the United States the effects of fine blowing snow have caused significant disruption in transport operations, including a recent storm in which a railroad's entire fleet of electric locomotives was immobilized. Design conditions for blowing snow should consider particles of one to three millimeters in diameter accompanied by wind speeds up to 40 miles per hour. The velocity of the GEM over the surface may increase the relative velocity of blowing snow, and sand and dust.

Blowing Sand and Dust. Blowing sand in the desert regions is a significant environmental consideration because the sand can clog small openings and cause breakdown of lubricated surfaces. Blowing sand is normally limited to heights of less than 5 feet above the ground. Approximately one-half of the sand remains within one inch of the ground. For GEM operations the disturbance of this sand by the downwash from the jets may be an important consideration. The maximum intensity of blowing sand may range up to 10 pounds per foot cross-section. Particle sizes will range from .18 to .30 millimeters diameter, with very few particles less than .08 millimeters. Wind speeds up to 40 miles per hour at 5 feet may accompany the blowing sand. Composition of the sand is mostly quartz. Blowing dust is distributed much more evenly throughout the atmosphere. Intensity of blowing dust will range up to 6 x 10^{-9} grams per cubic centimeter. Particle sizes will range from 0.1 micron to 10 microns, where one micron equals 0.001 millimeter. Again, winds up to 40 miles per hour may accompany the blowing dust, although winds lower than 15 miles per hour will be most common.

Salt Spray. Salt spray is a significant environmental element in ocean areas and in coastal areas. Corrosion due to deposits of salt on vehicle structure and on the engine [illegible]terior will greatly reduce the power plant output and structural life. Corrosion must be considered for coastal operations up to about 1,000 feet inland. In general, salinity of the oceans is approximately 3.5 per cent; that is, 35 parts dissolved materials per 1,000 parts water. Salt spray over the oceans is dependent on the wind. The intensity ranges from 4 micrograms per square meter at 10 miles per hour wind to 30 micrograms at 35 miles per hour winds, and up to 100 micrograms at 60 miles per hour winds. In surf areas salt spray intensity may be as much as 100 times as great as over the open ocean. Salt spray in rain over water areas and near the coast may range from 2 to 20 milligrams per liter. Military experience to date has shown that the results of even the most carefully designed salt chamber tests can be very misleading. The only satisfactory method of testing has been exposure of the test item to the actual environment in which use is intended.

Ozone. Ozone is the strong absorbent of ultraviolet radiation and causes degradation of rubber and other organic materials. Generally ozone is formed by the photo chemical reduction or organic pollutants. Natural concentrations of ozone near the surface range up to 3×10^{-2} parts per million, while in intense smog the concentration may exceed 5×10^{-2} parts per million. (Reference 4)

Fungus Growth. For all operations in the moist tropics and other high humidity areas, the growth of fungus on nonmetallic surfaces must be considered. The standards given in reference 5 or reference 6 (applicable to aircraft ground support equipment) should be used. Inasmuch as materials likely to be used for inflatable structures are normally considered to be subject to fungus growth, it is presumed that these materials will be tested to determine solutions to this problem early in their development cycle.

Hail. While hail is not a common occurrence in any part of the world, the effects of a heavy hailstorm can be catastrophic for military equipment, particularly that of aircraft-type construction. Maximum concentration of hailstorms is in the mid-latitude mountains and adjacent areas, e.g., a range of 5 - 10 hailstorms annually in Colorado, Wyoming, and Nebraska. Sizes of hailstones range from 0.5 inches to 0.7 inches in the United States and Western Europe, but up to 1.2 inches in India. Rare occurrences of larger hail may be encountered.

Abrasive Qualities of Surface Materials. Although a GEM is designed to operate without contact with the surface, occasionally during landing operations, or when encountering an obstacle, or in the event of power failure, the vehicle may scuff or slide along the surface.

Abrasion of the under surface of the vehicle due to sliding on beaches, desert dunes, and other surfaces, will be dependent primarily on the hardness and sharpness of the surface materials. On a world-wide basis, beach materials are predominantly sand with rounded-to-angular particles ranging up to 2 millimeters in size. On as much as 45 per cent of coastal beaches, larger particles such as gravel and pebbles range up to 2 inches in size (mostly rounded) and will be mixed with sand. In about only 5 per cent of the beaches will particles larger than 2 inches be encountered. For most GEM operations, mud flats will be a common environmental element; however, the abrasion of vehicle under-surface in occasional contact with mud surfaces is not considered to be serious. For operation over snow, hardness of 100 to 20,000 grams per square centimeter may be encountered for particles of 1 to 8 millimeters in diameter. Hardness of wet snow is much less. Abrasive qualities of other surface materials, including coral, bare rock, and pavement materials must also be considered. It is known that coral in its growing state (below the water surface) is very hard, angular, and may easily tear the bottom of a GEM which is operating as a displacement vessel.

Abrasive Qualities of Vegetation. Contact with coarse grasses, thorny underbrush, and rough trunks of trees and swamp vegetation may also scrape or tear the surface of the GEM. In this case damage will be due more to puncturing by sharp points than to abrasion by the vegetation material. Care must be taken in operation in mangrove swamps and other marshy areas where exposed roots, tree "breathers", and other sharp items pierce the water surface.

Other Environmental Considerations. For specific missions or specific geographical areas of operation there may be other elements of natural environment which must be considered in relation to the materials used in inflatable-type GEMs. It is possible, for example, that in some areas insects may create hazards, either by clogging intake screens and ducts or by chewing on materials used in construction. In most cases the specification for a particular vehicle application will include applicable environmental considerations peculiar to that use or area of operations.

Table I-6 presents a Summary of Environmental Conditions for Inflatable GEM Structures.

TABLE I-6
SUMMARY OF ENVIRONMENTAL CONDITIONS FOR INFLATABLE GEMS

	Region	Operational	Storage
Temperature (°F.)	Arctic Temperate Desert	-40 to+115 -25 to+115 -25 to+125	-65 to + 145 -30 to + 165 -30 to + 185
Humidity (Absolute)	Arctic Temperate Desert	> .1 grains/cu. ft. .01 at -25°F to 13 at 85°F > .5 grains per cu. ft.	
Precipitation	Steady	12 inches in 12 hours 2.25 mm. diameter drops 40 mph winds	
	Torrential	7 inches in 1 hr. to 2 inches in 5 min. 3.2 mm. to 44 mm. diameter drops Light winds	
Snow Loads & Icing Rates	Snow Load	10 lbs./sq.ft at 6 lbs./cu. ft.	20 lbs./sq.ft.at 6 lbs./cu.ft.
	Max. Icing Rate	6 inches per hour	
Wind Land	Max. 5 min.cont. Max. gust	40 mph 60 mph	80 mph
Sea	Max. 5 min.cont. Max. gust	75 mph 100 mph	
Pressure	Sea level to 3,700 ft. altitude	887 to 1,060 millibars	116 to 1,060 millibars
Solar Radiation	Desert	125°F with 360 BTU per sq.ft. per hr.	
	Moist Tropics	115°F with 310 BTU per sq.ft. per hr.	

TABLE I-6 (continued)		
Blowing Snow	Most severe characteristics	1 to 3 mm diameters blown at 40 mph
Blowing Sand and Dust	Sand	10 lbs. per sq. ft. of .18 to .30 mm. diam. with 40 mph winds at 5 ft.
	Dust	6×10^{-9} grams/cc. of .1 to 10 microns diam. with 15 to 40 mph winds
Salt Spray	Open Ocean	4 micrograms per sq. meter 10 mph wind 30 micrograms per sq. meter 35 mph wind 100 micrograms per sq. meter 100 mph wind
	Surf Areas	Up to 100 times open ocean quantities
	In Rain Near Sea	2 to 20 milligrams per liter
Ozone	Natural	3×10^{-2} parts per million
	Industrial	5×10^{-2} parts per million in intense smog
Hail	Max. freq. Size	5 to 10 hailstorms per year .5 to .7 inches in diam. normally, up to 1.2 inches occasionally
Abrasive Qualities	Surface Material Size & Character	Beaches - up to 2 mm in diam. rounded to angular particles } world-wide Gravel and pebbles up to 2 in. rounded in form } 45% of coastal beaches Particles larger than 2 in. } <5% of beaches
		Snow - hardness to 100 to 20,000 grains per sq. cm. Particles 1 to 8 mm. in diam.
		Coral - very hard and angular.
Abrasive Qualities	Vegetation	Most damage caused by direct puncturing by sharp points.
Insects		Clogging intake screens Possible attack on materials.

1.3 REVIEW OF CURRENT AND PROJECTED GEMS WITH PARTICULAR RESPECT TO THEIR STRUCTURAL DESIGN

1.3.1 Introduction

The vehicles reviewed and categorized in this section are drawn from those that are in operation or have been recently proposed, and are presented in order to indicate the probable characteristics and performance that may be expected from third generation vehicles. This performance growth has been borne in mind when reviewing the utility of inflatable structures and developing recommendations for future research and development in this area.

1.3.2 General Design Characteristics

1.3.2.1 Saunders-Roe SR N2

General Description

The SR N2 is a 25-ton to 35-ton vehicle, designed primarily to obtain operational experience, which will assist in the development of future commercial craft. The general layout is designed around a central load-carrying area to permit easy control of the C.G. regardless of the size and shape of the payload.

The buoyancy tank is incorporated so that the vehicle may float on water, and the bow shape has been determined from model tests carried out in towing tanks, to provide satisfactory hump behavior and acceptable impact forces while at sea.

Structural Aspects

Typical aircraft construction has been used in order to meet the necessary unloaded hover criteria. Materials used are high strength cladded aluminum sheets with protective coatings in keeping with normal flying boat practice.

Use of foam-filled light alloy panels in areas subjected to high noise levels is being made.

The basic structure is a composite beam formed by the boat-shaped buoyancy chamber, the deck above the air duct, and the main cargo area. Built in conformance with time-honored practice, this layout provides a rigid beam in bending and torsion.

The buoyancy chamber itself is divided into 15 watertight bilges and also contains two 425-gallon (imperial) fuel tanks laterally located and two 100-gallon water ballast tanks for fore and aft trimming.

Cargo Compartment

A 20 foot x 16 foot cargo area has been designed to accommodate 60 seated passengers or 100 standing troops, fully equipped. Alternate loads such as two jeeps or a 3-1/2 ton truck can be accommodated. Removable side and top panels facilitate loading.

Engines and Transmission Details

Four Blackburn A129 (NIMBUS) free turbine engines, rated at 815 horsepower (maximum control) are housed in pairs in an aft engine room and drive two fans and two reversible-pitch propellers. The fans supply pressurized air for lift and the propellers supply propulsive and maneuvering thrust. Each pair of engines is coupled to one fan/propeller unit. The shafting for the forward units passes through the cargo area roof beam.

Controls

The vehicle is completely controlled at all times by the propellers. Movement in the desired direction is achieved by pitch changes and swivelling of the propeller pylons through 30^{o} on either side of the center line.

Ground Handling

All normal maintenance, loading and fueling may be accomplished in and out of the water. On land, six steel pads are built into the base structure to provide a level and equally-distributed loading pattern.

Maintenance

Removal and installation of the fans is carried out by removal of the structure above the fans. This structure has been designed as a single assembly to facilitate this operation.

Engine room cowlings are also removable for "in situ" servicing and engine-changing as required.

Principal Characteristics

Over-all length	60 feet, 3 inches
Over-all width	29 feet, 6 inches
Power plants	4 x 815 BHP
Fuel	850 Imperial gallons
Typical cruise speed	70 knots
Cruise hover height	1.0 to 1.5 feet
Maximum hover height	2-1/2 feet

Comments on Design and Operation

Although GEMs are unlikely to be commercially competitive with conventional vehicles at gross weights less than 50 tons, Saunders-Roe consider that insufficient operational experience has been accumulated to justify going from the relatively crude SR N1 to a 50-ton gross weight machine. Consequently, the SR N2 has been designed primarily as an operational research craft for use by both military and civil operators, to obtain operational experience and to determine the limitations over water. One interesting facet of the design is the selection of fan/propeller/engine sizes based on their direct application to machines in the 50- to 100-ton gross weight size.

Experimental model tests carried out by Saunders-Roe on annular flexible trunks have now been confirmed to some degree by tests carried out on the SR N1. These trunks serve the dual purpose of reducing spray, thus increasing visibility at low speeds and increasing the maximum hover height for the same power. It can be expected that this development will be proven out on the SR N2 during 1962. When developed, Saunders-Roe see no problems to operating both ways, through breaking surf 6-8 feet high.

With the cargo floor designed to withstand loadings up to 300 pounds per square foot, a full range of military payloads can be tolerated. There are indications that the structural weights achieved on the SR N2 are close enough to provide a firm basis for projects specifically aimed at the military logistics application. However, in the LOTS carrier role with its high demand for extreme ruggedness, the SR N2 would have to be regarded as a minimal design. On this basis, one could estimate that to meet all of the design requirements for this mission, some 5 per cent to 10 per cent of the empty weight in additional "beefing up" material would be necessary.

From a transportability standpoint, the SR N2's configuration limits it to shipboard carriage, inasmuch as the structure has been designed as a homogeneous unit with no major unit breakdown.

At a selling price of some 400,000 pounds sterling ($1,120,000), a figure based on one to five quantities, the SR N2 yields a price per long ton of empty weight of $80,000, fully-equipped and ready to go.

1.3.2.2 VA 2 (Vickers-Armstrong - South Marston)

General Description

It is recognized that at this stage in GEM development, practical demonstration, particularly overseas, is essential with such a new type of vehicle. The difficulties attendant on transporting a large vehicle overseas have prompted Vickers to construct a vehicle small enough to be airfreighted and sufficiently developed to prove the engineering design and the practicability of application. It is probably the only GEM at this time being developed, from the start, to be air transportable. The vehicle readily breaks down into the main body unit and six separate ducting assemblies. The two vertical stabilizers and the propulsion powerplant are also capable of rapid breakdown. This is a utility vehicle, capable of carrying four or five people, with a speed of 40 knots and an endurance of 1-1/2 hours. The vehicle has immediate application as a fast, executive transport over sheltered and inland waters and for the transport of personnel over difficult terrain where existing vehicles cannot operate.

Structure

In keeping with the Vickers aircraft background, this machine makes full use, throughout, of aircraft materials properly designed and treated against the corrosive actions of salt-water operation. The basic design comprises a primary structure in the form of a stiff platform, taking the distributed pressure of the air cushion on the bottom surface. Fans, lift engines, and distribution ducts for the peripheral jets are mounted on this platform, with the remaining area providing cargo space or passenger cabin structure. Double curvature is reduced to the minimum and constructional details such as machined fittings are conspicuous by their absence. The primary structure is of the box beam type with bays approximately two feet apart. Transport joints are of the simple male and female spar type. Full use is made of treated, corrugated cardboard sandwiches for floorings and side panels. Simple truss mountings for the propulsion unit facilitate rapid breakdown, and the vertical stabilizers are attached with female and male union fittings of the simplest kind.

Cargo Compartment

A five-passenger seating arrangement is situated aft of the forward intake duct. Total payload is 1,000 pounds.

Engines and Transmission Details

Three Continental engines, using aviation gasoline, drive the two lift fans and separate propulsion system.

1.3.2.3 VA 3 (Vickers-Armstrong - South Marston)

General Description

The VA 3 has been designed as a true amphibian with a speed range of over 30 to 150 knots. As such, it will fit the transportation system in the role of an over-water passenger/goods carrier operating from land bases or terminals, or for use in areas presently inaccessible to other vehicles. At a cruise height of 8 inches, a 70-knot cruise speed is anticipated.

Structure

The primary structure comprises a buoyancy tank and a ducting system which form a load-carrying platform supporting the power units and the super-structure. The materials are mainly of aluminum alloy selected from a range of aircraft type gages.

The buoyancy tank is positioned low in the structure and provides 100 per cent reserve buoyancy and stability for the vehicle when operated over water in rough weather or in the event of a breakdown.

Passenger/Freight Compartment

The passenger or freight compartment measures approximately 17-1/2 feet long by 11-1/2 feet wide and can accommodate 24 passengers or 4,200 pounds of freight.

Engine and Transmission

Four Blackburn Turbo 603 engines, using kerosene, supply lift and propulsive power. The propulsive system employs two reversible, variable pitch, four-bladed propellers. The fuel tank is a single deep tank divided to make two compartments. Unpressurized refuelling is used. Submerged pumps and cross-feeds are incorporated.

Controls

In addition to the directional control provided by the propulsive engine and the propeller system, cable-operated control surfaces on the port and starboard coamings provide effective "keel" area to prevent drift and to assist turning.

1.3.2.4 VA 4 123-Ton Amphibious Ferry (Type 3032)

General Description

The VA 4 is a 123-ton ground effect machine designed as a medium size passenger and car ferry, operating over economic ranges of 50 to 200 nautical miles. Used in this way, the machine can carry 200 to 400 passengers with a car deck large enough to hold 14 large cars. The operational hover height of three feet allows the machine to negotiate six-foot waves. Cruise speed is 70 knots. Buoyancy tanks are incorporated for flotation and the bow has been shaped for over-water operation.

Structure

The main structure comprises a load-bearing platform and a rigid box section enclosing the freight decks. This design enables the machine to weather rough seas and to proceed as a displacement vessel in the event of lift system component failures.

Buoyancy tanks and the peripheral ducting system are integral with the platform. The tanks situated low in the structure provide a large reserve buoyancy and stability. Strong, heavily-plated bows are fitted to withstand the impact of heavy seas and surface debris.

The superstructure is comprised of cargo holds, passenger accommodations and facilities. The upper deck, which includes the navigating cabins, is mounted over the length and breadth of the main cargo and car deck.

Fuel tanks are accommodated adjacent to the engine rooms in the buoyancy chamber.

Engines and Transmissions

Lift and propulsion engines are aircraft-type gas turbines. The lift system comprises nine 1,000 SHP DeHavilland "Gnomes"; the propulsion system, two Rolls Royce "Tynes" (TY 12) of 4,400 SHP each.

The lift engines drive centrifugal fans and the propulsion engines drive fully reversible, variable-pitch, aircraft-type propeller systems through conventional aircraft-type transmission systems.

Controls

Directional control is achieved by air rudders and propulsion system engines with 60° of lateral rotation.

Roll stability is achieved by a central secondary duct running the length of the machine.

1.3.2.5 Design A (Detailed Project Study - Now in Construction)

General Description

This design is a 22-1/2 -ton machine designed for research purposes. The basic configuration consists of two propulsive units in tandem, mounted on a boat-shaped hull. Two vertical fins are located at the stern. The air cushion is produced by four fans which pump air out of the bottom through a peripheral jet nozzle. Stabilizing jets divide the base in both directions.

The pilot and copilot are located forward; the flight engineer, the passengers, and the equipment are located aft. Cargo space is centrally located and measures 6-1/2 feet high by 10 feet long and 18 feet wide. The hull bottom contains 12 buoyancy chambers. Hull lines have been developed in accordance with the over-water performance in a Sea State 3 condition.

Structural Aspects

The hull consists of aluminum alloy frames, longitudinals, and stringent stiffened skirts. The structure is welded where water tightness is required and riveted in other cases. In nonstructural areas, maximum use is made of low cost materials such as waterproofed plywood and hard-molded, reinforced Fiberglas laminate. With the exception of the bow structure, which incorporates frame spaces of 15 inches, frames are located at 30 inches. Sizes vary from bow to stem as required for strength. The top of the air duct is considered as an inner deck, the top of the hull being used as the shelter deck. Compartment between decks is designed to provide buoyancy in the event of damage and flooding of compartments below the air-duct deck. Fuel tanks occupy the space directly under the cargo area and run to half of the width of the buoyancy compartment. A companion way runs fore and aft on the center line of the machine. Each ducted propeller consists of an annular duct with an engine, gearbox, propeller, and cowling mounted to a center support structure.

Cargo Compartment

A 10-foot by 18-foot cargo area has been designed to accommodate 10,000 pounds of cargo at a designed overload gross weight of 27 1/2 tons. Total disposable load for both versions is 3.86 tons and 8.86 tons, respectively. A 3-foot by 2-foot hatch provides access through the upper deck.

Engines and Transmission Details

Four Solar Saturn free-shaft gas turbine engines, rated at 1,120 SHP (maximum control) are housed on top of the hull. Two units are mounted aft and drive all four fans through a mechanical transmission system. Two units are mounted in the propulsion ducts and swivel with the duct. All shafting is contained in the upper deck structure. A cross-over shaft links the two lift engines.

Controls

The vehicle is controlled as follows:

Side force is normally produced by rolling. Cushion fan speed is used to vary height. Fore and aft motion is produced by the tandem propeller pitch-power. Pitch and roll control are effected by differentially varying the fan inlet vane angles. Yaw is produced by a rudder on the rear duct. High-speed turns over water can be achieved by foils extending into the water.

Ground Handling

All maintenance, loading, and fueling may be carried out on land. Longitudinal skegs provide a level and equally-distributed loading pattern.

Principal Characteristics

Over-all length	67.5 feet
Over-all width	26.0 feet
Power plants	4 x 1,120 BHP
Fuel	1,715 U.S. gallons
Light gross weight	45,000 pounds
Heavy gross weight	55,000 pounds
Range	225 nautical miles
Typical cruise speed	70 knots
Typical cruise hover height	6 inches
Typical maximum hover height	9 inches

There can be little doubt that this design is the most advanced U.S. project at the present time. As a research vehicle, it will embody the latest "state-of-the-art" know-how with contributions coming directly from research programs in the U.K. and the U.S. As a prototype, it will also be capable of growth within and without the present null dimensions. It is anticipated that this vehicle will systematically evaluate handling qualities of GEMs, loading factors and other structural criteria. controllability, air cushion flow, duct losses, noise measurements, spray patterns, stability, and powering requirements.

It is also possible that a large number of appropriate missions suitable for the armed forces will be attempted and proven out.

While the design, particularly the design of the structure, is perhaps based on extremely conservative values, as a research machine required to operate in rough seas and other rugged environments it is probably the most realistic GEM to be configured at the present time and is in complete agreement with the Saunders-Roe series of test vehicles.

It is anticipated the over-all cost of the prototype vehicle, which is scheduled for completion in 1963, will not exceed $2.5 million.

1.3.2.6 Design B (Detailed Project Study)

General Description

This GEM study, recently proposed, is a four-fan, annular jet machine with mixed propulsion systems. The machine has a gross weight range of 33-1/2 tons to 42-1/2 tons overloaded. It has been designed to meet rigorous military specifications as an assault and support craft.

The cargo compartment starts immediately aft of the forward fans and extends to the stern. Clearance dimensions of this compartment are 43 feet long, 20-1/2 feet wide and 10-3/4 feet high.

To facilitate over-water operation, the design is built around a twin hull concept, the hulls being designed to flying boat practice with bow shape and dead-rise. Retractable landing pads are located at the extremities of each side hull.

Welded bottom plating on each hull is incorporated to insure an excess of buoyancy.

Structural Aspects

From a detailed analysis of materials and methods of construction, the designers decided to utilize 7075-T6 as the basic structural material, and aircraft-type construction utilizing rolled hats for stiffness.

The major load-carrying hull structure is formed by the cargo deck, a forward torque box, side beams, and a twin hull bow.

External skin is stiffened by roll-formed hat sections in the longitudinal directions, on the sides, and athwartships on the center bow, where flat or single curvature panels facilitate economical fabrication on automatic riveting and spot-welding machines. This skin-stiffener combination provides axial and shear strength for primary loading and distributes local pressure loads to frames spaced on 20-inch centers.

The cargo deck and its substructure consist of thin skins riveted to athwartship beams on 20-inch centers. Plywood panels cover the entire cargo area and are 3/8 of an inch thick. Longitudinal stiffeners on the bottom skin, which is welded, complete the structure.

The major longitudinal bending structure is made up of two side beams, the bottom skins, and stiffeners. The beams consist of the cargo compartment side walls the outer decks over the side internal air ducts, the hull sides, and the longitudinal members in the region of the side nozzles.

The twin hull bow structure is similar to flying boat design in all respects.

Cargo Compartment

A cargo compartment 43 feet by 20-1/2 feet has been designed to provide space for large, low-density items. A full width aft-loading door and ramp and full area overhead loading doors are incorporated. A 20-inch tie down grid pattern accessible through cutouts in the plywood flooring provides a flush tie down system. The deck is slightly convex to permit drainage.

Engine and Transmission Details

Two T64-GE-6 (normal rated power 2,680 ESHP) engines are mounted below the cargo floor with the engines facing aft, providing access from inside the machine for servicing, etc. In this location, one gearbox receives the power from both engines before distributing through shafting and gearboxes to the fans and propellers.

Controls

Two reversible, variable-pitch propellers are located behind the vertical stabilizers. Power is supplied as above. A twin annular jet curtain system incorporates propulsion louvers along the sides. An engine exhaust stability system divides the vehicle athwartships on the center line. This jet arrangement, together with the variable-pitch propellers and a horizontal stabilizer, is considered adequate for all design operations. Control functions are mechanically integrated into conventional operation controls.

Comments on the Design

The most important facets of this detailed design study are:

1. Compatibility with the LOTS carrier role.

2. Component breakdown suggests a high degree of transportability.

3. Structural design is representative of an open sea operational GEM.

1.3.3 Materials and Structural Concepts Now in Common Usage and Projected for Future Machines

1.3.3.1 Basic Structural Characteristics of GEMs

The preceding review of current and projected hardware in the GEM field indicates that no hard and fast rule has yet been laid down that would characterize these vehicles in the same way that other transportation devices are recognized. Aircraft, trains, ships, and trucks all fit into well-defined patterns where no doubt exists as to what the vehicle is intended to be. Consequently, loading criteria and general structural arrangements are constant or nearly so for each type of vehicle. A case in point is the ship with its central keel, hoop frames, bulkheads, and shear decks, a design unchanged from the earliest shipping times to the present day.

The Ground Effect Machine and its structure have yet to be finally evolved, in terms of shape, size, and loading criteria, although several relatively large vehicles have been built to fit the transportation of passengers and freight over water. The widely differing philosophy varies from the boat-shaped SR N2 to the rectangular barge shape of the VA 4. However, without exception, all first and second generation machines have utilized aircraft type, light alloy-clad sheets and box beam structures, a natural choice since their manufacturers have been largely airframe companies. It is also apparent that future machines now in the drawing board phase will follow the same design philosophy.

At this point it is well to note why the manufacture of GEMs has and will be in the hands of aircraft manufacturers for some time to come, certainly until a breakthrough in recirculation or power plant design is achieved.

From an aerodynamic point of view the GEM is inefficient. The best L/D that can be obtained at the present time at speeds close to 80 knots is of the order of 4 to 7. Compared with the average transport aircraft L/D of 10 to 20 or the ship L/D of 100 and up, it is evident that the GEM will be extremely high-powered, with a resulting poor operational economy, unless it is possible to operate at high payload to gross weight ratios.

Consequently, the GEM's economics are more dependent upon the efficiency of the structural layout than on any other single parameter. It is for this reason that it becomes necessary to use aircraft materials and design techniques for most GEMs. Perhaps the latest developments in the field are some indication of where the final selection of materials and structural layouts can be found in a current vehicle. The average skin thickness in the construction of the SR N2 is only .028 inch and the over-all structural weight is of the order of 17 pounds per square foot of cushion area (28% of the gross weight).

1.3.3.2 Strength Factors, Load Cases and Stressing Ranges of GEMs

At the present time there is considerable speculation on what loading criteria should be used in the design of GEMs. This is particularly true in the U. S., where little experience has been obtained on vehicles except as the result of extensive model test data. In the U. K., the general feeling is that loading criteria should be derived from existing aircraft practices in order to provide a firm basis for requirements which are now being considered by bodies such as the Air Registration Board and the Ministry of Aviation. A review of various companies' approaches follows.

1.3.3.2.1 Vickers-Armstrong (South Marston Ltd.)

General Considerations

For over-all strength assessment, the vehicle is considered in a number of representative conditions and placed in equilibrium under applied loads and reactive gravity and inertia loads. Normal, unaccelerated operating conditions provide no basis for strength, with factors of the order under consideration.

For over-water operation, it is envisaged that, in adverse operating conditions, a variety of impacts with appropriate reactive forces will occur.

In the cushion or airborne regime, this system of forces will be additive to the steady unit air and gravity loads. These combined conditions are deemed to cover emergency conditions due to engine or control failure since it is statistically improbable that adverse water conditions and engine or control failure would arise simultaneously. It is normal practice to design to lower factors in emergency conditions.

The proof and ultimate factors stated assume that all material used for stressed parts is produced to a recognized standard specification.

Strength Factors

In conformity with aircraft practice, the cases for strength assessment are based, on limit conditions, that is, conditions which are considered to be of such severity that they rarely occur.

The choice of such conditions must necessarily be somewhat arbitrary at the present stage of GEM experience and could imply speed and/or water roughness limitations if the behavior of the craft shows this to be necessary.

The factors provided on "LIMIT" conditions are:

Proof factor
Ultimate factor

The ultimate factor of 1.5 is used in all cases except crash cases.

Loading Cases Considered For Current GEMs

- Wave impact
- Crash
- Beaching, jacking, and towing
- Slinging
- Mooring
- Local and general water pressures on the craft's plating
- Control system loads.

For wave impact cases, a trochoidal wave of height to length ratio of 1 in 20 is used. It is assumed that the GEM can operate in waves of twice the design hover height.

Crash cases considered are the craft hitting quays, jetties, or obstacles at sea at high speed, or crash-landing on the shore. In all of these instances the safety of the crew and payload were considered. The inertia forces given below are used in the design of support structures for the crew and payload and for objects in the craft that could affect passenger or crew safety if they broke loose in a crash landing.

The inertia forces in terms of ULTIMATE acceleration are:

4g down to 3g up

6g forward to 3g aft

zero to 3g sideways

MAXIMUM RESULTANT - 6g.

The local water pressures on the bow of a GEM due to wave impact depend in a critical manner on the bow angle and dead-rise angle as well as the relative speed at impact. These pressures may be readily determined when the above angles are optimized and the speed of the GEM is known.

For initial stressing cases, the local water pressures on the bow are taken as 30 pounds per square inch unfactored.

All other areas of the GEM exposed to water impact are designed to withstand a water pressure of 5 pounds per square inch unfactored. This pressure can act on sufficient area of the buoyancy platform to give the acceleration required in the wave impact cases.

From a fatigue standpoint, an operational life of 20,000 hours is used.

1.3.3.2.2 Saunders-Roe Ltd. (A Division of Westland Aircraft Co. Ltd.)

Detailed design work on GEMs started at this company in October 1958, the result of which was the SR N1. When the design was laid down, there was very little information upon which to base the design stressing conditions. However, since the machine was primarily intended for operation over water, it was assumed that the engine failure case would provide a suitably severe design criterion. The assumptions made were that there would be a sudden engine failure when the machine was operating over waves of critical length and two feet high from trough to crest, leading to two main conditions.

The first case is the condition where the machine has been rotated by striking a wave with its stern in such a way that the maximum slamming load comes onto the bow. This leads to about an 8g acceleration on the pilot which is equivalent to a load approximately equal to 1.25

times the weight of the machine acting on the bow. Furthermore, it was assumed that this could apply to angles of yaw up to 45 degrees. This virtually stresses the attachment point of the outer rims, the engine mounting and determines such things as the fan clearance.

The second case is the condition where the machine just dives over one wave and ploughs into the next, so that a maximum longitudinal deceleration of about 1-1/2g is obtained.

From a crash standpoint, such as if the machine hits a piece of wood so that the front structure is crumpled, this gives a maximum acceleration of about 4g.

Water pressure acting on the machine is considered to be at a maximum pressure of 10 pounds per square inch on the bottom plating.

These criteria were based on models with no cushion, so that when the engine failed, the accompanying descent did not illustrate the decay effects of a cushion; consequently, the vertical rate of descent of the inert model was much greater than would otherwise be the case.

When allowance is made for the cushion, the maximum acceleration on the bow is only of the order of 1/2g instead of 8g, indicating that the criterion is more appropriate to speeds of 90 knots than to the speeds of 30 to 40 knots consistent with the SR N1 performance characteristics. It was therefore logical that these design criteria should be used for the development of the SR N2, resulting in a primary stressing case of 12g at the bow. This acceleration is approximately equivalent to a force equal to the gross weight of the machine distributed over the bow area. As pointed out by Saunders-Roe (reference 8), this is in keeping with all forms of transports where the design stressing condition is equal to the gross weight applied at its extremities.

1.3.3.2.3 Ryan Aeronautical

A structural criterion using a minimum number of design conditions constituting reasonable maximum values for the critical loading conditions has been developed (reference 9) by this company. This criterion has been used across the board in comparing GEMs of all kinds and sizes.

During 1961 this company analyzed the machine developed by Lt. Col. J. L. Wosser and Lt. Cmdr. Van Tuyl in some detail from a structural standpoint. The specifications and performance data available on this machine were utilized.

In the development of structural criteria, Ryan assumed that the primary structure would be designed in accordance with accepted aircraft design practices.

All loads and load factors in these criteria are limit loads unless otherwise specified. An ultimate safety factor of 1.5 is applied to all limit loads.

The primary structure was designed to prevent permanent deformation at ultimate loads. The secondary structure is designed to prevent deformation at limit loads and to prevent failure at ultimate loads.

A summary of load factors used for the Ryan structural design and analysis is shown in Table I-7.

1.3.3.2.4 Company Design A (Now in Construction)

General Considerations

Three cushion-borne conditions have been assumed to be of major importance as far as the design of basic structure is concerned. The first of these simulates an impact with the wake of another craft at the maximum speed. The second condition corresponds to a wave impact in rough water. The third condition simulates an unsymmetrical wave impact in rough water during a turn.

The maximum bow and stern impact loadings for the second condition were established in order to produce a load factor of approximately 8g's at the crew compartment and is in general accord with the Saunders-Roe data for the loads that can be resisted by crew members. The bow impact loa s of the first condition are arbitrarily half of those for the second condition since it is intended that high-speed operation would take place only over smooth calm water with the exception of a possible impact with the wake of another craft. The bow and stern loads of condition three have again been arbitrarily established as three-quarters of those given by the second condition.

TABLE I-7
RYAN DESIGN LOADING CONDITIONS

Component	Design Condition	Limit Load Factor n_x	n_y	n_z
Primary Structure (Basic Framework Including Bow)	Gross Weight, Flight	+ 2.00	± 1.00	± 1.25
	Gross Weight, Landing	0	0	+ 2.00
	Gross Weight, Collision	-3.00	± 1.33	0
Engine Pods & Supports	Maximum Loads Acting Simultaneously	+ 2.00		+2.00
		-3.00	±1.33	-1.25
Secondary Structure (Enclosures, etc.)	Maximum Loads Acting Separately	+2.00		+2.00
		-3.00	±1.33	-1.25
Crew and Personnel Safety Structure	Minor Crash Loads Acting Simultaneously	-6.00	±1.00	-2.00
Ducting, Nozzles & Air Pressure Loaded Structure	Maximum Calculated Pressure	1.33 x Calculated Pressure		
Flotation Hull	Landing Pressure	No factor applied to calculated pressure		
Exposed Deck	Water Pressure	No factor applied to calculated pressure		

Both the second and third conditions have been fixed as general conditions; in other words, they are applicable to all speeds but the bow and stern loads are maintained constant. One specific case that the second condition will cover is a 50-knot speed in a Sea State 3. In a similar way, the third case will encompass the situation where the radius of turn is equal to 24 craft lengths at a speed of 50 knots.

Deck Loads

To Specification MIL-A-8865 (ASG). Limit floor loads for personnel floors - 300 pounds per square foot.

Propeller Support Loads

The inertia load factor requirements were established at one-half the inertia load requirements outlined under General Design Information of MIL-E-17341A. ($\pm$ 14.0 vertical, $\pm$ 11 athwartships, $\pm$ 7.5 fore and aft).

Docking Loads

A load factor of 1.0 at an impact velocity of 3 feet per second covers impacts between the craft and a dock.

Crash Loads

The ultimate crash load factors are based on MIL-A-9965 (ASG). (Longitudinal load factor shall be 20.0 and shall act anywhere within 20 degrees of the longitudinal axis. The vertical load factor shall be directed downward, normal to the longitudinal axis, and shall be equal to 10.0. The load factors shall act separately.)

Hoisting Loads

The vertical load factor of 2.0 is taken from the aircraft hoisting requirements of MIL-A-8862 (ASG).

Ground Contact Loads

Two conditions when the craft is supported by irregular surfaces during ground handling are considered. The first condition is an unsymmetrical two-point contact across the corners of the craft causing a 2g vertical resultant. The second condition is a symmetrical two-point contact across the center line of the craft causing a 2g vertical resultant.

Repeated Wave Impact Loads

A total life of 2,500 hours has been specified, assuming two hours of operation per day, five days per week, fifty weeks per year, for five years. In addition, a particular distribution of hours of operation in various speed ranges has been assumed which provides for one-half the service life being spent in the 40- to 60-knot speed range.

In the determination of the total number of potential wave impacts, an average wave length (crest to crest) of 62.5 feet has been assumed, corresponding to a Sea State resulting from a 15-knot wind based on $\overline{\lambda}_{av} = 0.278V^2$.

1.3.3.2.5 Company Design "B" (Detailed Preliminary Design)

General Considerations

In this design, five loading conditions were considered, based on empirical considerations which conservatively cover the infinite variety of possible operating loads. They were wave impact, flotation in rough seas, ground landing, crash, and hoisting conditions.

Wave Impact

It was considered that there was sufficient evidence that a GEM would either deflect a wave crest or be deflected by momentarily increased cushion pressures as the wave was approached. Once again, the experience of Saunders-Roe was utilized and calculations were carried out to check the transient cushioning effects. To some extent the selection of wave height and GEM speeds have been governed by this work. A Sea State 3 (3 to 5 feet trough to crest), used in conjunction with MIL-A-8864 and the above work, resulted in the assumptions that the design wave impact condition would consider a 10-foot wave with a length to height ratio of 20, and that the GEM would be in a level steady flight altitude prior to contact with the water slope following the characteristic sine-shaped wave. GEM hover height at 60 knots was established at three feet. Operations at higher speeds would be restricted to calmer water.

Rough Sea Flotation

Due to the complexity of estimating hogging and sagging loads, a simplified condition was assumed, leading to a 1.25g load factor.

Ground Landing

Specification MIL-A-8862 specified a jacking load factor of 1.35. A lateral load factor due to small velocities during landing was used in addition to the jacking load.

Crash Loads

It was considered that the fully loaded GEM should be designed to withstand crash load factors, to prevent injury of crew or passengers. Specification MIL-A-8865 was used (8g forward, 1.5g aft, 1.5g side, 4.5g down and 2g up). These factors govern the design of the major items such as the cargo floor, engine and fuel tank structure, etc.

Hoisting

Hoist fittings and carry-through structures were designed to a limit load factor of 2g in accordance with MIL-A-8862.

1.3.3.3 Discussion and Summary

The Vickers, Saunders-Roe. and Ryan data on the strength requirements for GEMs are the only published statements on this subject at the present time. The first two companies are both builders of research craft that have for some time been exposed to over-water and over-land operations. In the case of Saunders-Roe, the SR N1 has shown general agreement of the estimated loads with model tests and has already accumulated several hundred hours of operation over water.

However, it would be unwise, at the present stage of development, to place too much emphasis on any one approach, regardless of the fact that operational suitability has been demonstrated to the satisfaction of the designer. With this in mind, a number of unpublished reports were examined with respect to design philosophy, structural criteria, and general stressing aspects. It has been necessary to refer to them as Design A or B in order to protect proprietary interests. Table I-8 has been drawn to summarize the leading particulars and to provide some guidance on the transportability aspects. Modular and inflatable possibilities have been examined for each design and the likelihood of utilizing these techniques has been indicated in a qualitative fashion.

TABLE I-8

SIZE, PERFORMANCE AND STRUCTURAL CHARACTERISTICS OF CURRENT AND PROJECTED GEMS

		Saunders-Roe SR N2	Vickers-Armstrong VA-2	Vickers-Armstrong VA-3
Height - ft.			10-1/3	17-3/4
Width - ft.		20-1/2	15	25
Length - ft.		60-1/4	28-1/3	52-1/2
Speed - knots		70	40	70
Gross weight - tons		30	3-1/2	12-1/4
Range - N.M.		200	60	87
Payload - tons		10	3/4	2-1/4
Hover height - ft.		1.0 to 1.5	3/4 to 1.0	3/4 to 1.0
Structure weight - %		33% approx.	45% approx.	40% approx.
Major material used		Aircraft clad aluminum sheets	Aircraft clad aluminum sheets	Aircraft clad aluminum sheets
Type of structure		Flying boat hull with close pitched stringers and frames.	Widely spaced frames with box beams and trusses.	Widely spaced frames with box beams and trusses.
Design criteria		Sheltered water operation (see text)	Sheltered water operation and restricted overland amphibious duties.	Sheltered water operation and restricted overland amphibious duties.
Transportability	Land	Marginal, would require special preparations.	Truck	Marginal, would require special preparation.
	Sea	Freighter deck & self ferry	Freighter	Freighter
	Air	Not possible	Transport aircraft	Not possible
Conversion Capability	Modular	Propulsion units & vertical stabilizer	This vehicle is designed on a partial module concept.	Small changes in design would provide modules.
	Inflatable	Not possible	Complete vehicle	Marginal

TABLE I-8 (continued)

		Vickers-Armstrong VA-4	Britten-Norman CC-2	William Denny D.1
Height - ft.		42	8-1/2	
Width - ft.		58	17	10
Length - ft.		173	27	66
Speed - knots.		70	50	17
Gross weight - tons		123	2-3/4	4-1/2
Range - N.M.		250	0 - 500	
Payload - tons		35	1/2	
Hover height - ft.		3.0	1.0	Water contact
Structure weight - %		43%	26% approx.	
Major material used		Aircraft clad aluminum sheets	Aircraft clad aluminum sheets and plastic foam	Sheet steel and plywood
Type of structure		Box beams and trusses - Aircraft practice.	Lightweight aircraft construction. Large unsupported panel areas.	Boatbuilding frames, decks and bilge compartments. Sidewalk keels.
Design criteria		Operation in 6 feet seas.	Occasional water impacts in sheltered waters at 50 knots	Inland waterway operation.
Transportability	Land	Not possible	Truck	Marginal
	Sea	Self ferry	Freighter	Freighter
	Air	Not possible	Would require breaks	Not possible
Conversion Capability	Modular	No advantage	Marginal	No advantage
	Inflatable	No advantage	Complete vehicle	No advantage

TABLE I-8 (continued)

		Design "A"	Design "B"	Ryan Aero, Wosser/ Van Tuyl
Height - ft.		24-3/4	23	19-1/2
Width - ft.		22	40	40
Length - ft		67-1/2	66-1/2	68
Speed - knots		70	60 - 145	50
Gross weight - tons		22-1/2	33-1/2	31-1/4
Range - N.M.		225	200	200
Payload - tons			15 - 24	15
Hover height - ft.		1/2 to 3/4	3	2
Structure weight - %		41-1/2	24%	
Major material used		6061 - T6 weldable aluminum alloy in extruded sheets reinforced fiberglas & plywood	Aircraft aluminum alloys	Aircraft aluminum alloy sheet 6061
Type of structure		Lightweight boat techniques and aircraft stressing	Aircraft construction throughout.	Aircraft construction throughout
Design criteria		Rough water up to Sea State 3.	Sea State 3.	Sea State 3.
Transportability	Land	Not possible	Requires preparation	Requires preparation
	Sea	Self ferry or tow	Self ferry	Self ferry
	Air	Not possible	Not possible	Not possible
Conversion Capability	Modular	Not possible	Small design changes would be required	Small design changes would be required.
	Inflatable	Not possible	Some components	Some components

TABLE I-8 (continued)

		Ford Aeronutronic LOTS Carrier	Saunders-Roe SR N3
Height - ft.		17	
Width - ft.		35.7	29-1/2
Length - ft.		51.3	70-1/2
Speed - knots		70	80
Gross weight - tons		22	40
Range - N.M.		300	
Payload - tons		11	16-3/4
Hover height - ft.		5	2
Structure weight - %		21%	33%
Major material used		Aircraft light alloys and honeycombs	Aircraft clad aluminum sheets
Type of structure			Flying boat hull with closely pitched stringers and frames.
Design criteria			
Transportability	Land		Marginal, would require special preparation
	Sea		Freighter deck & self ferry
	Air		Not possible
Conversion Capability	Modular		Propulsion units and vertical stabilizer.
	Inflatable		Not possible.

From these data there can be little doubt that two distinct philosophies are in effect at the present time, although it appears generally true that all GEMs are using modified aircraft techniques for construction and powering. The predominant philosophy appears to utilize Saunders-Roe design criteria or modifications of the main loading conditions where greater hover heights relieve the frequency of encounter of the design wave height system. However, in some cases there is little or no provision for drop loads from maximum hover heights, a condition resulting in maximum bending of the structure. The remaining philosophy is hard to substantiate for a military vehicle, which must be rugged, dependable, and capable of long service life. In an attempt to correlate these variations, three methods of comparison have been used. The first, shown in Fig I-1, plots the gross weight of the machine against the percentage of the gross weight required for structure; structure in this case includes all primary, secondary structures, as well as the ducting where ducting constitutes an integral part of the total structure. From this figure, the differing design philosophies are evident with variations as wide as 14 per cent to 45 per cent over the size range. Over the 30- to 40-ton gross weight range, however, the range narrows from 28 per cent to 35 per cent with perhaps a reasonably good mean of 33 per cent.

In terms of the structural weight per square foot of cushion area, over the same size range the same wide variety of design philosophy is evident. Figure I-2 has been drawn for a size range in keeping with the LOTS mission. It will be seen that for five vehicles, roughly in conformance with this mission characteristic, the variation in structure weight per square foot of cushion pressure is between 10 pounds per square foot and 25 pounds per square foot.

One other attempt has been made to correlate meaningfully the various designs, using the well known "Driggs" parameter for aircraft fuselages and hulls (shown in Fig. I-3), where the total surface area of the enclosed structure is used to obtain preliminary weight estimates. From a mean line drawn through the Vickers data, no satisfactory correlation can be expressed except to say that above the line, the vehicles may be characterized as rugged, heavy-duty type vehicles while, below the line, vehicles must be somewhat suspect, dependent upon the operational condition expressed by the designer to relieve frequency and size of load applications.

At this time, it would appear that, until service requirements are agreed to by industry, no reliable weight estimates or suitability for operations criteria can be formed.

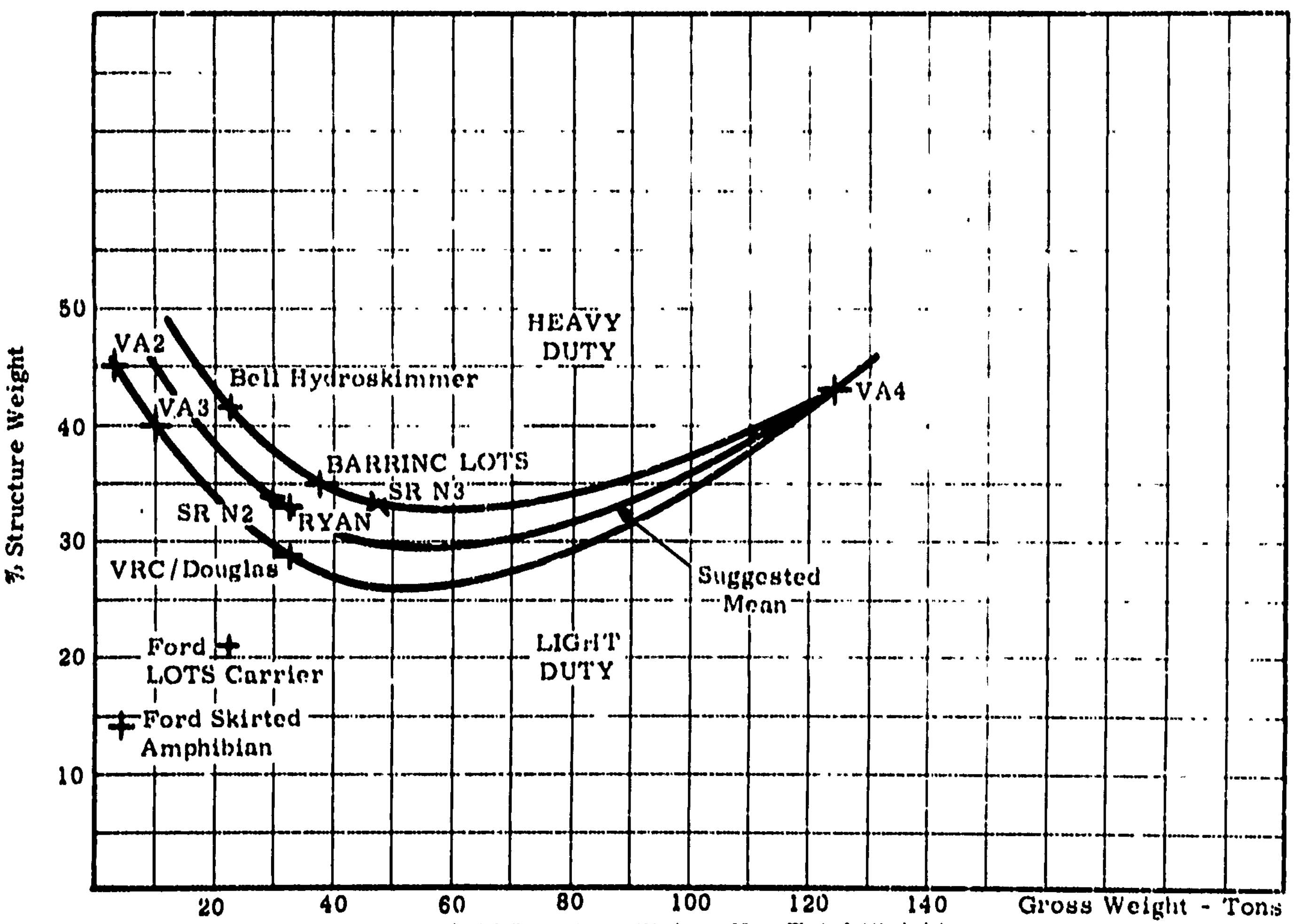


Fig. I-1. % Of Structure Weight Vs. Total Weight

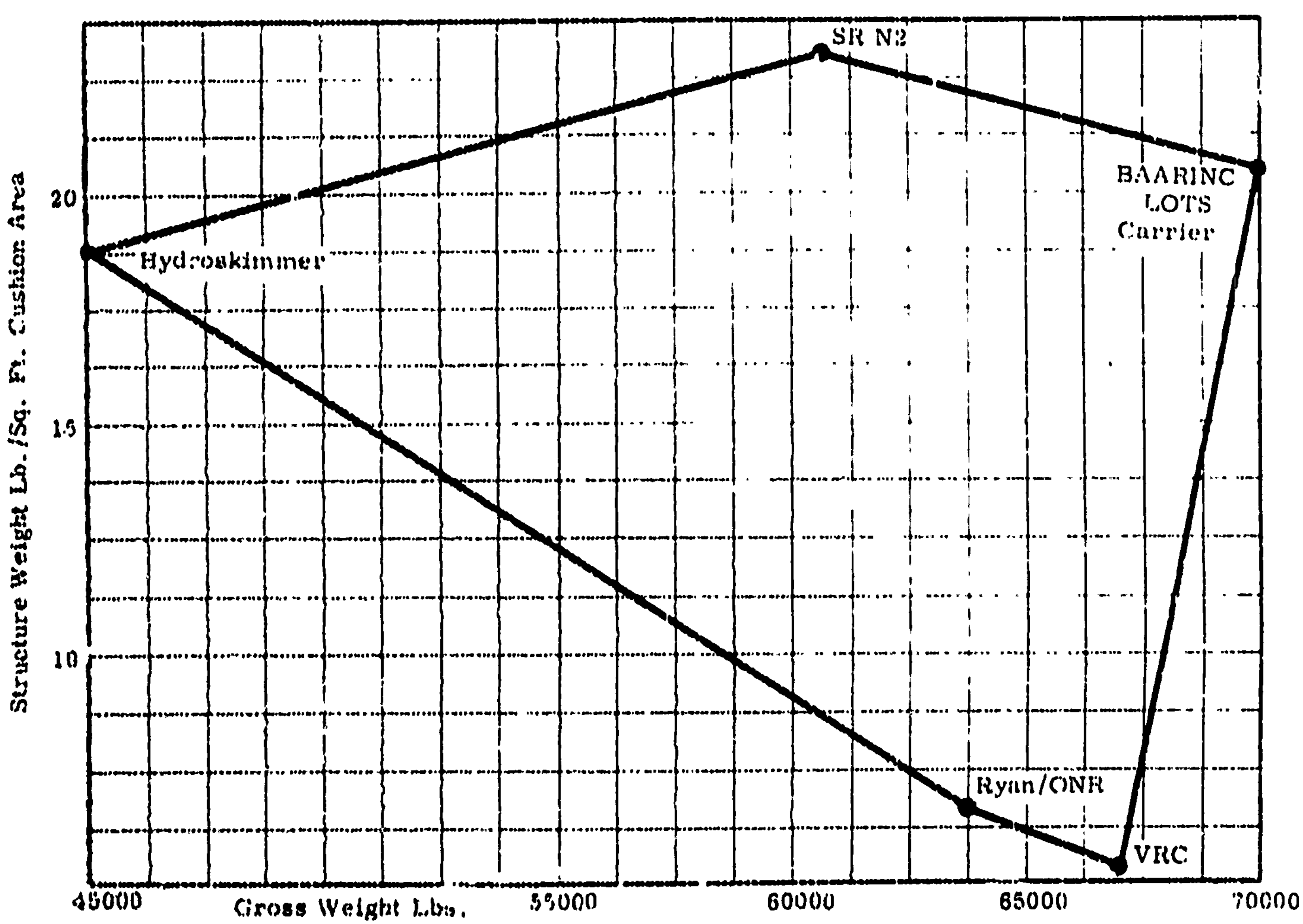


Fig. I-2. Structure Weight Variation for Similar GEMs

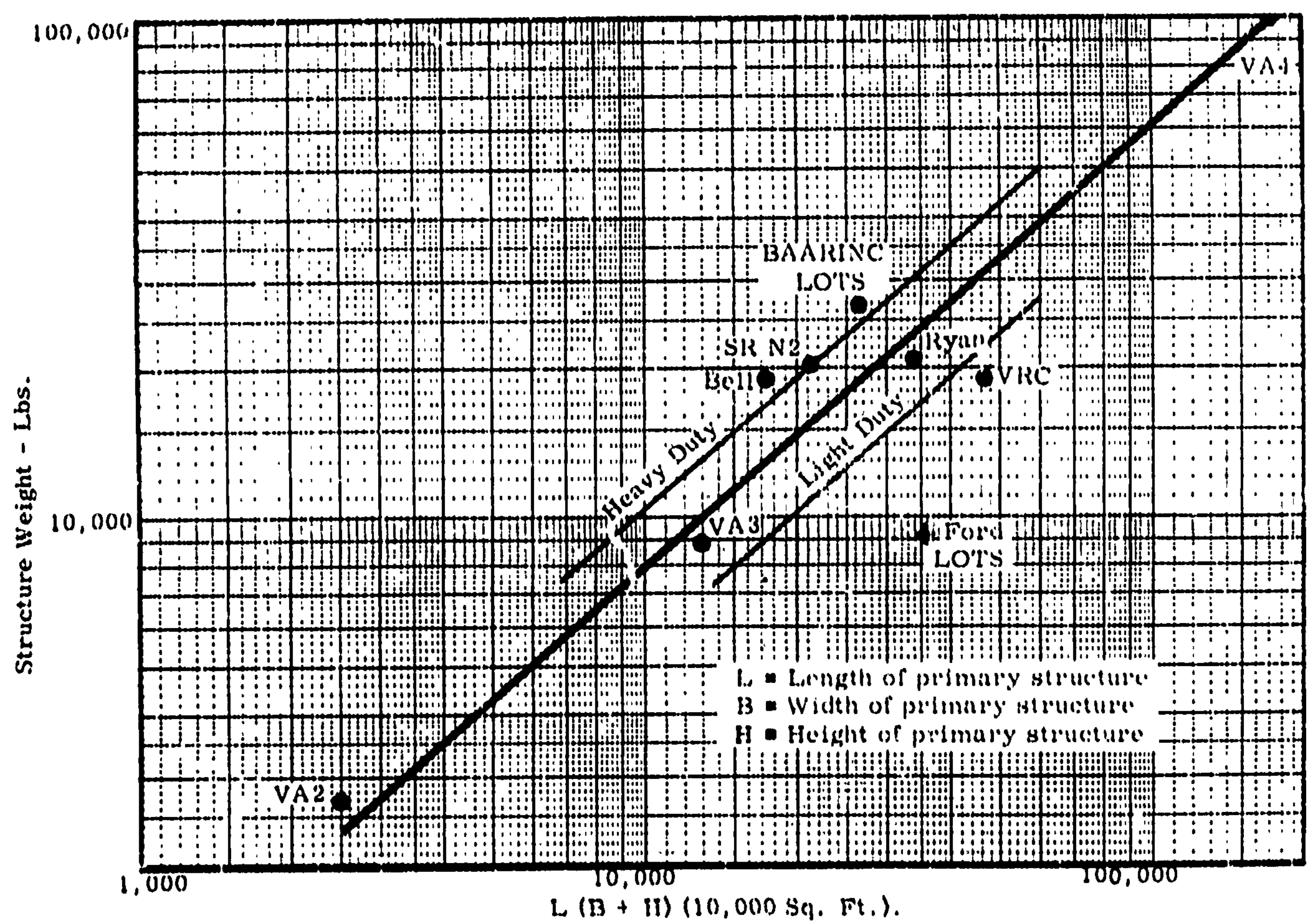


Fig. I-3. Structural Weight Vs. Total Wetted Area Of Primary Structure

1.3.3.4 Conclusions and Recommendations

1.3.3.4.1 Conclusions

1. No firm structural criteria exist for Ground Effect Machines.

2. Loading cases and load factors being used at present vary according to the particular design philosophy, viz; low hover heights - high loading cases and factors; high hover heights - low loading cases and factors.

3. Shape of cushion area and ratio of planform area to cushion area are major considerations in assessing structural and payload area efficiency, viz., the bow-shaped hull of the SR N2, and its low cushion area to planform area result in a dense structure leading to high unit weights per square foot of cushion area. The rectangular planform and high ratio of cushion area to planform area of Design B lead to low unit weights per square foot of cushion area. However, in both cases, the percentages of structure weight to gross weight are approximately equal.

1.3.3.4.2 Recommendations

1. Studies are required to make a detailed analysis of existing designs and projects with respect to the interchange of power plant weight for greater hover heights and reduced structure weight.

2. Flexible base components such as annular trunks require further analytical and experimental work to verify current claims. A better understanding of the aerodynamic and structural problems associated with flexible structures is required.

3. A means for evaluating the correct choice of structural criteria for a particular GEM is required. In this respect, current and projected environmental conditions must play an important part.

1.3.4 The Probable Utility of Inflatable Structures in Vehicles Reviewed and in Future Developments

The economy of GEMs is vitally dependent on the vehicle structure weight, which itself is vitally dependent upon the loading conditions utilized. To date, the correct loadings for GEM designs, particularly over-water, have not been determined. Consequently, the designer has to rely on his own intuition, experience, and ingenuity in deciding what loading conditions he shall use. The results so far have been, as indicated, very widely varied; the heavy structures arising because the critical loading conditions comprise impact conditions during which the precise loads are unknown, but undoubtedly large, and the light structures developing as a result of discounting the probability of impact, with consequently lighter structure.

One very immediate application of inflatable structures consists in providing an outer structure capable of absorbing impact loading and smoothing out the extreme transient loads, thus enabling the heavy primary structure to be assigned to carry relatively steady loads of appreciably reduced magnitude compared to the impact loads themselves. A direct parallel to this situation in the past history of transportation was the introduction of the pneumatic tire. As anyone who has tried to drive a car with solid tires over rough roads can verify, the pneumatic tire extended the vehicle service life and speed capability immensely; the same can be anticipated for GEMs.

The development of purely inflatable GEM structures is a possibility, probably more remote than the above application since the previous section has indicated the severity of the basic vehicle loading, and this would tend to demand pressures well beyond present practice, or sizes that would not be compatible with current thinking in GEM configurations. The possibilities in this area are the subject of Chapter III of this study.

As GEMs develop in the future, improvements in lift efficiency, mechanical efficiency, fuel consumption, and structural techniques, will tend to expand the range of practical GEMs from those currently under development for commercial and military use to much larger GEMs and for some specialized applications, much smaller GEMs.

In the case of the larger GEMs, some consideration should be given to enabling appreciable power reduction for a given lift, in order to offset the rapid increase in vehicle weight with size. The most probable road to success in this direction is to reduce the mass flow required in supporting the vehicle, and this in turn requires relatively small clearances between the vehicle air-jet system and the surface at all times. In these larger vehicles, the obvious need for flexible ducting of some kind to satisfy this requirement can probably best be met by inflatable construction, since some measure of bending stiffness would be desirable in the ducts, and it would be difficult to provide this without going to a great deal of material if approaches other than inflatable structures were used.

For smaller GEMs in the future, there may possibly be a case for developing a complete vehicle of inflatable structure, making the whole vehicle take the loads imposed, rather than a single beam or plate. Such an approach may be perfectly feasible structurally for small one or two-man runabouts, carriers, fast short-range assault vehicles, and so on. From the viewpoint of both structural efficiency and performance, such an approach could be satisfactory; however, the environmental conditions for its operation would most likely determine the success of its application.

1.4 MATERIALS AND THEIR CHARACTERISTICS FOR INFLATABLE STRUCTURES

1.4.1 General Considerations

Since inflatable structures may be constructed of any flexible pressure-tight material, a wide range of possible materials and material constructions can be projected. This section outlines those materials that have been considered, indicates those that may warrant investigation, and summarizes the characteristics that have been established for those materials suitable for inflatable structures.

The materials from which the skin of an inflated structure is composed generally have two major components: the fiber from which the cloth is woven and the elastomer that is used to seal and protect the fiber.

Fibers that have been considered range from cotton through a wide range of man-made fibers. The choice of fiber is determined from two basic requirements, plus other important considerations. First, it must have adequate strength, and second, it must possess a sufficiently long operational life.

The elastomer is chosen to provide adequate pressure seal capabilities and satisfactory resistance to environment. The combination of the two provides an effective structural material, as indicated by relative strength/weights in Figure I-4. This figure shows the ratio of the material ultimate tensile strength in pounds per square inch to the effective material density for a wide variety of materials suitable for use in sheet form. The ratio for the fiber structures is shown for both the fiber alone and for typical, practical, cloth construction, providing adequate air-retention and environmental capabilities. The values indicate that, for ultimate strength, the fiber materials can be comparable to metal sheets. Thus, adequate vehicle life can be ensured when strength factors are taken into account.

I.4.2 Fibers

The most desirable characteristics of fibers for inflatable construction, are high strength-to-weight, ease of manufacture and subsequent working, low cost, compatibility with easily available coatings, and good energy absorption characteristics. Some of these characteristics are shown in Table I-9 for a wide range of fibers that are in use in the textile industry today. The work of reference 10 was used extensively in the construction of this table.

It can be seen from the table that many fibers in common use today possess high stiffness and high strength, and would appear to be suitable for inflated structures. The experience of fiber manufacturers, cloth weavers, cloth coaters, and inflatable structure designers and constructors has been that nylon and Dacron fibers are, in general, the most acceptable materials to use, with some preference for nylon mainly because of its lower cost and the greater experience in handling and manufacturing available for design and construction. It should be noted that most manufacturers' final products have not been weight critical so that a greater weight of lower strength materials have been acceptable than would be desirable in a weight-sensitive item such as a GEM.

Nylon and Dacron in fabric form can be satisfactorily bonded, coated, and used to provide design strength in the final article. The life of both fibers is well established as a function of applied load, so that a firm design study can be undertaken with guaranteed results, for a specified vehicle life. Typical fiber life as a function of load is shown in Figure I-5*, which clearly illustrates why nylon and Dacron are preferred to many other fibers. It should be noted that Fortisan, which

*Goodyear Aircraft Corporation.

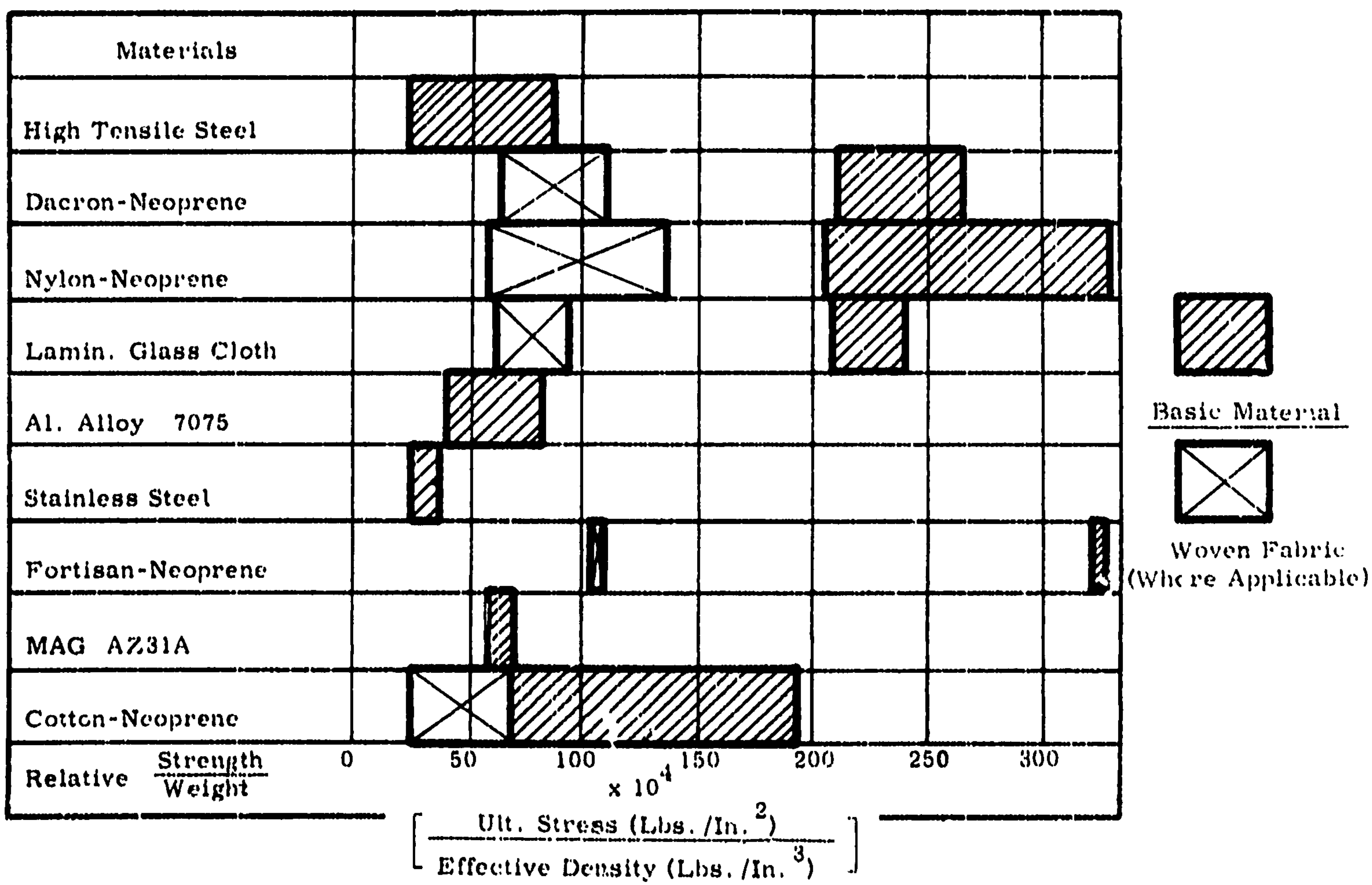


Fig. I-4. Relative Strength/Weight Ratios For Structural Skins

TABLE I-9
TYPICAL CHARACTERISTICS OF A RANGE OF FIBERS

Family	Fiber	Average Specific Gravity	Average Stiffness ksi	Tensile Strength ksi	Elongation at Failure Dry	Elongation at Failure Wet
Rayon	Viscose	1.49	490 to 550	65 to 105	9 to 22	14 to 30
Rayon	Fortisan	1.50	2,240	136	6	6
Acetate	-	1.32	93	22 to 28	23 to 34	30 to 45
Nylon	-	1.14	510 to 720	86 to 134	16 to 28	13 to 32
Acrylic	Zefran	1.19	167	53	33	33
Mod-acrylic	-	1.30	128	50	39	39
Poly-ester	Dacron	1.38	990 to 1120	111 to 138	10 to 14	10 to 14
Olefin	Poly-ethylene	0.95	244 to 610	50 to 90	10 to 20	10 to 20
Saran	-	1.70	175 to 262	44	15 to 25	10 to 25
Glass	-	2.54	10,500	200 to 220	3.0	2.5
Cotton	-	1.54	1,120 to 1,180	44 to 109	3 to 7	-

Reference 10: "Man-Made Fibre Table", "Textile World", McGraw-Hill.

TABLE I-9 (continued)

Family	Fiber	Hysteresis: Elastic Recovery %	Hysteresis: % Strain	Wet Strength % Dry Strength
Rayon	Viscose	70 to 100	2	63 to 72
	Fortisan	100 60	20 40	85
Acetate	-	48 to 65	4	62 to 80
Nylon	-	100	4	87 to 85
Acrylic	Zefran	99 72	2 10	88
Mod-acrylic	-	100 98 95	2 5 10	100
Poly-ester	Dacron	100 90	2 8	100
Olefin	Poly-ethylene	95 80 to 85	5 10	100
Saran	-	95	10	100
Glass	-	100	-	100 (65)
Cotton	-	74 45	2 5	110 to 130

TABLE I-9 (continued)

Family	Fiber	Water Absorbency	Heat	Acids
Rayon	Viscose	11 to 27%	Does not melt, loses strength at 300°F	Attacked
	Fortisan	11 to 20	Does not melt, loses strength at 300°F	Attacked
Acetate	-	6 to 14	Loses strength before 350°F	Attacked
Nylon	-	4 to 9	Loses strength before 350°F	Boiling or concentrated acids will attack
Acrylic	Zefran	2 to 5	Loses strength before 490°F	Resistant
Mod-acrylic	-	.5 to 1.0	Loses strength before 250°F	Resistant
Poly-ester	Dacron	.4 to .8	Loses strength before 480°F	Concentrated acids will attack
Olefin	Poly-ethylene	Negligible	Loses strength before 240°F	Resistant
Saran	-	0 to .1	Loses strength before 240°F	Generally resistant
Glass	-	Negligible	Loses strength by 1,300°F	Resists most
Cotton	-	7 to 27	Loses strength by 250°F	Attacked

TABLE I-9 (continued)

Family	Fiber	Alkalis	Chemicals	Organic Solution	Mildew
Rayon	Viscose	Strength reduction	Attacked	Generally insoluble	Attacked
	Fortisan	Strength reduction	Attacked	Resistant	Attacked
Acetate	-	Saponification	Good resistance	Soluble in some	High resistance
Nylon	-	Inert	Good resistance	Generally insoluble	Not attacked
Acrylic	Zefran	Attacked by strong solutions	Good resistance	Generally insoluble	Not attacked
Mod acrylic	-	Resistant	Good resistance	Soluble in some	Not attacked
Polyester	Dacron	Attacked by boiling strong solutions	Good resistance	Generally insoluble	Not attacked
Olefin	Polyethylene	Resistant	Good resistance	Soluble in some	Not attacked
Saran	-	Generally resistant	Good resistance	Generally insoluble	Not attacked
Glass	-	Resists most	Good resistance	No change	Not attacked
Cotton	-	Undamaged	Reacts	Resistant	Attacked

TABLE I-10

COMPARATIVE PROPERTIES OF VARIOUS TYPES OF RUBBER

Type of Rubber	Tensile Strength 1000 psi	Ultimate Elongation (per cent)	Low Temp. Flexibility	High Temp. Stability	Fuel Resistance	Compression Set	Elec. Resistance	Abrasion Resistance	Ozone & Weather Resistance	Acid Resistance
Natural Rubber	3.5-4.0	500-800	Good	Fair	Poor	Good	Good	Good	Poor	Poor
SBR (GRS)	2.5-3.5	400-600	Fair	Fair	Poor	Good	Good	Good	Poor	Poor
Low Styrene SBR	1.5-2.5	400-600	Very good	Fair	Poor	Good	Good	Good	Poor	Poor
Cis-Polyisoprene	3.5-4.0	600-900	Poor	Fair	Poor	Good	Good	Good	Poor	Poor
Nitrile	3.0-4.0	300-600	Poor	Good	Very good	Good	Good	Good	Poor	Fair
Neoprene	2.5-4.0	300-600	Fair	Good	Good	Good	Very good	Very good	Very good	Good
Butyl	3.5-3.5	400-700	Fair	Good	Good	Poor	Very good	Good	Very good	Very good
Thiokol	0.8-1.5	300-500	Good	Good	Excellent	Fair	Good	Fair	Fair	Good
Silicone	0.8-1.8	300-800	Excellent	Excellent	Fair	Fair	Very good	Fair	Good	Fair
Hypalon	2.5-3.5	400-600	Fair	Good	Fair	Good	Good	Very good	Very good	Good
Polyurethane	4.0-7.0	500-800	Poor	Fair	Good	Poor	Very good	Excellent	Good	Good
Acrylates	2.0-3.0	300-700	Fair	Very good	Good	Poor	Good	Good	Good	Fair
Fluorocarbons	2.5-3.5	300-800	Poor	Excellent	Very good	Fair	Very good	Good	Excellent	Excellent

Reference II: Braunston, L.K., and Montermoss, J.C., "Rubber in Electronics", Military Electronics, 1956.

is a very high-strength fabric, has very poor life under load which renders it unsuitable for vehicular structures.

1.4.3 Coatings and Bondings

The choice of coating and bonding material is related to the environmental conditions, and must be compatible with the base fabric material. Such questions as resistance to mechanical working (involved in packaging and storing), creep characteristics, temperature suitability, resistance to fuels, ozone, fungus, ease of application, etc., must all be answered satisfactorily before a material can be finally accepted.

Materials meeting the above fall into the family pattern of rubber and have the general characteristics shown in Table I-10 from reference 11; of these, only a few are generally suitable for use in inflatable structures. Present practice has indicated a strong preference for neoprene as a bond material and hypalon for abrasion and ozone resistance under ordinary handling conditions. Other materials such as viton and adiprene are under investigation for increased abrasion resistance.

The following information from reference 11 shows that rubber is ideally suited for the purpose of coating other materials. It is elastic, flexible, airtight, adhesive, and a good nonconductor of electricity. Different types of rubber have varying degrees of electrical resistivity. Rubber also absorbs shock and vibration. Because of the unique combination of these properties, rubber is constantly finding new areas of use

While each type of rubber has its own outstanding characteristics that distinguish it from the others, proper compounding, processing, and vulcanization are necessary to bring out the optimum qualities desired (e.g., high tensile strength and elongation, low compression and permanent set, good abrasion resistance).

1.4.3.1 Natural Rubber

The outstanding characteristics of natural rubber are its high gum tensile strength, low heat build-up or hysteresis, and low compression and permanent set. Gum stock natural rubber has a tensile

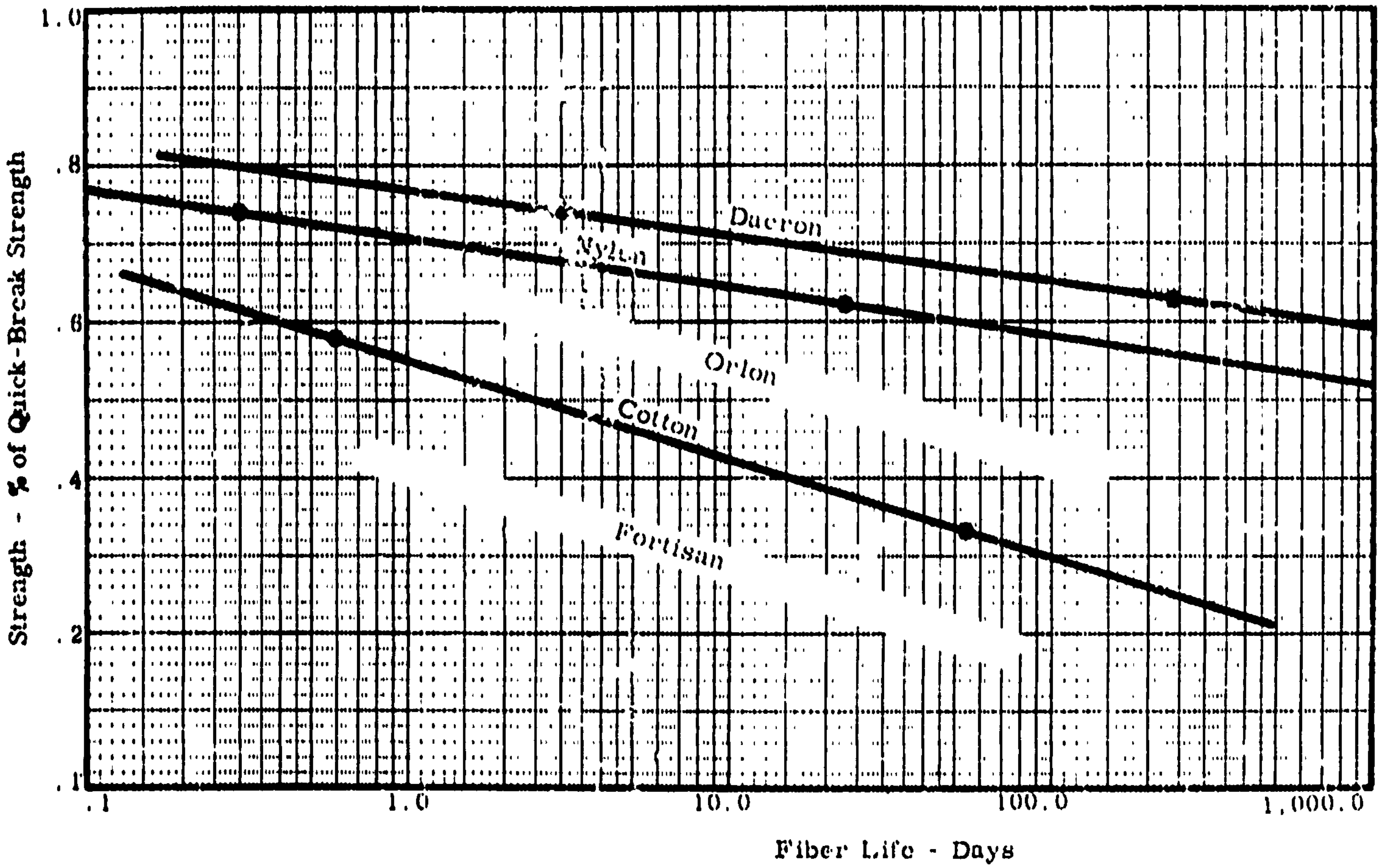


Fig I -5. Typical Creep - Rupture Characteristics of Materials

Compiled from Goodyear Aircraft Corporation data.

strength of about 4,000 pounds per square inch and elongation at break of 800 per cent. Natural rubber compounds are not resistant to fuel and oil. These compounds swell and lose tensile strength when in contact with gasoline, iso-octane, benzene, toluene, and other hydrocarbons.

As an insulating material, natural rubber compounds have good electrical resistance. Being hygroscopic, natural rubber gradually loses its electrical efficiency under high humidity over an extended period. Rubber can be made to adhere to metals. The most effective adhesion to ferrous metals can be obtained by brass plating the adhesion area.

1.4.3.2 Synthetic Rubbers

1.4.3.2.1 Styrene Butadiene Rubber (SBR)

Styrene Butadiene Rubber (SBR) is made by the copolymerization of butadiene and styrene monomers. It was formerly called GR-S (also known as Buna-S) meaning "government rubber-styrene"; however, since the government-owned synthetic rubber plants were sold to industry last year, the term SBR has been more or less unofficially adopted. Like natural rubber, SBR is not resistant to oils and hydrocarbon fuels. SBR is the workhorse of the rubber industry and is most commonly substituted for natural rubber. Enough tonnage of this rubber was produced by this country during World War II to save us from the critical rubber situation. Unlike natural rubber, SBR requires reinforcing fillers such as carbon blacks, clay, and other pigments for the needed tensile strength. Gum stocks of SBR have tensile strength of about 300 pounds per square inch. On the other hand, tensile strengths of 3,500 pounds per square inch are obtained in stocks loaded with carbon blacks.

There are several types of SBR. One type, known as Cold Rubber, which was polymerized at 41°F instead of 122°F for the common types, possesses better abrasion resistance and tensile strength than the regular SBR, although it is not so flexible at extremely low temperatures. It is the one most commonly used for passenger tires. Another type, known as the Low Styrene Rubbers, is flexible and serviceable at temperatures down to -65°F. Highly plasticized or oil-masterbatched SBR are also used for low temperature service.

1.4.3.2.2 Neoprene

Neoprene is the generic name for chloroprene-base synthetic rubber. Some of the properties which make Neoprene valuable are its (1) resistance to oils, fuels, and certain solvents, (2) resistance to heat and oxygen, (3) resistance to ozone and weathering, and (4) flame retardance.

Its ozone and weathering resistance qualities have made it the standard in many electronic applications. It can be compounded and processed as readily as natural rubber and in a similar fashion. The end items made from this chlorinated rubber possess the good qualities of natural rubber plus its own desirable properties previously mentioned. However, its flexibility at sub-zero temperatures is not as good as that of natural rubber. Neoprene can be made to adhere exceptionally well to all metals.

1.4.3.2.3 Nitrile

Nitrile, like Neoprene, is a specialty rubber. Resistance to fuels and oils is the most important property of nitrile rubber, and is the main reason for its extensive use, even though it costs substantially more. Nitrile rubber, originally known as Buna N, is a copolymer of butadiene and acrylonitrile. The proportion of acrylonitrile in the polymer determines the degree of fuel resistance. Generally, nitrile rubber is more oil-resistant than neoprene, but it does not have a good ozone and weathering resistance. It has better low temperature flexibility. Nitrile rubber is processed and vulcanized in a fashion similarly to natural rubber and SBR.

1.4.3.2.4 Butyl Rubber

The most striking property of butyl rubber is its high impermeability to gases, including air. It has been shown that butyl rubber is about 7 to 10 times as impermeable to air as natural rubber. Butyl rubber has a very good resistance to deterioration by ozone and heat. It possesses excellent tear resistance. It is also resistant to acids, alkalies, and aromatic hydrocarbons. Butyl rubber has very good electrical properties. The largest outlet for butyl rubber in non-transport items is in the production of insulated wires and power cables. Good electrical properties are retained even after immersion in boiling water for varying periods.

Butyl rubber is made by copolymerizing isobutylene with a small amount of isoprene (1.5 to 4.5 per cent). Several grades of butyl rubber representing varying degrees of unsaturation (depending on the amount of isoprene) and different ranges of molecular weights are commercially available. Butyl rubber vulcanizes more slowly than natural rubber due to its limited unsaturation.

1.4.3.2.5 Polysulfide Rubber

The polysulfide rubber, commercially called Thiokol, is known for its resistance to fuels, oils, and organic solvents. It also has the best low temperature flexibility of all commercially available fuel-resistant rubbers. It may be mentioned that some unpublished data show Thiokol to be as good as butyl rubber with regard to air impermeability; however, Thiokol has much lower tensile strength and abrasion resistance than natural rubber or SBR.

Thiokol is prepared by a condensation reaction of aliphatic dihalide and sodium polysulfide. Various types of polysulfide rubber, both in liquid and in solid forms, are commercially available.

1.4.3.2.6 Silicone Rubber

Silicone is unique in many ways. No other rubber surpasses silicone rubber in the range of temperature over which the useful rubbery property is retained. Silicones are flexible down to -80°F and stable at temperatures as high as 500°F. In addition, it is resistant to ozone and weathering, and to lubricating oils. It possesses good electrical properties. Several types of silicone rubber, both in compounded and in gum stocks, are commercially available. Each type possesses a particular set of properties.

Silicone rubber is compounded and vulcanized differently from natural rubber or SBR. A short press cure followed by a long oven cure is required. The curing agents are usually organic peroxides. In the finished products where close tolerance in dimensions, particularly in electronic applications, is important, care must be taken to consider the shrinkage of silicone rubber during vulcanization.

1.4.3.2.7 Polyurethane Rubber

Polyurethane rubber, known as Vulkollanes, was originally made in Germany by von Bayer and his co-workers. Various types and grades of polyurethane rubber are presently being produced in the United States. The outstanding properties of this rubber are: (1) superior tensile strength up to 7,000 pounds per square inch; (2) high abrasion resistance (several times that of natural rubber); and (3) ozone and weathering resistance. It is also flame-retardant and fuel resistant.

Although one of the newest commercial rubbers on the market, it is finding uses in the electronics field, since it can be used in compression molding, liquid casting, or injection molding (transfer). Intricate cores and shapes can be molded. Polyurethane rubber has good electrical properties with low loss factor at high frequencies.

1.4.3.2.8 Hypalon

Hypalon is a chlorinate sulfonated polyethylene rubber. Its outstanding properties are stability at high temperatures and resistance to chemicals, oxidizing agents, and to ozone. It is used in hose, gaskets, diaphragms, O-rings, tank linings, etc., where resistance to oxidizing chemicals is desired. It also has good resistance to hydrocarbon fuels.

Hypalon processes well in ordinary rubber equipment. It is not vulcanized by sulfur like natural rubber. Metallic oxides are used as cross-linking agents. Magnesia has some advantages over litharge as a curing agent for Hypalon.

1.4.3.2.9 Fluorinated Rubbers

Fluorinated rubbers are recent developments. Two fluorinated rubbers on the market today are known as KEL-F Elastomers and Fluoro-Rubber 1F4, recognized for their resistance to strong oxidizing chemicals, acids, hydrocarbon fuels, lubricating oils, hydrazine, etc. They are flame-retardant and thermally stable at high temperatures. They also possess good electrical properties and excellent ozone resistance. However, they become stiff and brittle at subzero temperatures.

1.4.3.2.10 Polyacrylate Elastomers

Polyacrylate elastomers such as Acrylon EA-5, Acrylon BA-12, and Hycar 4021, are on the market today. They are copolymers of acrylic acid esters. They are oil and fuel resistant. Polyacrylate rubbers are of special interest because of their ability to withstand the deteriorating effects of sustained high temperature (up to 400°F) in air and in many nonaqueous immersion media. They also possess good ozone resistance and electrical properties. They become stiff, however, at sub-zero temperatures.

1.4.3.2.11 Synthetic Cis-Polyisoprene

Synthetic cis-polyisoprene on most counts is a duplicate of natural rubber in every respect, including the low heat build-up characteristics. The significant application of this new rubber will be as a replacement to natural rubber in big truck and aircraft tires in times when natural rubber is not available.

1.4.4 Fabrics

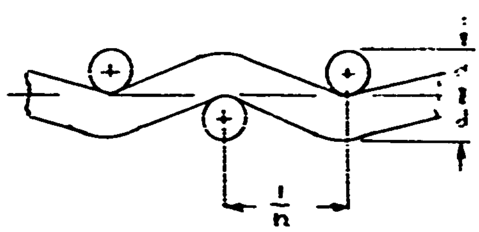
1.4.4.1 Bonded Plied Fibers

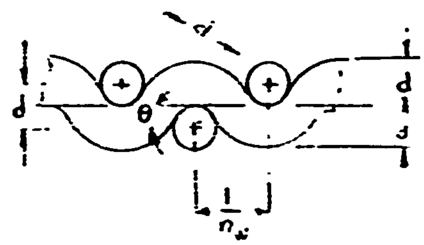
The fibers may be used directly as fibers laid up in the required direction to take the load and bonded together. By this means practically the full tensile strength of the individual fibers can be realized, with poor tensile strength at right angles to the fibers (directly due to the bond strength). This process is particularly useful where unidirectional stresses are encountered; a more efficient load-carrying section can be developed with lower bond weights and coating weights, but with low strength normal to the fibers.

1.4.4.2 Woven Fibers

When woven as a cloth, the fibers are much more readily handled and utilized in manufacturing a product; however, the full strength of the fibers is not realized, except perhaps when a load is applied in the direction of only one set of fibers. Several weaves are possible; however, the simple weave is most widely used. Its performance is easier to predict, and it is much cheaper to make. When loads of roughly equal magnitude are applied in the warp and fill direction the fiber stresses are higher than the applied loads, due to fiber crimping. A reasonable assessment can be made of the fabric strength as a function of fiber strength, by considering fabric geometry.

Consider a simple weave with the same fibers for warp and fill, woven with even tension on both warp and fill so that the two directions are indistinguishable from each other. Assume that the fabric is so made that the fibers are packed together in both directions, as far as they can go, yet still result in a symmetrical warp and fill as shown in the following sketches:





For the closed symmetrical fabric, equally loaded in both directions

$$\left|\frac{1}{n_w}\right| = 1.732\, d \qquad \text{since } \theta = \sin^{-1}\frac{d}{2d} = 30^\circ$$

If T_o is maximum fiber tension, the longitudinal load supported by each fiber is

$$T_o \cos 30^\circ = \frac{\sqrt{3}}{2}\, T_o .$$

Load per inch of fabric = load per fiber x fibers/inch.

$$= \frac{\sqrt{3}}{2} T_o \times \frac{1}{\sqrt{3}d} = \frac{T_o}{2d}.$$

Fiber load = Fiber stress, f x cross-sectional area

therefore $T_o = f \times \frac{\pi}{4} d^2.$

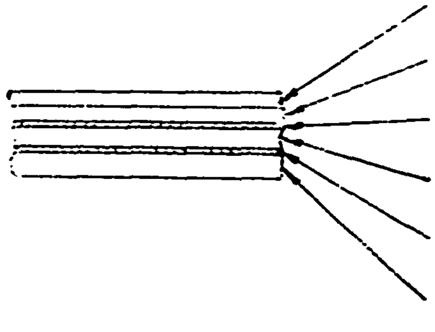
and Load per inch $= \frac{\pi}{8} f d.$

Hence, for a fiber with an ultimate strength of 150,000 pounds per square inch, for example, woven into a fiber with equal warp and fill threads of .015 inches diameter and equally loaded along all four sides, the load per inch at breaking will be approximately $\frac{\pi}{8} \times 150{,}000 \times .015$ or 884 pounds per inch. Such a fiber, if made from Dacron, would be of approximately 1,400 denier. This example agrees reasonably well with a particular "hot stretch" Dacron of 1,100 denier, which has a base fabric strength of approximately 800 pounds per inch.

When woven, the fabric can be utilized either for exterior cloth with very low permeability, or for interior structural material. The internal structure can be made of the cloth as woven, whereas the exterior cloth has to be treated in order to provide good air retention capabilities and good resistance to the environmental conditions referred to in the previous section.

1.4.4.3 Exterior Cloth Construction

The following sketch is typical of cloth construction designed to provide good air retention and good environmental resistance.



The base fabric may consist of one or more plies bonded to the straight weave, or several layers of straight weave, depending on the complexity of the loading requirements.

Other configurations are quite possible--the way in which the fabric skin is made up is dependent on the loading and environmental requirements and the method of construction--but the arrangement outlined above is typical of most areas that are not subjected to highly complex loadings.

1.4.4.3.1 Typical Practical Fabrics

A typical development of a practical cloth having a construction very similar to that described has resulted in the following detailed weight-breakdown, enabling strength and weight relationships to be outlined for use in preliminary design work and structural comparisons.

Base Fabric Strength (lbs/inch)	325	800
Component Weights (ozs/sq.yd.)		
Abrasion Coat - Hypalon	4	4
Bond - Neoprene	1	1
Air Retention Fabric - Nylon or Dacron	1	1
Bond - Neoprene	5	8
Base Fabric - Nylon or Dacron	7	12
Coating - Neoprene	4	5
Total Weight ozs/sq.yd.	22	30

Utilizing these data, the characteristics of typical practical fabrics are as outlined in Fig. I-6. Strengths higher than 600 pounds per inch have been achieved by bonding together two or more layers of base fabrics. with a corresponding increase in weight. From the tabulated data the approximate weight at any strength of practical outer fabric can be estimated. The lower line on the figure shows the plain base fabric weight, in which higher strengths are obtained by adding more fabrics. Bonding may be undesirable for internal structure where permeability is required to permit inflation air flow. When bonding is necessary, the weight may be increased by approximately six ounces per square yard per additional fabric layer. It is considered that these data are typical of fabrics required for heavy duty outdoor applications such as in the construction of parts for a GEM structure.

1.4.4.3.2 Environmental Test Data on Fabrics

A limited amount of environmental test data on fabrics and coated fabrics has been published. The following data have been compiled from reference 12 and apply directly to GEM inflatable structures.

1.4.4.3.2.1 Salt Water Environmental Test Program for Fabrics

A two-phase program, established with the Materials Division of the Bureau of Ships, was adopted on 5 February 1960 to explore the effects of sea water on fabric specimens continuously submerged in the Miami area. Tests completed to date demonstrate that sea water operations of GEMs will not materially reduce fabric life.

Phase I consists of an evaluation of the basic available cloth and elastomer materials (Dacron, Nylon, Rayon, Cotton, Neoprene, Chemiqum, Chemiqum-Vinyl, Hypalon and Natural Rubber).

Phase II consists of a limited literature survey to determine what work has been done with regard to inhibition of fouling organisms on structures. Based on this survey and the Phase I test results, a limited number of test specimens using improved materials containing and/or coated with anti-fouling materials will be evaluated.

Eleven specimens of each of eleven Phase I materials were delivered to the Miami Marine Research Test Station, Miami Beach, Florida, on 1 March 1960. One of each type of the eleven materials will be

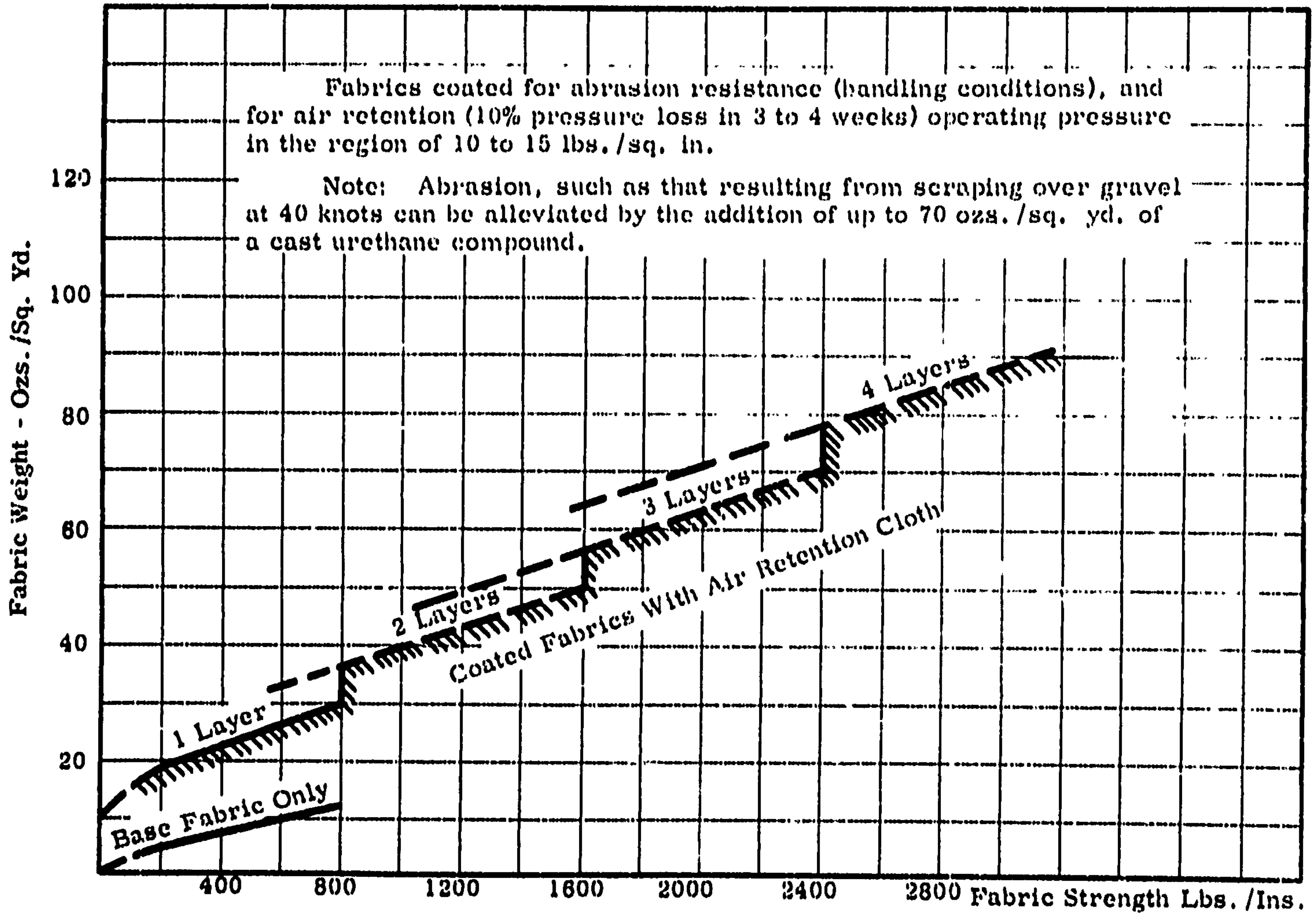


Fig I -6. Approximate Weight/Strength For Practical Fabrics for Inflated Structures marked for weight, tensile (strip and grab) elongation, tear and burst tests*.

BuShips' monthly reports on marine fouling of these specimens not only cover the fouling of the specimens and glass reference surfaces, but also provide pertinent atmospheric data. In addition to Phase I, a program was initiated to determine the effects of the ocean environment on the seam strength of various coated fabric seam constructions which might be used in the fabrication of underwater fabric devices. To accomplish this, ten seam samples each of four different constructions were prepared for exposure purposes. These specimens were sent to the Miami Marine Research and Test Station on 2 May 1960. These specimens will be given seam strength tests. The results of the tests to date on these specimens are published by BuShips along with the fouling reports on these particular specimens.

As part of the Phase II effort, specimens have been prepared of an elastomer which has been impregnated with various concentrations of either a Vanderbilt anti-fouling compound or a chloradane compound. These specimens were shipped to the Miami Marine Research and Test Station on 22 August 1960 and will be exposed, inspected, and shipped back for test on the same schedule as the Phase I specimens.

Two other special specimens have also been prepared by using an elastomer sheet material that could be metal plated. One of these specimens was plated with copper and the other with nickel. These specimens were shipped to the Miami Marine Research and Test Station on 22 August 1960. They were exposed and observed for fouling like the Phase I specimens and returned for observation and physical tests after several months exposure.

The initial program was started by submerging samples of cotton, rayon, nylon, and selected materials from this group coated with various elastomers into an ocean environment to determine the effects of such an environment on the anti-fouling and physical properties of the materials. Additional specimens of elastomers compounded with anti-fouling agents have also been subsequently submerged at the Miami Marine Research and Test Station.

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*For explanation of tests, see Federal Textile Specification 5512, CC-T-191b.

At the end of three months' immersion, the cotton cloth completely deteriorated. A similar situation occurred with the rayon cloth at the end of four months' immersion.

On the basis of the test results obtained thus far, of the uncoated cloths, nylon 293N is exhibiting the best tensile strength and elongation properties, while the Dacron cloth is running 10-15% below the nylon.

Examination of the data collected over an eight-month period reveals that no single coated nylon specimen in the group covering coated fabrics using mutually similar cloth (nylon) exhibits superior physical properties in all categories when compared to the other coated specimens in the group.

On the basis of strip tensiles, the chemigum/vinyl-nylon fabric shows the most favorable results thus far. The hypalon-nylon, neoprene-nylon, and chemigum-nylon fabrics are closely grouped in second place while the crude rubber-nylon fabric falls into third place.

In the grab category, the tensile values fall into two distinct groups. The chemigum/vinyl-nylon and chemigum-nylon fabrics show the best results, while the neoprene-nylon, hypalon-nylon and crude rubber-nylon fabrics exhibit lower values and fall into the second group.

The elongation results reveal about a 12% spread thus far with no one material clearly showing superior characteristics. A tentative ranking based on elongation would put the materials in the following (minimum or maximum) order:

(1) Crude rubber - nylon fabric

(2) Hypalon - nylon fabric

(3) Chemigum - nylon fabric

(4) Neoprene - nylon fabric

(5) Chemigum/vinyl - nylon fabric

From a tear standpoint, the material values once more fall into three distinct groups with about a 20% spread. In order of their superiority, they are hypalon-nylon fabric in group 1; the crude rubber-nylon, chemigum/vinyl-nylon, and neoprene-nylon fabrics in group 2; and the chemigum-nylon fabric in group 3.

The evidence collected in the hydrostatic (Mullens burst test) category to date reveals that, for all practical purposes, the materials are equal.

One interesting characteristic that all five of these materials have exhibited, at various times thus far in the exposure cycle, is the tendency to show increases in test values over the original unexposed control specimens in all categories. The appearance of this phenomenon is erratic both in time and magnitude. Consequently, it is difficult to conclude at this time whether the cause is due to variation in the materials, aging, accuracy of tests or some yet to be determined factor. Future observations of test data will include an effort to pinpoint the origin of this phenomenon and determine the extent of its importance.

In the coated fab -s with mutually similar elastomer (neoprene) category, neoprene-cotton and neoprene-Dacron fabrics, are those materials being considered. Examination of the test data collected on the neoprene-cotton fabric, over an eight-month period, reveals that, for all practical purposes, this material had over-extended its useful life in an ocean environment. This is illustrated in the following summary of test results:

Strip Tensile	Warp - 33 lbs/in. *(370 lbs/in)	Fill = 4.5 lbs/in (339 lbs/in)
Grab Tensile	Warp - 0 (268 lbs/in)	Fill = 0 (244 lbs/in)
Elongation	Warp = 33% (19%)	Fill = 13.4% (15.5%)
Tear	Warp - 1.2 lbs. (9 lbs)	Fill = 1.5 lbs. (9 lbs)
Hydrostatic	0 (1,000 PSI)	

*Figures in parentheses indicate original control specimen values.

The neoprene-Dacron fabric when compared with neoprene-nylon fabric exhibits equal or better characteristics in all categories. When compared with other materials, the neoprene-Dacron fabric appears to possess the best over-all physical characteristics at this stage of the exposure cycle.

At the end of six months exposure, the four types of seams fabricated from neoprene-nylon fabric show the following over-all percentage drops from original strengths:

Cemented[1] - Air Cure	25.6%
Cemented[1] - Open Steam Cure	32.6%
Taped[2] - Air Cure	19.6%
Taped[2] - Open Steam Cure	27.1%

[1] Plain lapped cemented joint.

[2] Lapped cemented joint with tapes applied along edges of joint.

The spread in the results thus far is 13% but not conclusive enough to indicate any marked advantage being held by a particular seam type.

Initial test specimens of elastomer compounds for coated fabrics containing anti-fouling compounds are in the process of being tested.

On the basis of the test data accumulated through eight months from cloth and fabric specimens submerged in an ocean environment, the following conclusions are drawn:

Cotton and rayon cloths deteriorate after 3-4 months exposure to sea water; therefore, these materials would be satisfactory for short-term applications only.

Cotton and rayon cloths coated with elastomers extend the lifetime of these materials to about eight months. This would be satisfactory for short-term applications.

Nylon and Dacron fabrics exhibit the best physical characteristics thus far.

At the end of ten months' exposure, no single elastomer coating shows an over-all advantage in physical characteristics over the other coatings in the group.

The coated Dacron fabric compares quite favorably with the coated nylon fabrics with respect to all physical characteristics.

At the end of eight exposures, the neoprene-nylon seam specimens do not indicate any decided advantages among the four types being tested.

Figures I-7, I-8, I-9 contain summary information obtained to date on the various specimens.

1.4.4.3.3 Strength - Temperature Properties of Materials

The strength of fabric materials and the coating compounds are affected by the ambient temperature at which they are used; neoprene-coated nylon and Dacron, for instance, show a pronounced strength reduction as the temperatures increase to 300° to 400°F, respectively. In the case of fabric made of metal threads and coated with silicone compounds, relatively high temperatures exceeding 1,100°F will reduce the strength of the material only about 50%.

In the case of low ambient temperatures the synthetic fabric shows improved structural qualities. At -60°F the tensile strength increases 25% and the elongation decreases from 35 to 25%. At temperatures from -60° to -100°F, little data are available but no damage is expected unless the material is subjected to severe folding, packaging, or buckling. In general, however, at temperatures below -40°F, the coating compound becomes very brittle, hence limiting fabric utility at low temperatures.

Strength-temperature properties of fabrics are shown in Fig. I-10.

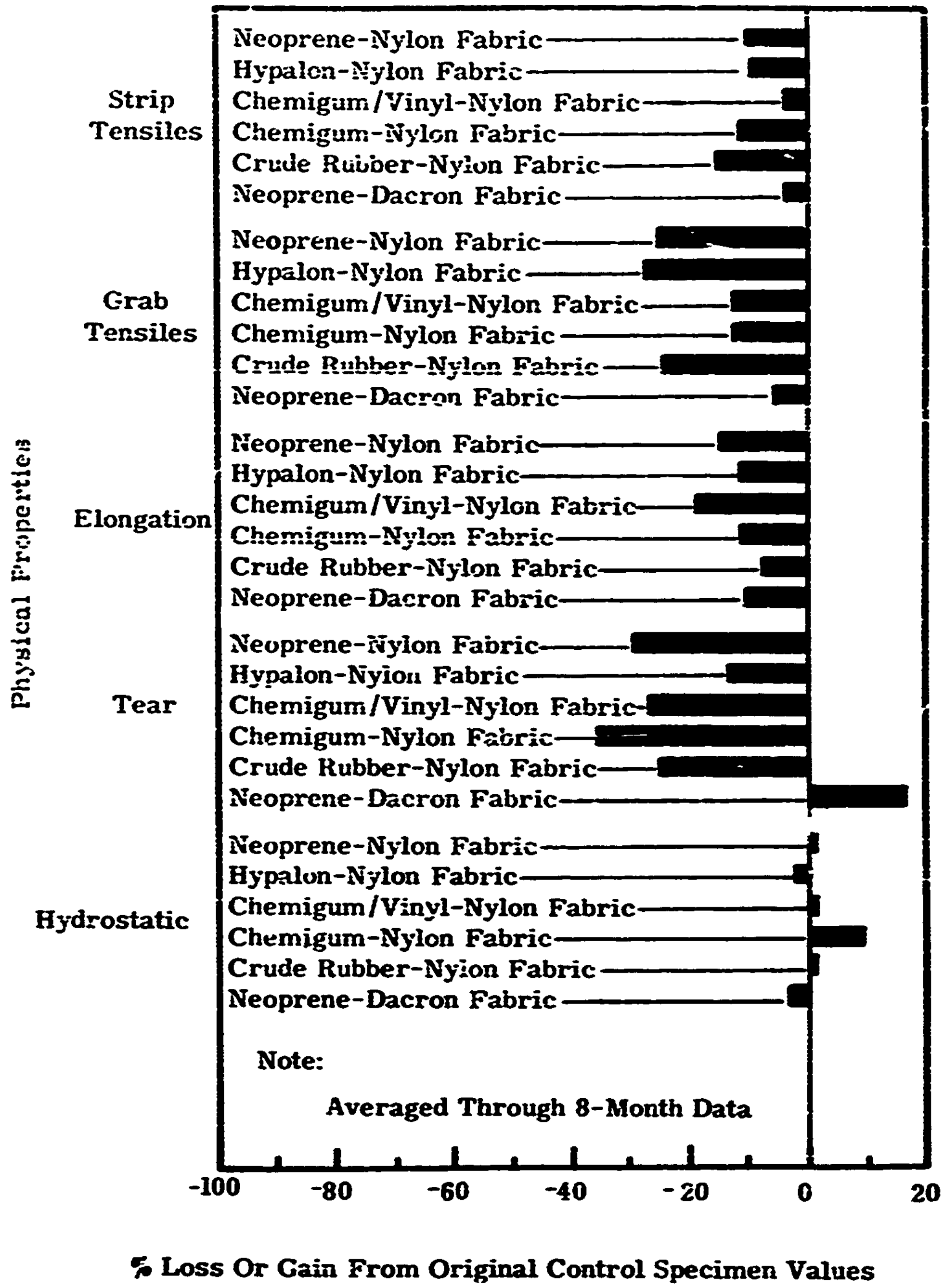
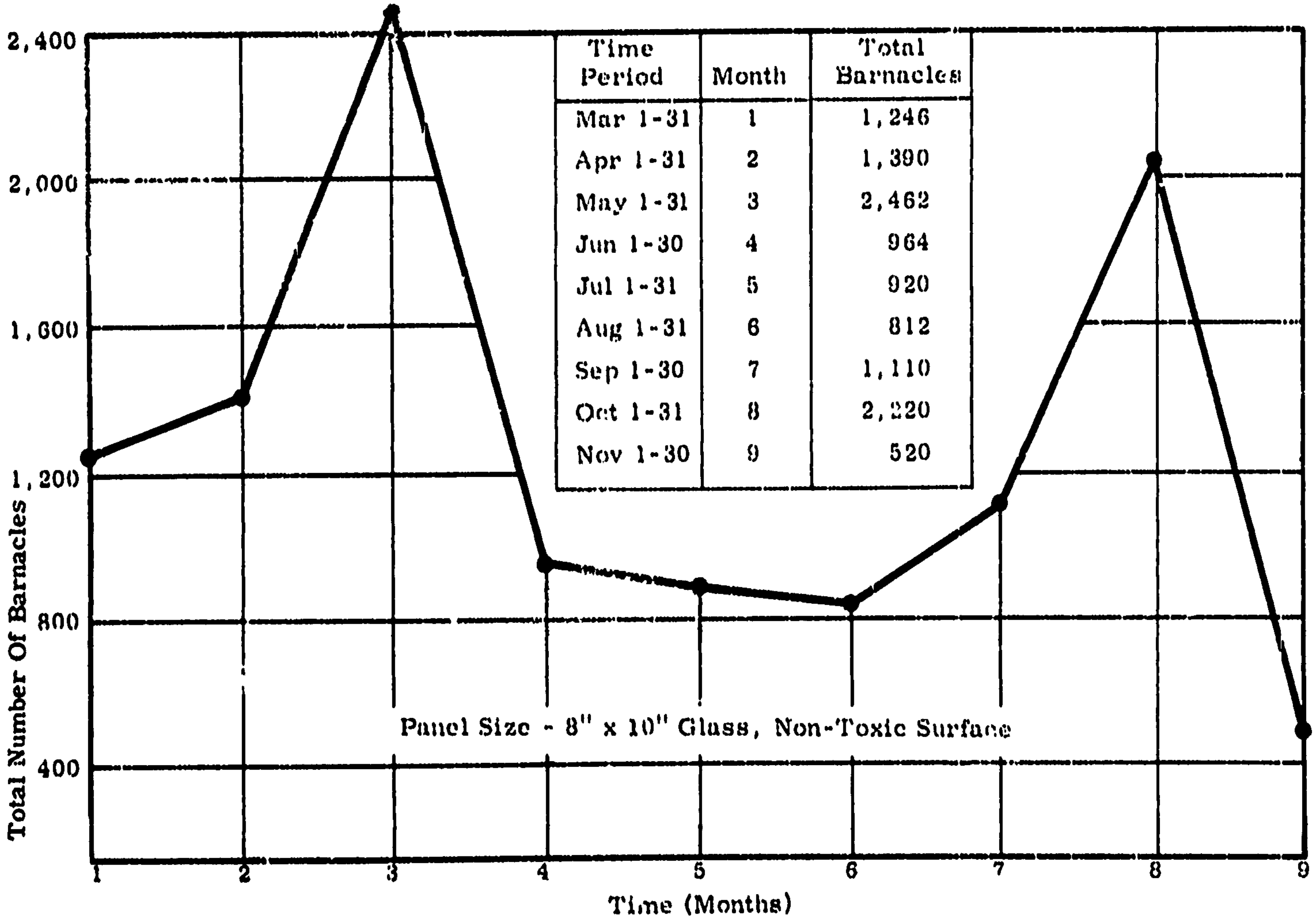
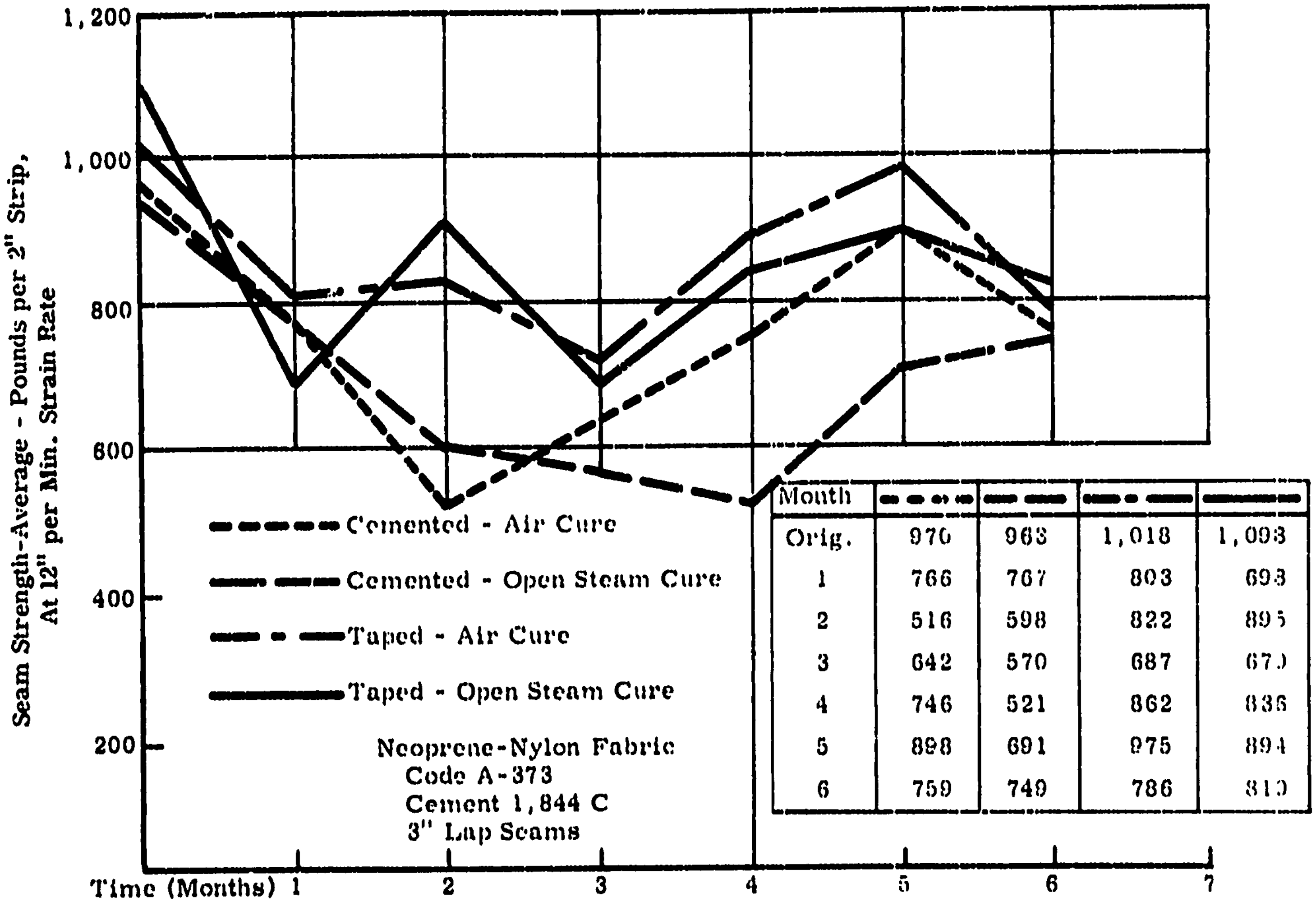


Fig. 1-7. Variation In Physical Properties Of Fabrics Under Environmental Test



Time Period	Month	Total Barnacles
Mar 1-31	1	1,246
Apr 1-31	2	1,390
May 1-31	3	2,462
Jun 1-30	4	964
Jul 1-31	5	920
Aug 1-31	6	812
Sep 1-30	7	1,110
Oct 1-31	8	2,220
Nov 1-30	9	520

Fig. 1-8. Organic Growth During Environmental Test



Month	Cemented - Air Cure	Cemented - Open Steam Cure	Taped - Air Cure	Taped - Open Steam Cure
Orig.	970	963	1,018	1,093
1	766	767	803	693
2	516	598	822	895
3	642	570	687	670
4	746	521	862	836
5	898	691	975	894
6	759	749	786	810

Fig. I -9. Variation In Seam Strength During Environmental Test

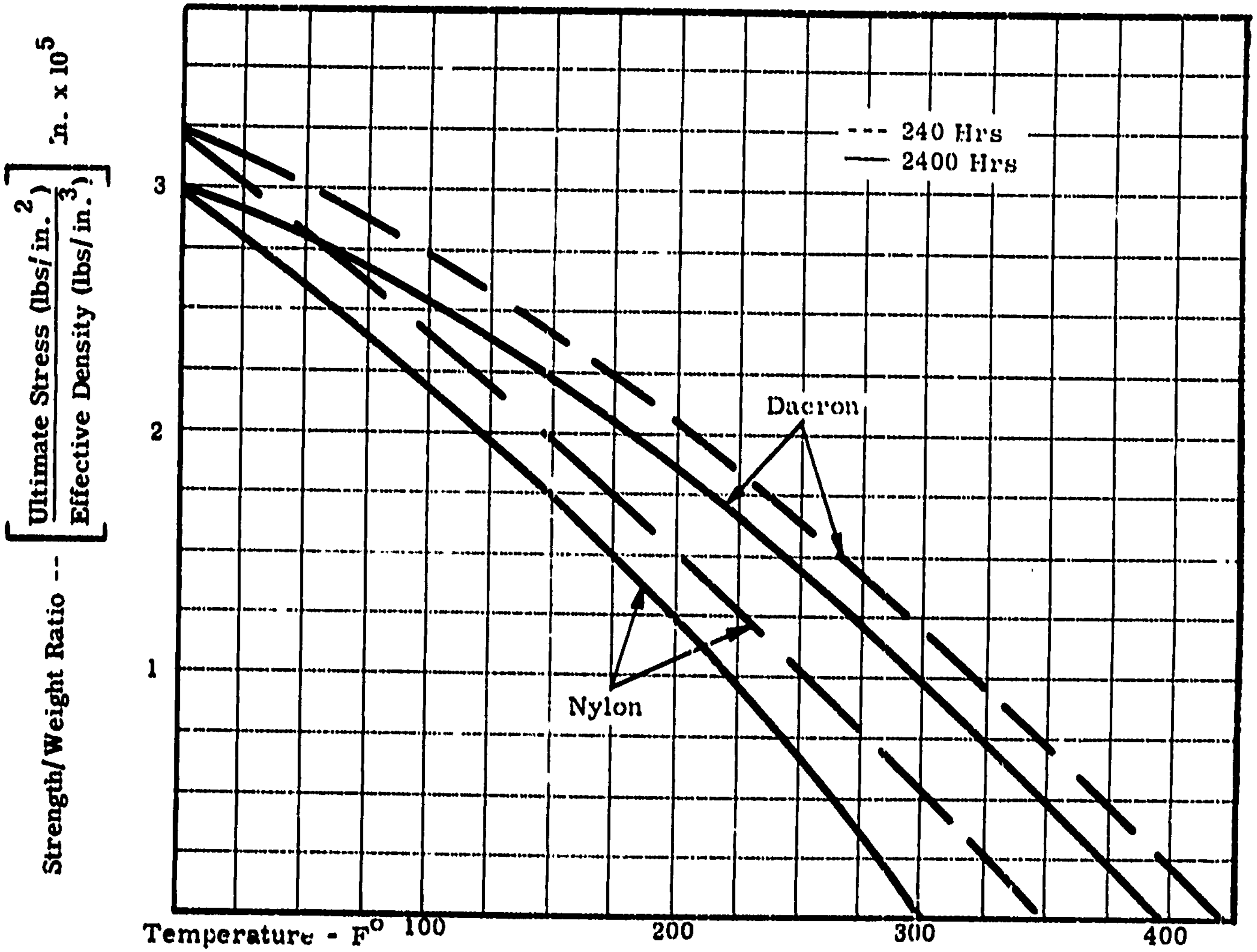


Fig. I-10. Strength-Temperature Properties of Dacron And Nylon

1.5 INFLATION SYSTEMS REVIEW

1.5.1 Inflation System

The inflation system is dependent on the following primary parameters:

1. Inflation pressure
2. Inflation volume
3. Rate of inflation

and the following secondary parameters:

4. Number of compartments
5. Inflation sequence
6. Maximum damage capability (pressure maintained vs. leakage area)
7. Regular "topping-up" requirements.

1.5.2 Components of the System

Required typical components of the inflation system are:

1. Compressed air source
2. Distribution system
3. Sequencing valves
4. Shut-off valves
5. Pressure relief valves
6. Dump valves

1.5.3 Inflation Pressure

Inflation pressures will vary in accordance with the design requirements. Pressures from 10 to 20 pounds per square inch for primary structures and up to 40 pounds per square inch for secondary structures are considered maximum practical values from a safety standpoint, although inflatable structures have been subjected to pressures as high as 150 pounds per square inch.

1.5.4 Inflation Volume

This parameter is a fixed figure dictated by the size and design of the craft.

1.5.5 Rate of Inflation

Rate of inflation will be dictated by the desired assembly time of the machine and must be controlled at proper rates consistent with handling vehicle components.

1.5.6 Number of Compartments

Design considerations such as inflation pressures, and buoyancy compartmentation to maintain the floatability of the machine in case of a badly damaged structure, will dictate the number, positions and size of compartments.

1.5.7 Inflation Sequence

This will be dependent upon the different pressures used and the appropriate erection sequence of the different parts of the structure.

1.5.8 Maximum Damage Capability

It is expected that during the normal operating life of the vehicle, the inflatable structure will be subject to different kinds of damage dependent on the operating environment or due to battle damage. Since it is vital to maintain the inflation pressure for structural integrity, the compressed air system must be designed with enough capacity to cope with the maximum leakage anticipated. A system designed to take care of this kind of damage will have more than enough capacity to take care of the normal leakages associated with this kind of structure.

In order to minimize the weight penalty of a system designed to meet the requirements of extensive damage, it is suggested that internal sealants or automatic foam injection systems be studied to stabilize the damaged part of the structure.

1.5.9 Compressed Air Sources

There are three periods during which an inflatable structure requires a source of compressed air. They are:

1. Inflation of the structure during assembly.
2. Inflation of the structure during actual operations for compensating the losses due to leakages or in the event of damage.
3. In the event that there is an "increased strength" requirement, for short periods of operation.

The prime source of compressed air can be divided in two categories:

1. Ground support equipment.
2. Craft-borne equipment.

The first category does not require any special treatment since by definition, its capacity and delivery pressure characteristics will be adequate for the inflation requirements.

The second category can be of several different types such as:

1. Independent compressor units.
2. Compressed air storage cylinders.
3. Air compressors driven from the auxiliary power outputs of the main engines.
4. Compressed bleed air from the main power turbines.
5. Ram air intakes at high speed.

The first two may be called independent since they do not require the operation of the main engines. The independent air compressor can be used for inflation, operational leakages, and pressure changes, since the total volume pumped is unlimited and only dependent on time of operation.

Figure I-11 presents the power requirements versus C.F.M. and inflation pressure. Table I-11, covers a range of representative air compressors, showing their weights and prices.

1.5.10 Compressed Air Storage Cylinders

These are not considered a practical solution either for inflation or emergency, due to the limited capacity of air and the big penalty in weight to be paid, plus the need of special charging facilities.

1.5.11 Bleed Air from Main Turbine

This system, although it provides an appropriate capacity for inflation purposes, requires special cooling installation in order to bring the temperature from the levels depicted in Table I-12 to a temperature appreciably less than 180°F.

From Figure I-12, for a capacity of 10 C.F.M. at 150 pounds per square inch, or approximately 100 C.F.M. at 15 pounds per square inch the percentage of power lost is 1 per cent or 26.9 SHP. This figure does not compare favorably with the power of 5 to 10 HP required by an independent compressor, either in terms of power, or in weight of fuel necessary to compress the same amount of air.

1.5.12 Ram Air Intakes

A ram air intake system is not considered practical since the speed necessary to compress air to 10 pounds per square inch is far beyond the range of anticipated operating speeds of the GEM.

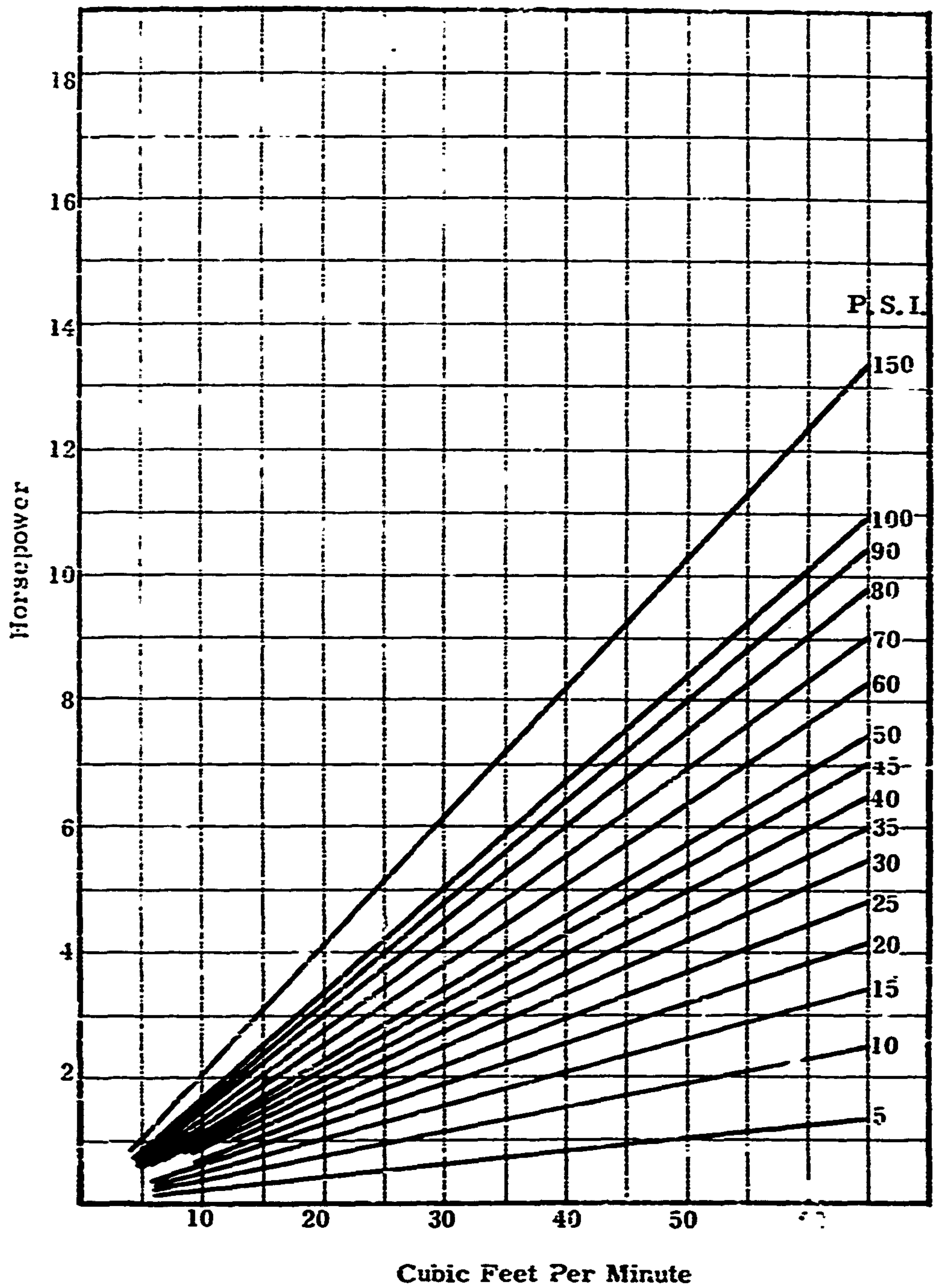


Fig. I -11. Horsepower Vs Capacity and Pressure

TABLE I-11
CHARACTERISTICS OF AVAILABLE COMPRESSOR UNITS

Brand	Model	C.F.M.	P.S.I.	HP	Weight lb	Price $	Remarks
Bell & Gosset	SYCO4-1	6.51	35	3/4	51	184.75	Electric driven
	SYCO8-1	5.4	75	3/4	51	197.75	Electric driven
		32	125	5	500	600.00	No Tank
		48	125	7.5	700	1000.00	No Tank
Gast	1550	10.5	10	3/4	29	102.00	Without motors
	3040	30.0	10	2	68	168.00	Without motors
	1065	8.3	25	1	33	111.00	Without motors
	2565	21.0	15	2	51	147.00	Without motors
	4565	45	15	5	92	256.00	Without motors
Worthington	5-C2S	40	60	5	180	340.00	Bare unit
	7 1/2 C1	58	80	7.5	295	455.00	Bare unit
Westinghouse	2YS	20	80	3	200	377.00	Bare unit
	3YS	47.7	80	7.5	350	467.00	Bare unit
	5G	6.0	80	1	150	149.00	Bare unit

TABLE I-12
TYPICAL BLEED AIR CHARACTERISTICS AND ENGINE POWER LOSSES
[T64 - 2850 H.P. (Military Power Rating)]

% Bleed	CFM	Temp. °F	Pressure P.S.I.	Engine Power Loss - %
0	0	684	162	0
.7	25	679	159	2.5
1.4	50	674	155	5.0
2.1	100	664	149	10.0
3.4	125	659	146	12.0
4.1	150	654	143	14.5
6.0	225	641	134	20.0

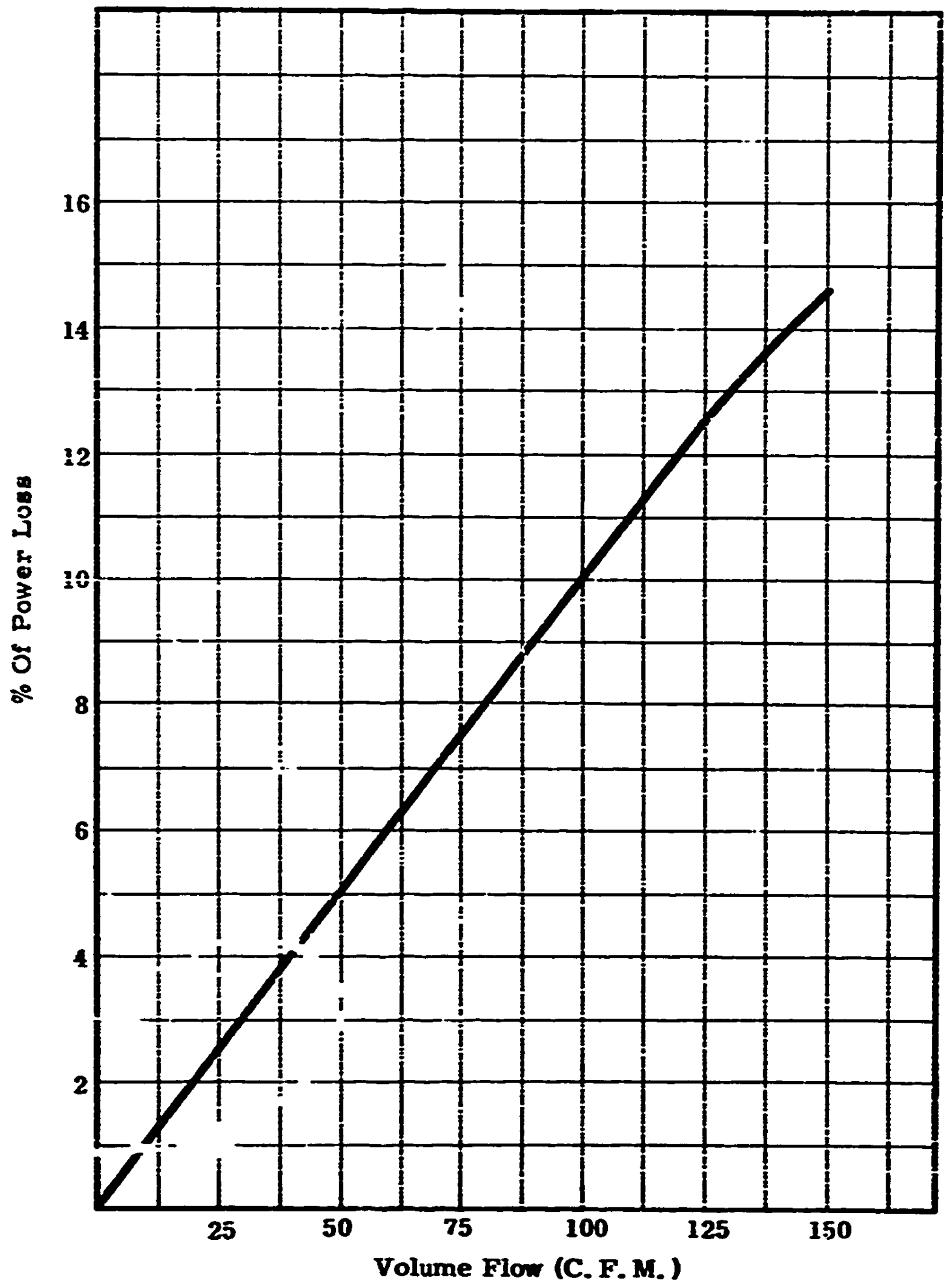


Fig. I -12. Volume Flow Vs. Power Loss At Constant Bleed Pressure

CHAPTER II

BASIC STUDIES

2.1 STRUCTURAL CHARACTERISTICS OF INFLATABLE STRUCTURES

2.1.1 Introduction

The basic differences between the behavior of inflatable structures and conventional structures are:

(1) The strength of the inflatable structure can be varied by changing the inflation pressure.

(2) Shear loads in the inflated structures are taken largely by components of fiber tension, which primarily arise from either initial fiber orientation or deflection of the fibers under load, rather than by the shear stiffness of the structural material.

(3) The fibers of the construction material can be selected and oriented to carry the maximum principal stresses at any point in the structure.

(4) Elastic limit loads on the inflated structure in general arise when the fabric buckles initially; reduction of the load immediately restores the structure to its original configuration and strength.

There are many possible forms that an inflated structure member can take depending on the selection of basic skin material and on the method of construction. A structure may be fabricated at one extreme from a large number of simple components, connected by noninflated joints, or at the other extreme by one complex but complete structure, utilizing sophisticated and exact selection and distributions of material and fibers to provide the desired strength and no more, and to maintain an accurate contour.

In the first case, simplicity of construction, ease of maintenance, and inherent safety in the event of the failure of one section are obtained at the expense of a smooth accurate contour and a multiplicity of connections to provide inflation air. In the second case, smooth accurate contour and structural efficiency are obtained at the expense of complexity and cost of construction, need for careful maintenance and careful repair, and the vulnerability inherent in having only one inflatable compartment. The decision to utilize one form or the other, or some compromise between them, can only be made after a thorough analysis of the advantages and disadvantages of each type in terms of weight, cost, maintenance, and vulnerability.

Several investigators have published analytical works on inflatable structures that provide a good base for future investigations, and some experimental data has been published to verify the theories, with modifications. In particular, for GEM configurations, the analyses on air mat beams and plates in references 13, 14, 15, and 16 are of interest. No attempt has been made here to utilize the results of these analyses, except where they develop fundamental relationships, or present basic data. Additional data is available in references 17, 18, 19, 20 and 21.

Basically, an inflated structure exhibits behavior very similar to that of a rigid structure. With proper design, it can take bending loads, torsion loads, tension loads, and shear loads; the load-deflection relationships are essentially the same for both types of structures, except for shear loads. Except in those cases where the structural material is thick as a result of small physical dimensions and high inflation pressure, shear loads are essentially taken by components of fiber tension introduced either by initial orientation of the fiber, or by change of the fiber orientation due to the applied load, or both. This means that shear deflection of the structure may be a greater proportion of the total deflection than would be the case for a rigid structure.

2.1.2 Technical Analyses

2.1.2.1 Definitions of Symbols Used in the Analysis

A area enclosed by the periphery of a cross-section,

A_y area enclosed by the periphery above a line drawn distance y from, and parallel to, the neutral axis,

A_m cross-sectional area of the material at a cross-section,

A_I A for an inflated cross section,

A_s surface area of inflated section,

b width of the flat sides of the example section,

B a constant,

c a constant,

C modulus of ridigity,

ϵ_x longitudinal strain at x, assumed uniform,

E modulus of elasticity,

E_i E for inflatable structure material,

E_R E for rigid structure material,

$\left.\begin{matrix} f_L \\ f_H \end{matrix}\right\}$ two stresses normal to each other,

f_p principal stress resulting from f_L, f_M, and q,

f_s hoop pressure stress,

f_{x_p} longitudinal pressure stress,

f_{x_B} longitudinal bending stress,

f_ℓ a longitudinal fibre stress

h depth of the example section,

I second moment of inertia of cross-sectional material, about the neutral axis,

I_I I for an inflatable cross-section,

I_R	I for a rigid cross-section,
J	polar moment of inertia of cross-sectional material about the torsion axis,
J_I	J for an inflatable cross-section,
J_R	J for a rigid cross-section,
K	factor allowing for non-uniformity of shear stress due to bending across beam section,
L	length of beam,
M	bending moment at a section in the beam,
n	normal from 0 to the tangent to a point on the periphery of the section,
O	a datum point within the enclosed area A,
P	perimeter of the inflatable cross-section,
p	inflation pressure,
q	a general shear stress,
q_y	q at distance y above neutral axis,
q_T	q due to torsion,
R	radius of curvature in bending of the beam,
r	radius of curved part of inflated cross-section,
s	distance around the periphery from a datum,
S	shear force at a section,
t	material thickness,
T_S	torsion in the inflatable skin,

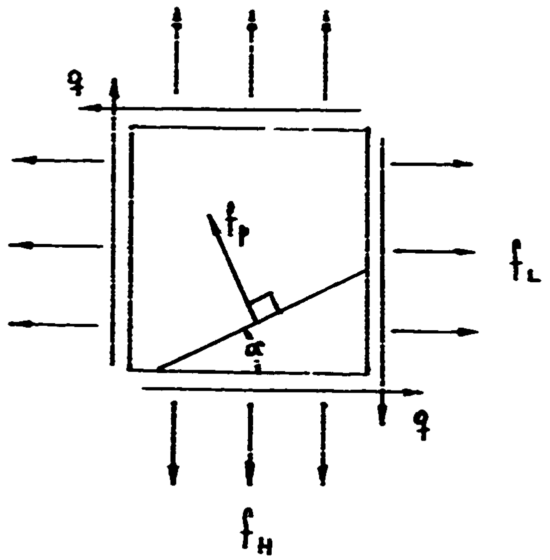
T	torque on a section,
U	total strain energy,
w_y	material thickness at y from the neutral axis,
x	coordinate distance along the beam,
$\bar{y}$	distance of centroid of A_y from the neutral axis,
y	coordinate distance from neutral axis,
Y	deflection of the neutral axis
z	coordinate distance normal to x and y,
α	angle between principal stress and f_m direction,
β	angle used in developing expression for hoop stress,
μ	Poisson's ratio,
ϕ	shear strain,
θ	angle of twist under torsional load.

2.1.2.2 Design Inflation Pressure

The first step in developing an inflatable structure to withstand a given loading situation is to determine the inflation pressure necessary to prevent wrinkling of the structure under the maximum operating loads, and to prevent collapse of the structure under design ultimate loads.

2.1.2.3 Loading Conditions at Which Wrinkling Starts

Consider that the fabric is homogenous and isotropic for the purposes of analysis. In an element of the material under load, the stresses applied are as shown in the sketch, and comprise tensile stresses, f_L, f_H in mutually perpendicular directions, and shear stresses q as shown in sketch.



The resultant principal stresses of this stress system are

$$f_p = \frac{f_L + f_H}{2} \pm \sqrt{\left(\frac{f_L + f_H}{2}\right)^2 + q^2 - f_L f_H} ,$$

so oriented that

$$\tan \alpha = \frac{f_p - f_H}{q} \quad \text{or} \quad \frac{q}{f_p - f_L} .$$

When wrinkling occurs, one value of the principal stress approaches zero, from a tensile direction; i.e., $f_p = 0$. For this to occur,

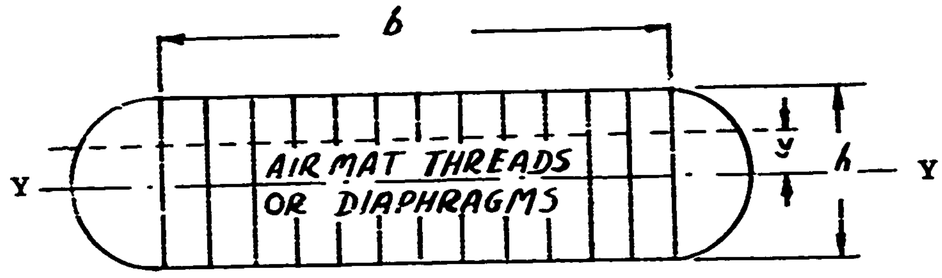
$$q^2 - f_L f_H = 0 \text{ or } q = \sqrt{f_L f_H} \text{ and } \tan \alpha = -\frac{f_H}{f_L} .$$

2.1.2.4 The Stresses, f_L, f_H, and q

The stresses at a location in the structure derive from the applied loads and are dependent upon the cross-sectional geometry of the structure at the location in question. The applied loads comprise; tension, compression, bending, shear, and torsion. Probably in any combination, together with the basic inflation loads required to provide structural integrity.

In evaluating the wrinkling loads for a given loading situation, the air is to relate inflation pressure to the applied load as a function of section geometry. The total values of f_L, f_H, and q, will be the algebraic sum of all stresses in the L, H, and shear directions induced by the applied loadings.

For simplicity in the analytical examples, a simple air mat beam, closed along the side with semi-cylindrical sections, will be utilized wherever analytically feasible; otherwise, a cylindrical section will be used. Using these sections the individual contributions due to the applied loads that go to make up the total stresses at any point in the structure will be examined.



The Example Section

The example section is composed of a parallel-faced air mat type section, consisting of two fabric load-carrying skins separated by drop-cords or woven diaphragms to which have been added sides of semi-cylindrical section.

The over-all dimensions are as shown in the sketch. It is assumed that the material thickness of the section is constant around the periphery and has the value t.

2.1.2.5 Geometric Characteristics of the Section

The geometric characteristics of the section are:

P	perimeter,
A	area of enclosed section,
I	second moment of area of load carrying material, about the horizontal plane of symmetry,
$(A_y \cdot \bar{y})$	first moment of area of load-carrying material between y and y h/2, taken about the horizontal plane of symmetry,
W_y	material thickness at plane y, = constant 2t.

The expressions for the above characteristics are:

Perimeter (P)

$$P = 2b + \pi h = \underline{2b\left|1 + \frac{\pi h}{2b}\right|} = \underline{\pi h\left|1 + \frac{2b}{\pi h}\right|}$$

Area (A)

$$\underline{A} = bh + \frac{\pi h^2}{4} = \underline{bh\left(1 + \frac{\pi h}{4b}\right)} = \underline{\frac{\pi h^2}{4}\left(1 + \frac{4b}{\pi h}\right)}$$

2nd Moment of Area (I)

$$I \doteqdot 2\,bt\left(\frac{h}{2}\right)^2 + \pi\left(\frac{h}{2}\right)^3 t$$

$$= \frac{bth^2}{2}\left[1 + \frac{\pi h}{4b}\right] = \frac{\pi th^3}{8}\left[1 + \frac{4b}{\pi h}\right]$$

1st Moment of Area A_y, $(A_y \cdot \bar{y})$

$$(A_y \cdot \bar{y}) = tb\left(\frac{h}{2}\right) + th\sqrt{(h/2)^2 - y^2}$$

$$= \frac{h}{b}\left[\frac{tb^2}{2}\right] \div th\sqrt{\left(\frac{h}{2}\right)^2 - y^2}$$

$$= \frac{b}{h}\left[\frac{th^2}{2}\right] \div th\sqrt{\left(\frac{h}{2}\right)^2 - y}$$

2.1.2.6 Stresses in the Inflated Beam

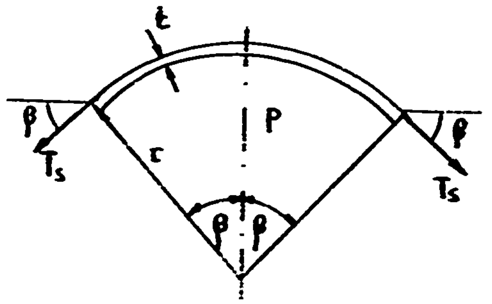
Let the y axis be normal to the neutral axis of the inflated beam, positive downwards. Let the x direction be along the beam, coincident with the L direction and the s direction be transverse at any point in the skin, coincident with the H direction (Sketch under 2.1.2.3) Hence, stresses in the x direction comprise longitudinal pressure stresses, bending stresses, compression and tension stresses. Stresses in the s directions are entirely hoop pressure stresses. Shear or q stresses arise from the shear associated with a varying bending moment and the shear due to torsion.

2.1.2.6.1 The Pressure Stresses in the Section

2.1.2.6.1.1 Hoop Stress

The hoop stresses in the curved part of the section are defined by the balance of forces acting on an element of the curved section.

Consider an element of unit width and length S along the curved portion. As indicated in the sketch, let x be the radius of curvature, let the angle formed between the radii at each end be 2β, and let the skin tension be T_S, reacting the load due to pressure p. For equilibrium of the element,



$$2T_S \sin\beta - p \cdot 2r \sin\beta = 0.$$

Therefore,

$$T_S = p \cdot r.$$

Now,

$$T_S = f_S \cdot t$$

$$\therefore\ f_S \cdot t = pr$$

and

$$f_S = \frac{pr}{t}$$

In the section under consideration

$$r = \frac{h}{2}$$

$$\therefore\ f_S = \frac{ph}{2t}$$

Since the drop cords, or diaphragm, take the pressure stresses between the two parallel sides of the section, it can be seen that this value of f_y holds true for the whole periphery of the section.

2.1.2.6.1.2 <u>Longitudinal Stress, f_{x_p}</u>

The longitudinal stress developed is dependent on the end conditions of the tube or beam, whether it be constant stress across the section or constant strain throughout the length of the beam. Because of the appreciable values of Poisson's ratio μ that have been established for some woven-coated fabrics, reference 19, the assumption of uniform stress cannot be fully justified a priori. However, uniform strain does seem a practical condition and is used here.

For uniform strain e_x, and constant Young's modulus, E.

$$Ee_x = f_{x_p} - \mu f_s$$

$$= f_{x_p} - \mu \frac{ph}{2t} .$$

Now for longitudinal equilibrium at any cross-section,

$$pA = \oint t f_{x_p} . ds$$

where ds is an elemental length of the perimeter. Therefore,

$$pA = \oint t \left| Ee_x + \frac{\mu ph}{2t} \right| ds$$

and since e_x is assumed constant

$$pA = t \left| Ee_x + \frac{\mu ph}{2t} \right| P$$

$$= t \left[\left| f_{x_p} - \frac{\mu ph}{2t} \right| + \mu \frac{ph}{2t} \right] P$$

$$= Ptf_{x_p}$$

This same result is arrived at in this case by assuming uniform stress distributions. Therefore,

$$f_{x_p} = \frac{pA}{Pt} = \frac{ph}{4t} \frac{\left(1 + \frac{4b}{\pi h}\right)}{\left(1 + \frac{2b}{\pi h}\right)}$$

2.1.2.6.2 <u>The Bending Stresses in the Section</u>

With the same assumptions that govern the conventional bending theory, the same expressions apply "in toto".

Bending moment $M = EI . \frac{d^2 y}{dx^2}$

$$\frac{M}{I} = \frac{E}{R}, \text{ and } \frac{E}{R} = \frac{f_{x_B}}{y}, \text{ from conventional theory.}$$

Therefore, since $f_{x_B} = \frac{My}{I}$

$$f_{x_B} = \frac{My}{\frac{\pi t h^3}{8}\left[1 + \frac{4b}{\pi h}\right]}$$

2.1.2.6.3 <u>The Shear Stresses in the Section</u>

2.1.2.6.3.1 <u>Vertical Shear, S, Accompanying Bending</u>

The shear stress distribution in a beam subject to both bending and shear from a given applied load is

$$q_y = \frac{S}{2t} \cdot \frac{A_y . \bar{y}}{I}, \text{ from conventional theory.}$$

where the expression for $\frac{A_y \cdot \bar{y}}{I}$ from the previous section characteristic summary, becomes

$$\frac{A_y \cdot \bar{y}}{I} = \frac{\left(\frac{b}{h}\right)\left[\frac{th^2}{2}\right] \cdot th\sqrt{\left(\frac{h}{2}\right)^2 - y^2}}{\frac{\pi th^3}{8}\left[1 + \frac{4b}{\pi h}\right]}$$

$$= \frac{\left(\frac{b}{h}\right) + \sqrt{1 - \left(\frac{2y}{h}\right)^2}}{\frac{\pi h}{4}\left[1 + \frac{4b}{\pi h}\right]}$$

therefore

$$q_y = - \frac{S}{2t} \; \frac{\left(\frac{b}{h}\right) + \sqrt{1 - \left(\frac{2y}{h}\right)^2}}{\frac{\pi h}{4}\left[1 + \frac{4b}{\pi h}\right]}$$

2.1.2.6.3.2 Rotational Shear Due to Torsion

Again from thin-walled tube analysis, with the same assumptions, the following applies:

$$q_T = \frac{T}{2At} = \frac{T}{\frac{\pi h^2 t}{2}\left[1 + \frac{4b}{\pi h}\right]}$$

2.1.2.7 Evaluation of Inflation Pressure p Required to Prevent Wrinkling

At each point in the section, wrinkling commences when $q = \sqrt{f_L f_H}$, as developed earlier. The required inflation pressure is that value which, if used, will prevent wrinkling anywhere in the section; hence, it must be equal to or greater than the maximum value derived from the above expression.

The expression, when appropriate substitutions are made, becomes

$$S \cdot \left(\frac{A_y \cdot \bar{y}}{W_y \cdot I} \right) + \frac{T}{2At} = \sqrt{\left\{ \frac{My}{I} + \frac{ph}{4t} \left[\frac{1 + \frac{4b}{\pi h}}{1 + \frac{2b}{\pi h}} \right] \right\} \frac{ph}{2t}} \; .$$

From this $\quad p^2 + Bp - C = 0.$

where

$$B = \left\{ \frac{M \frac{h}{2tI}}{\frac{1}{2} \left(\frac{h}{2t} \right)^2 \left[\frac{1 + 4b/\pi h}{1 + 2b/\pi h} \right]} \right\} y$$

and

$$C = \left\{ \frac{\left[S \left(\frac{A_y \cdot \bar{y}}{W_y \cdot I} \right) + T \left(\frac{1}{2At} \right) \right]^2}{\frac{1}{2} \left(\frac{h}{2t} \right)^2 \left[\frac{1 + 4b/\pi h}{1 + 2b/\pi h} \right]} \right\} .$$

Solving the quadratic gives

$$p = -\frac{B}{2} \pm \sqrt{\left(\frac{B}{2} \right)^2 + C} \; .$$

Now, as y decreases from $+\frac{h}{2}$ to zero

	$\lvert B \rvert$	decreases
	C	increases
until at $y = 0$	$\lvert B \rvert = 0$	
	C	is a maximum.

Hence, the maximum value of $+p$ between $y = +\frac{h}{2}$ and $y = 0$ occurs when $y = 0$. However, as y decreases from zero to $-\frac{h}{2}$, $\left(-\frac{B}{2}\right)$ increases and C decreases, until at $y = h/2$

$$p = \pm\frac{B}{-2} + \frac{B}{2} = B \text{ or } 0.$$

In order to determine whether this is the maximum value of p required in the section, the value of p needs to be determined as a function of y; a rather complex analysis is required in order to determine the precise location that gives the value of the maximum required inflation pressure, and is considered too involved for the purpose of this study.

In a practical thin-walled structure, and particularly in the section under consideration, the shear stress in the compression and tension faces of the section is quite substantial, becoming more so as the section $\left(\frac{b}{h}\right)$ parameter increases. Under these conditions it is realistic to consider the maximum pressure condition as arising at the compression face, with full shear taken into consideration.

For many beam problems, where the loading can be considered equivalent to that on a simply supported beam, the maximum bending moment coincides very closely with zero shear, and except for very short beams the bending stresses are the dominant stresses in the beam. Under these conditions, the inflation pressure can be estimated purely from the maximum applied bending moment (and the applied torsion if it exists) and the maximum pressure condition taken to occur on the compression face. Consequently, for the purposes of this analysis, it will be assumed that p does not reach a maximum between $y = -\frac{h}{2}$ and $+\frac{h}{2}$, but has its highest value at $y = -\frac{h}{2}$, i.e., on the compression face, at which point the contribution of the shear stresses to the stress distribution is minimized.

If both T and S are zero, then the shear stresses are zero, and the principal stresses depend entirely on the bending and pressure stresses.

Under these circumstances, $f_{x_B} + f_{x_p} = 0$ at wrinkling, i.e.

$$\frac{My}{\frac{\pi t h^3}{8}\left[1 + \frac{4b}{\pi h}\right]} + \frac{ph}{4t}\,\frac{\left[1 + \frac{4b}{\pi h}\right]}{\left[1 + \frac{2b}{\pi h}\right]} = 0.$$

Therefore,

$$p = \frac{32}{\pi} \cdot \frac{(-My)}{h^4}\left[\frac{1 + \frac{2b}{\pi h}}{\left(1 + \frac{4b}{\pi h}\right)^2}\right].$$

Clearly the maximum value of the pressure occurs for $|y| = \frac{h}{2}$ and when (-My) is positive; or, in other words, when the pressure stress counterbalances the compression stress due to bending, to give zero stress in the outer fibers on the compression face of the beam. Therefore, the desired result is

$$p = \frac{16}{\pi} \cdot \frac{M}{h^3}\left[\frac{1 + \frac{2b}{\pi h}}{\left(1 + \frac{4b}{\pi h}\right)^2}\right].$$

2.1.2.8 Inflated Column in Compression

Another loading condition of interest is that of the column in compression -- here, the requirements that govern columns with shear flexibility are satisfactory, when modified to allow for the effective shear modulus of the inflated column.

2.1.2.9 Discussion of Inflation Pressure p to Prevent Collapse

Having discussed the applied pressure loads required to prevent wrinkling of the structure, some consideration must be given to collapse of the structure, i.e., the load at which the structure will completely collapse at some critical section. The collapsing load is equivalent to the ultimate load for conventional structures, whereas the wrinkling load is equivalent to the elastic limit load of such structures.

The collapse load is reached when the structure has wrinkled all the way around until only the outside fibers remain in tension. At this point, these fibers carry all of the pressure load (pA). If it can be assumed that:

(1) The neutral axis of the section does not change during wrinkling,

(2) The outer fibers can carry the total tensile pressure load, then collapse moment can be taken as

$$\left(pA \cdot \frac{h}{2}\right)$$

or

$$p \cdot \frac{\pi h^3}{8} \left|1 + \frac{4b}{\pi h}\right|$$

and this is equal to

$$(M\ \text{wrinkling}) \cdot 2 \left[\frac{1 + \frac{2b}{\pi h}}{1 + \frac{4b}{\pi h}}\right] .$$

With these assumptions, collapse moment divided by the wrinkling moment lies between 1 and 2.

Note, however, that it is doubtful that these assumptions can be rigorously held, since the neutral axis must move toward the tension side, and the strength of the outer fibers will be limited. If the assumptions were true, the stress in the outer fibers would tend to infinity. Since no available materials can withstand such stresses, such conditions should result in failure of the fabric; however, many tests have been carried out without fabric failure, indicating the assumptions cannot hold.

More sophisticated hypotheses have been developed indicating collapse values that lie in between the simple value and the first wrinkling moment. Much work remains to accurately determine and verify the collapse conditions; for preliminary design work, "collapse" and "first wrinkling" can be considered as coincident, and the structure can be designed to its ultimate strength at "first wrinkling."

2.1.2.10 Structural Deflection

The basic structural strength having been determined by the collapse and wrinkling loads on the structure, the structural stiffness can now be determined by choice of material properties and thickness.

The total structural deflection in any given direction is composed of translation and rotation in that direction. For a beam, the total deflection is made up of deflection due to bending and deflection due to shear. For a torque tube the deflection is due purely to the torsion loads.

2.1.2.10.1 Beam Deflections

2.1.2.10.1.1 Deflection Due to Bending

From conventional bending theory:

$$Y_{x_2-x_1} = \frac{1}{EI}\int_{x_1}^{x_2}\left(\int_{x_1}^{x_2} M\,dx\right)dx.$$

This is true, as long as the basic assumptions for bending hold true, which are:

(1) Plane sections normal to the beam longitudinal axis remain plane.

(2) Section depth is small compared to the radius of curvature.

(3) Neutral axis of the beam coincides with the line joining the centroid of area of every cross-section.

(4) Beam cross-section is constant.

(5) Beam material is uniform.

2.1.2.10.1.2 Deflection Due to Shear

The over-all shear modulus of a portion of structure in the elastic region is:

$$\frac{\text{Over-all Shear Stress}}{\text{Shear Strain}}.$$

The over-all shear stress is:

$$\frac{\text{Shear Force}}{\text{Material Cross-Sectional Area, } A_m}.$$

In an inflated beam, at any cross-section where the shear strain is Φ,

$$\text{Shear force} = (\text{Net tensile force} \times \Phi)$$

and since the tensile forces due to bending cancel out along the beam axis

$$\text{Net Tensile Force} = \text{Pressure Force } (p \times A)$$

where A is the enclosed area of the beam. Therefore,

$$\text{Shear Stress} = \frac{p \times A \times \Phi}{A_m}$$

and

$$\text{Shear Modulus} = \frac{p \times A \times \Phi}{A_m \times \Phi} = \frac{pA}{A_m} .$$

Shear Deflection

Again,

$$\text{Shear Modulus} = \frac{\text{Shear Stress}}{\text{Shear Strain}}$$

$$= \frac{K.q}{\frac{dy}{dx}}$$

where K allows for non-uniformity of shear stress across section and is greater than 1

$$= K . \frac{S}{A_m} . \frac{dx}{dy} .$$

Now $$S = \frac{dM}{dx}$$

therefore, $$\text{Shear Modulus} = \frac{K}{A_m} \cdot \frac{dM}{dx} \cdot \frac{dx}{dy}$$

$$= \frac{K}{A_m} \cdot \frac{dM}{dy} .$$

Hence $$\frac{pA}{A_m} = \frac{K}{A_m} \cdot \frac{dM}{dy}$$

$$dy = \frac{K}{pA} \cdot dM,$$

and $$\text{Shear Deflection} = \frac{K}{pA} (M_2 - M_1)$$

where subscripts 1 and 2 refer to the points between which deflection is desired. The equivalent formula for a rigid beam with the same cross-sectional area is

$$\text{Shear Deflection} = \frac{K}{CA} (M_2 - M_1)$$

where C is the modulus of rigidity of the material. Hence, an <u>equivalent</u> modulus of rigidity in this case appears to be the pressure p, when comparing inflatable and rigid beams of the same external form.

<u>Note:</u> K is obtained from a strain energy analysis of the beam with unit loading. For example, a cantilever with the desired cross-section, loaded with unit load at the free end gives rise to

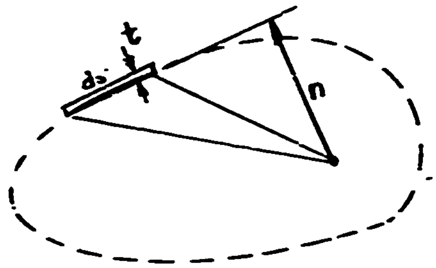
$$K = \frac{U}{\left(\frac{L}{2A_m C}\right)}$$

where L is the beam length and U is the total strain energy absorbed due to unit load. A typical value for K is $\frac{6}{5}$, for a rectangular cross-section

Torsional Deflections

As discussed earlier, the material of the inflated structure possesses virtually no shear stiffness on its own account. However, inflated tubes have demonstrated torsional rigidity; therefore the torsional rigidity must arise primarily by virtue of the inflation pressure.

Consider a peripheral length ds of the cross-section of an inflated tube of length L (see sketch). Let the angle of twist on the surface, measured by a straight line generator on the surface of the tube, be ϕ. Let the longitudinal pressure stress be f_ℓ, and let transverse plane sections remain plane.



Then the component of longitudinal pressure stress in the plane of the cross-section = $f_\ell \sin \Phi$.

Hence, the torque developed by element ds, about some point 0 within the section, is $[(f_\ell \sin \Phi \cdot t\, ds)\, n]$ and the total torque on the section is = § $(f_\ell \sin \Phi \cdot t\, ds) n$, integrating around the complete periphery.

In this expression, f_ℓ is constant for uniform longitudinal strain, t is constant, $\phi = \frac{n\theta}{L}$, therefore,

$$\text{Torque } T = \oint f_\ell \sin \Phi \; t \; n \; ds$$

$$= \oint f_\ell \frac{n\theta}{L} \; t \; n \, ds$$

$$= f_\ell \ell \frac{\theta}{L} \oint t \, n^2 \; ds$$

$$= f_\ell \ell \frac{\theta}{L} \; J$$

where J is the polar moment of inertia of the section material. Now, $f_s = \frac{A}{Pt}$ where P is the section perimeter

$$\therefore \; T = \frac{\theta}{L} \; J \; \frac{pA}{Pt}$$

Conventional torsion theory gives.

$$T = C. \; \frac{\theta}{L} \; J .$$

Hence, $C \equiv \left(\frac{pA}{Pt}\right)$ as found for the bending case. The torsional deflection θ is obtained from

$$\theta = \frac{\frac{TL}{J}}{\left(\frac{pA}{Pt}\right)}$$

2.1.2.10.1.3 **Deflection Comparison Between Rigid and Inflatable Structure**

The ratios between corresponding rigid and inflatable structure deflections, for beams of the same applied loading, are as follows:

Bending $\quad \frac{\text{Rigid Deflection}}{\text{Inflatable Deflection}} = \frac{E_I I_I}{E_R I_R}$

Shear and Torsion $\quad \frac{\text{Rigid Deflection}}{\text{Inflatable Deflection}} = \frac{J_I \cdot A_I \cdot p}{J_R \cdot A_m \cdot C}$

Note:

The appropriate material geometry in each case must be used in evaluating geometric quantities I_I, I_R, A_m, J_I, and J_R, while A_I is purely the inflated cross-sectional area.

2.1.3 INFLATABLE STRUCTURE STRENGTH ESTIMATION AND STRESSING

The maximum values of bending moment, shear force, torsion, etc., that will be met with, first in normal operation, and secondly in emergency (crash) operations are first determined for each portion of the structure.

A safety factor is applied to these values to account for estimating discrepancies, and then the resulting loads utilized to determine

(1) the inflation pressure required to prevent wrinkling under normal load conditions, and

(2) the inflation pressure required to prevent collapse under emergency load conditions

at each point in the structure for which loading data is available. At this stage in design, it will be advisable to check the structure geometry, to provide a common required inflation pressure for as much of the structure as possible so that reasonable structural efficiency is maintained. The choice of inflation pressure will generally tend towards the highest that is acceptable from construction and environmental considerations. Further investigation may be necessary to optimize structure weight after the design has progressed into the material selection stage.

Having decided on the required inflation pressure, and modified the structural geometry where necessary, the detailed stressing analysis is conducted to determine the required cloth strengths and orientations for maximum effectiveness. In selecting or specifying cloth or fibers to meet the strength requirements, an allowance must be made to ensure that the fiber stress level is sufficiently below the "quick break" strength to ensure adequate service life for the vehicle. (Typical variations of fiber strength with duration of load were shown on Fig. 2 in Chapter I). Accepted factors on the "quick break" strength of nylon and Dacron fibers are of the order of .25 to .75 (reference 22) depending on the required vehicle life, the method of determining the "quick break" strength, and the probable frequency and duration of the designing load condition during the vehicle life.

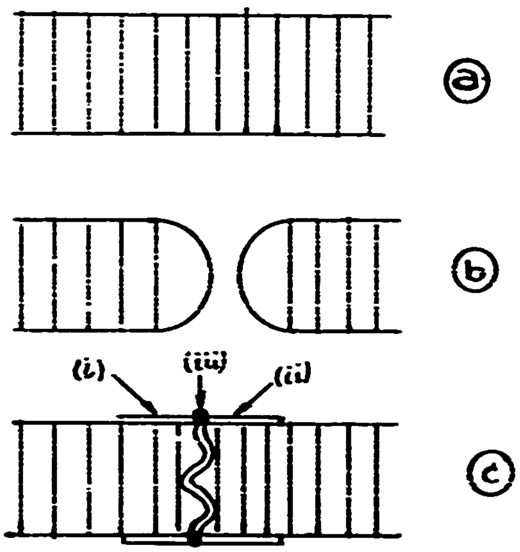
2. 2 LOAD DISTRIBUTION TECHNIQUES AND JOINT ANALYSIS

Because of the flexible nature of the structural material of inflatable structures, conventional techniques for load transfer from one member to another, or from an inflated member to a rigid member, must be carefully examined for validity in this type of structure.

2. 2. 1 JOINTS BETWEEN INFLATED SECTIONS

If an inflated structure can support a given loading condition, and is then divided into two or more sections, the joints between the sections must be capable of providing the same strength as the material that was originally present. Hence, if the joint is so constructed as to operate essentially with the same loading as the original material as satisfactory solution will have been reached.

One approach that has been used in practice is illustrated below, and can be used in areas subject to any loading condition for which an inflatable structure can be evolved; sketch (a) indicates the basic structure. Sketch (b) shows the structure divided into two inflated sections. Sketch (c) shows how the stressing conditions of (a) are essentially reproduced in joining (a) and (b). The joint is provided by two strips (i) and (ii) on each face, joined at (iii). These strips are sized to take the tension load



in the skins, and to provide adequate bond strength between themselves and the skins. The location of the strips is the important factor for adequate joint performance. The joint line is located at the end of the straight surface on each inflated section, and the joint is made prior to full inflation. When fully inflated, the mating hemispherical surfaces become redundant, and do not carry any appreciable load, merely serving as pressure seals. As long as the tensions that are present in both faces of the inflated section are substantially normal to the joint line, the joint will be satisfactory, regardless of the loading condition that gives rise to the skin tensions.

2. 2. 2 JOINT BETWEEN FABRIC AND METAL COMPONENTS

Because of the uncertainties attendant upon bonding metal to nonmetallic surfaces in this field, it would be wise at the present time to provide these joints without such bonding. This means that an intermediate jointing device must be used between the fabric and the hardware. Such intermediate devices are lacing flaps bonded to the fabric, load speading nets, cables looped into fabric load spreading plates, continuous joints comparable to zippers.

2. 2. 3 JOINT DESIGN

No complex analyses are required to design joints of the types indicated in 2. 2. 1 and 2. 2. 2; these will constitute the majority of joints in an inflated structure. It is wise to distribute the loads as much as possible to prevent stress concentrations in the structure; the provision of adequate material with the right orientation, together with careful loading techniques, is all that is required thereafter to ensure an adequate joint strength.

CHAPTER III

CONCEPTUAL STUDIES

3.1 DISCUSSION OF LOTS CARRIER ROLE AND MISSION REQUIREMENTS

3.1.1 ROLE OF THE LOTS CARRIER

In this program a particular GEM is considered whose requirements are relatively well defined, notably an amphibian primarily for the logistic-over-the-shore (LOTS) operations capable of transporting 15 tons of cargo. A vehicle of this type can reasonably be expected to make a real contribution to mobility of forces. Much of the material developed in other GEM studies and already published bears this out and it is not within the scope of this program to further develop this particular aspect of the GEM. This study does, in any event, derive its importance largely from considerations of strategic mobility--the ability to move significant forces to any point in the world with minimum delay.

The 15-ton carrier, while primarily an amphibian for LOTS operations, will be capable of use on inland waterways and across country. In addition, it is expected to permit operations in remote and otherwise inaccessible areas. The speed of the vehicle under adverse conditions could be sustained at between 25 and 40 miles per hour, although, from technical considerations, the optimum speed is much higher for a GEM.

A more complete impression of LOTS operations can be obtained by reference to current staff manuals of the Army. There are also technical studies (reference 23 is representative) from which the developing trends can be inferred. One point stands out: supply movements across beaches can be expected to a greater extent with increasing port vulnerability to nuclear weapons. A lighter or amphibious craft for unloading ships standing off shore will therefore play a greater role than it would if it were intended exclusively for support of amphibious assault operations. It will handle bulk cargo, containerized cargo, vehicles and major equipment items, and personnel. In short, each craft could handle the major resupply for an active overseas theater of operations.

Some of the characteristics of LOTS operations are not precisely determinable and this is what one would expect. The distance off shore at which the ships will orbit depends on size of the operation, weather, enemy activity, etc. The distance inland at which the amphibians will unload will depend on terrain, progress of operations, and availability of all types of transport. The time for the lighter to perform one full cycle of operation depends on the above two factors, plus unloading rates from the ship's cargo hatches, unloading of the lighter at the inland resupply point, queueing operation, and other sources of lost time. The ranges being contemplated to as high as 30 miles off shore distance and 10 miles inland distance. If GEMs are selected for the role of LOTS carrier, it will most likely be for the advantage of speed they offer; and the speed would be no real asset unless distances toward the high end were contemplated. For planning purposes, in this study, it was assumed that a combined over water- overland distance of 20 miles would be representative.

The environment under which LOTS operations take place must, in the over-all view, be regarded as unfavorable. The major equipment item in the system--the deep draft, ocean going vessel--has evolved slowly over the years and is geared to cargo transfer at dockside or in sheltered harbors. Major changes in this characteristic are not envisioned within the next 10-year period. The exigencies of war mean operations in unfavorable weather (and the sea can multiply the effect of weather) at poor locations, and with incompletely trained personnel. The consequence of all this is that equipment needs to be adequately rugged to perform a mission of the kind under consideration. A full development of this subject appears later in this report.

3.1.2 SUMMARY MISSION DATA

It is necessary to define the kind of operations intended for the vehicle(s). A primary mission takes first priority, but it is often possible to incorporate design features to make a vehicle flexible and versatile. Hence, a number of secondary or alternate missions are postulated.

3.1.2.1 Primary Missions of a 15-ton Cargo Amphibious GEM LOTS Carrier

1. LOTS carrier normally arrives at locale of amphibious operation aboard ocean-going transports.

2. Carrier is lowered to water, fully assembled.

3. Cargo is transferred to LOTS carrier by booms of transport.

4. Reach of boom with 5-10 tons is adequate without reversing carrier if beam does not exceed 36 feet.

5. Handling lines and low propulsive power are used for steadying carrier during loadings.

6. Fuel for hovering during loading is not contemplated.

7. Typical range without refueling is 60 miles, i.e., one round trip from ship to over shore and return.

8. Low density cargo down to 75 pounds per square foot to be accommodated.

9. High density cargo up to 300 pounds per square foot to be accommodated

10. Containerized cargo to be sized: 6 feet 3 inches wide, 6 feet 10-1/2 inches high, 8 feet 6 inches long.

11. Off-loading is to be accomplished generally by fork lift.

12. A ramp will facilitate off-loading by fork lift or by manual effort.

13. Typical passage might be in range 10 per cent to 60 per cent over land and remainder over water, depending on nature of operation.

14. Surf up to five feet in height must be negotiated but a cargo reduction for extreme heights does not compromise essential characteristics.

15. Alternate use in lightering unit personnel and light unit equipment ashore.

16. Maintenance operations to be minimized with probable absence of third and fourth echelon.

17. Speeds in the range 25-40 miles per hour are consistent with LOTS operations.

18. Movement of LOTS carriers by air transport (Phase III airborne operations), either inter-theater or intra-theater is a possibility.

19. Carrier shall be capable of sustaining and operating with a threefold overload.

3.1.2.2 <u>Secondary Missions for a 15-ton Amphibious GEM LOTS Carrier</u>

1. Types of Operations:

   - Over inland waterways
   - Movement from off-shore bases
   - Penetration into land areas prohibitive to conventional equipment.

2. Movement of tactical units on the carrier may be equal to or greater than movements of bulk or packaged supplies.

3. Maximum diversity of palletized loads and vehicles to be accommodated.

4. To expedite vehicle handling, ramps fo.. and aft are desirable.

5. Personnel may also be carried more frequently.

6. Provisions for fueling to increased ranges up to 200 miles shall be included.

7. Higher echelon maintenance may be possible.

8. Operation for extended periods at reduced surface clearances is likely.

9. Benefits of lower operation may be taken either as reduced fuel consumption, greater payload capacity or greater range.

10. Carriers may be frequently knocked down and transported overland by truck and/or train.

11. Transport by air is a necessity.

3.1.3 THE SEQUENCE OF GEM VEHICLE EMPLOYMENT

There are other phases in the employment of the vehicles besides actual accomplishment of missions as defined in the foregoing sections. It is well to list the entire sequence of uses to assure that any factors which might influence the design are not overlooked; they are:

1. Depot storage--adequate preservation and protection from environmental extremes is probable.

2. Training use--considerable numbers of vehicles must be assigned to new operator training; also unit training will be necessary and may take place under deliberately rigorous conditions.

3. Inactive periods--these periods often are a hazard to mechanical equipment.

4. Maintenance operations--the design must be conducive to simplified maintenance.

5. Deployment--administrative moves of this size vehicle are most probably by rail and water; partial knockdown.

6. Tactical moves--these invoke maximum transportability features of design, including rapid reassembly.

7. Tactical employment--mission summaries give the details.

3.1.4 DEVELOPMENT OF TYPICAL GEM LOTS CARRIER FOR STRESSING PURPOSES

For the dual purposes of this analysis and reference 24, a preliminary design study has been completed to establish the geometry and power requirements of a GEM suited to the QMR (reference 25).

3.1.4.1 Ground Rules

The ground rules established for this study are limited to the framework of the QMR (reference 25) and the earlier discussions of this chapter; they are:

Normal Operation

15-ton payload.
2-1/2 foot maximum operating height.
25 knots continuous cruise speed.
40 knots maximum cruise speed (still air).
(Payload + Fuel) weight approximately equal to empty equipped weight

Overload Operation

Overload (Payload + Fuel) equal to 3 x normal (Payload + Fuel) capacity.

3.1.4.1.1 Definitions of Symbols used in the Analysis

A_L louver area per louver

C cushion perimeter

Dh hub diameter

De equivalent diameter of cushion

D_{opt} optimum fan diameter

h operating height

ΔH_D head loss in duct between fan and jet exit

H_e total head at jet exit, normal operating height

H_E total head at jet exit, reduced operating height

H_L total head at louver exit

i approximate cushion length, for 2:1 planform

$(L/D)_{EFF}$ effective Lift/Drag ratio

N number of fans

p_c cushion pressure

p_o atmospheric pressure

Q volume flow through fan, at normal operating height

Q_E volume flow through fan, at reduced operating height

Q_L volume flow through louver, per louver

S cushion area

SFC specific fuel consumption

t jet thickness

t_e jet thickness at exit

T_L louver thrust per louver

V vehicle speed

V_L flow velocity through each louver

W vehicle weight

W_E equipment weight

W_F fuel weight

W_G gross weight

$(W_G)_o$ overload gross weight

W_{PL} payload weight

W_{PP} total power plant installed weight

$(W_{PP})_L$ lifting system power plant installed weight

$(W_{PP})_P$ propulsion system power plant installed weight

η_a = augmentation efficiency

η_p = propulsion efficiency

ρ = atmospheric density

σ = fan characteristic parameter

ω = fan revolutions per second

η_L = louver efficiency

3.1.4.2 Vehicle Layout

A preliminary assessment of vehicle size can be made from consideration of cargo-deck requirements for typical LOTS carrier cargo loads. The loads considered include Conex containers and a wide variety of vehicular cargo. Table 3.1 shows a typical cargo composition for the normal payload of 15 tons, the data having been derived from Army handbooks FM 101-10.

3.1.4.2.1 Cargo Area

The QMR requires a maximum overload capability of three times the normal (payload plus fuel) load, which in turn is about equal to the empty weight of the vehicle. Hence, gross weight of the vehicle in the overload condition will be:

$$\left(W_G\right)_o = 3\left(W_{PL} + W_F\right) + 1/2\, W_G = 2W_G$$

if

$$W_G = \left(W_{PL} + W_F\right) + 1/2\, W_G \,.$$

Assuming that all of the addition is taken in payload, the overload total payload

$$= 2\,W_G - 1/2\,W_G - \left(W_F\right) = 3/2\,W_G - \left(W_F\right)$$

therefore

$$\left(\frac{W_{PL}}{W_G}\right)_o = 3/2 - \left(\frac{W_F}{W_G}\right)_o$$

In order to size the cargo area, an estimate must be made of the number and suitable disposition of the Conex containers in the loaded condition. Conex containers are chosen as being the most likely cargo to be generally used in the overload case, since they are very widely utilized for delivering all forms of supplies.

By consideration of a suitable range of gross weights from two to three times the design payload of 15 tons, the expression above gives a range of Conex containers, from 7 to 11, as being probably usable in the overload case.

It is necessary to match the cargo capability closely between the overload and normal load cases in order to ensure the maximum utilization of cargo area with the minimum of wasted space. A study of the probable cargo listed in Table 3-1 reveals that a cargo deck length of at least 46 feet will permit the loading of several vehicular cargo combinations of close to 15 tons, and still leave sufficient deck space to load additional cargo to make up a total of 15 tons. With this in mind, a layout of Conex containers was examined, and the layout chosen that will provide the maximum number of containers between 7 and 11, in a cargo length not much

Table 3-1
Typical Cargo Composition for 15-Ton Payload

	Vehicle	Loading Condition	Number of Vehicles	Men & Materiel Additional Cargo
1	2 1/2-T LWB truck	Empty	2	2.4 Tons
2	2 1/2-T XLWB truck	Full	1	4.7
3	105 mm Howitzer	-	2	10.0
4	155 mm Howitzer	-	1	9.0
5	1/4-T Utility	Full	2	3.4
6	1 1/2-T Trailer or Water Carrier	Full	3	6.9
7	2 1/2-T Dump truck	Empty	2	0.4
8	5-T Cargo Carrier	Full	1	0.1
9	7 1/2-T Prime mover	-	1	0.4
	Vehicle Combinations			
A	1 + 6	Full	1	3.5
B	2 + 6	Full	1	3.0
C	2 + 3	Full	1	3.7
D	1 + 4	Full	1	0.2
E	1 + 7	Empty	2	1.4
F	1 + 6 of 5	Empty	7	1.5
	Conex Containers	Full	2	4
	5-T Capacity 5.5 T Gross Wt. Full	2 @ 5-T 1 @ 3-1/2 T	3	0

greater than 46 feet. Figure 3-1 shows that the condition is achieved for 10 containers, in a cargo area at least 47 feet long by 20 feet wide.

Note that 10 Conex containers, to meet the overload requirement, correspond to a normal gross weight of more than 37 tons, and probably as much as 40 tons.

3.1.4.2.2 Over-all Dimensions and Possible Cushion Area

The proposed vehicle layout is as follows:

1. A main cargo deck, flanked on either side by lift power modules, with ramps at each end.

2. The annular jet is, where possible, located close to the axis of the lift fans to minimize losses.

3. Forward bow sections are located in front of the two module groups, and provide locations for control cabins.

4. Propulsion is provided by separate variable-pitch air propellers mounted at the rear of the vehicle.

With the above arrangement, further dimensions can be determined. Module width will be on the order of 7 feet, giving an over-all width of about 34 feet, and a width of the cushion of approximately 27 feet.

Further definition of dimensions is dependent on the desired range and speed characteristics of the vehicle, and the resulting optimum cushion pressure.

3.1.4.2.2.1 Cushion Pressure, Installed Power and Gross Weight Relationships

The following expressions are used, based on simple annular jet theory, conservative structure and power estimations, for a 2:1 planform ratio design:

Total Lift HP

At 2-1/2 feet operating height, including an allowance of 15 per cent for stability:

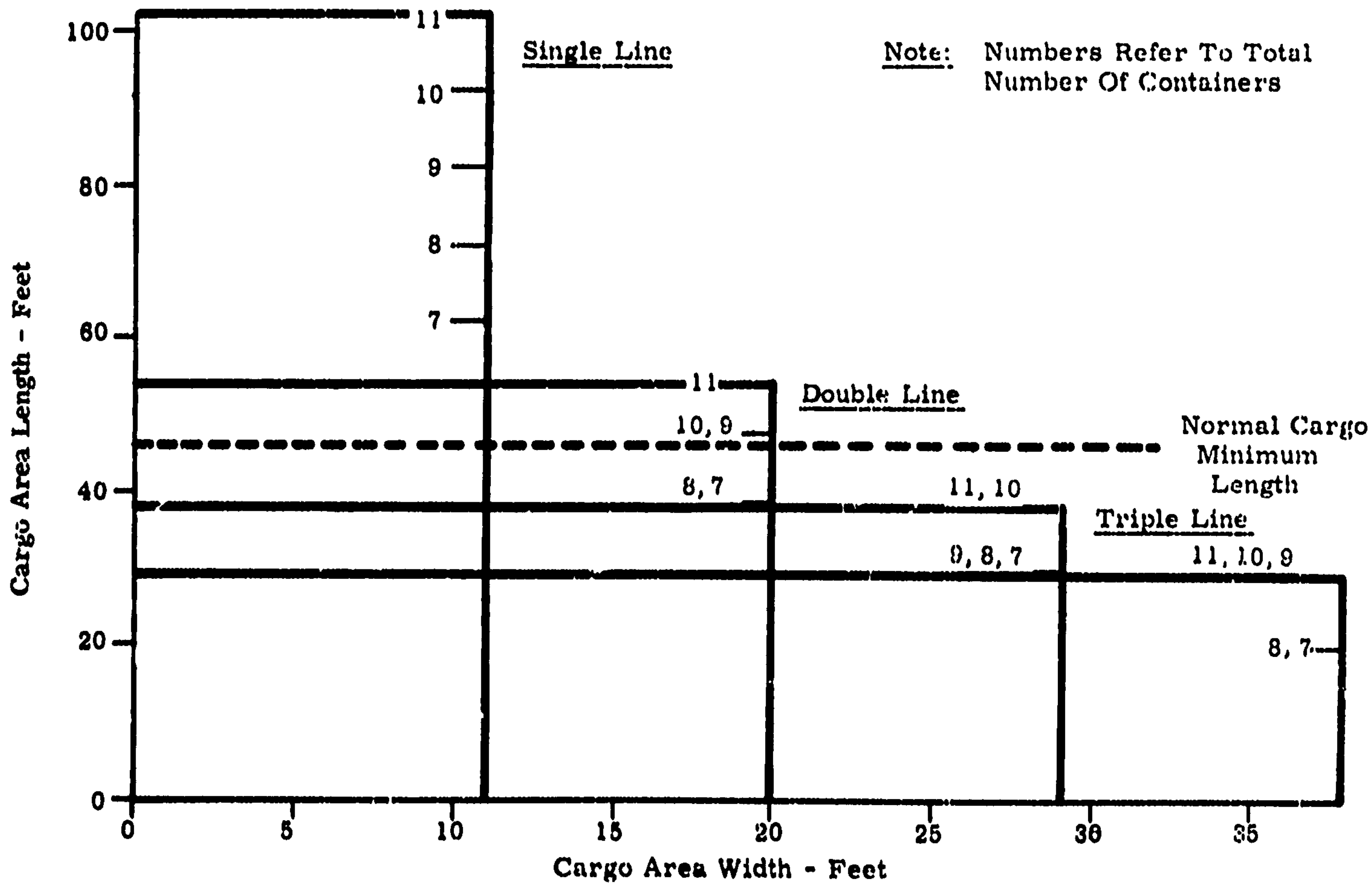


Fig.III-1. Containerized Cargo Distribution

$$LHP = 15.62\, p_c \sqrt{W_G}\ .$$

<u>Total Propulsion HP at 40 Knots (still air)</u>

With 45 per cent propulsion efficiency, at 2-1/2 feet operating height,

$$PHP = \left(\frac{258}{p_c} + \frac{163}{W_G^{\,1/2}} \right) W_G \quad ,$$

where W_G is Gross Weight in tons and p_c is cushion pressure in pounds per square foot.

<u>Structure Weight--W_S</u>

$$W_S = \left[20 \text{ x cushion area (sq. ft.) } . \right] \text{ lbs.}$$

therefore, $\dfrac{W_S}{W_G} = \dfrac{20}{p_c}$

<u>Power Plant Weight--W_{PP}</u>

W_{PP} = (1.25 pounds per installed HP)(installed HP)

$$\frac{W_{PP}}{W_G} = \frac{1.25 \times (LHP + PHP)}{W_G \times 2000}$$

<u>Equipment Weight--W_E</u>

Including crew, communications, furnishings, systems, emergency and auxiliary equipment,

$$W_E = .05\, W_G$$

<u>Vehicle Performance</u>

$$\text{Range} = \frac{375}{SFC} \cdot \left(\frac{L}{D} \right)_{eff} \log_e \frac{W_G}{W_G - W_F}$$

where SFC = .8 pounds per BHP per hour (lower figures are quoted by manufacturers, but since at this stage in the study it is not certain that the desired power will coincide with the optimum power of an available engine, a conservative SFC is chosen)

$$\left(\frac{L}{D}\right)_{eff} = \frac{2000\ W_G \times V\ kts}{\text{Total HP} \times 325} = 246\ \frac{W_G}{THP}$$

therefore

$$\text{Range NM} = 469 \times 246 \left(\frac{W_G}{THP}\right) \log_e \frac{1}{1-\frac{W_F}{W_G}}$$

Evaluation of these expressions results in the ralationships that are shown in Figures 3-2, 3-3, 3-4, and 3-5.

3.1.4.2.2.2 Cushion Area

With some of the vehicle dimensions determined, together with an approximate value for the vehicle gross weight, the layout can be further advanced by considerations of range and cushion pressure.

From the mission discussed at the beginning of this chapter, a range of 60 nautical miles (including an overland portion, which would be conducted at a lower altitude, particularly if prepared surfaces are available) would appear to be about adequate for the vehicle under consideration.

A range of 60 nautical miles could be achieved, depending on cushion pressure and weight carried, by a vehicle whose gross weight lies somewhere between 35 tons and 45 tons. A preliminary estimate of the width was 27 feet, and a minimum value of the length of the cargo deck was 46 feet. From previous studies, cushion pressure of 60 pounds per square foot seems to be near optimum for this type of GEM; with this cushion pressure, the selected range is achieved with a gross weight of 39 tons and a cushion area of 1.?90 square feet. This results in a cushion perimeter 27 feet by 48 feet (150 feet in length), which is close to that required by preliminary analysis.

3.1.4.2.2.3 Power Plants

From Figures 3-2, 3-3, 3-4, and 3-5 using the cushion pressure and gross weight determined, 5,850 horse power for lift and 1,220 horse power for

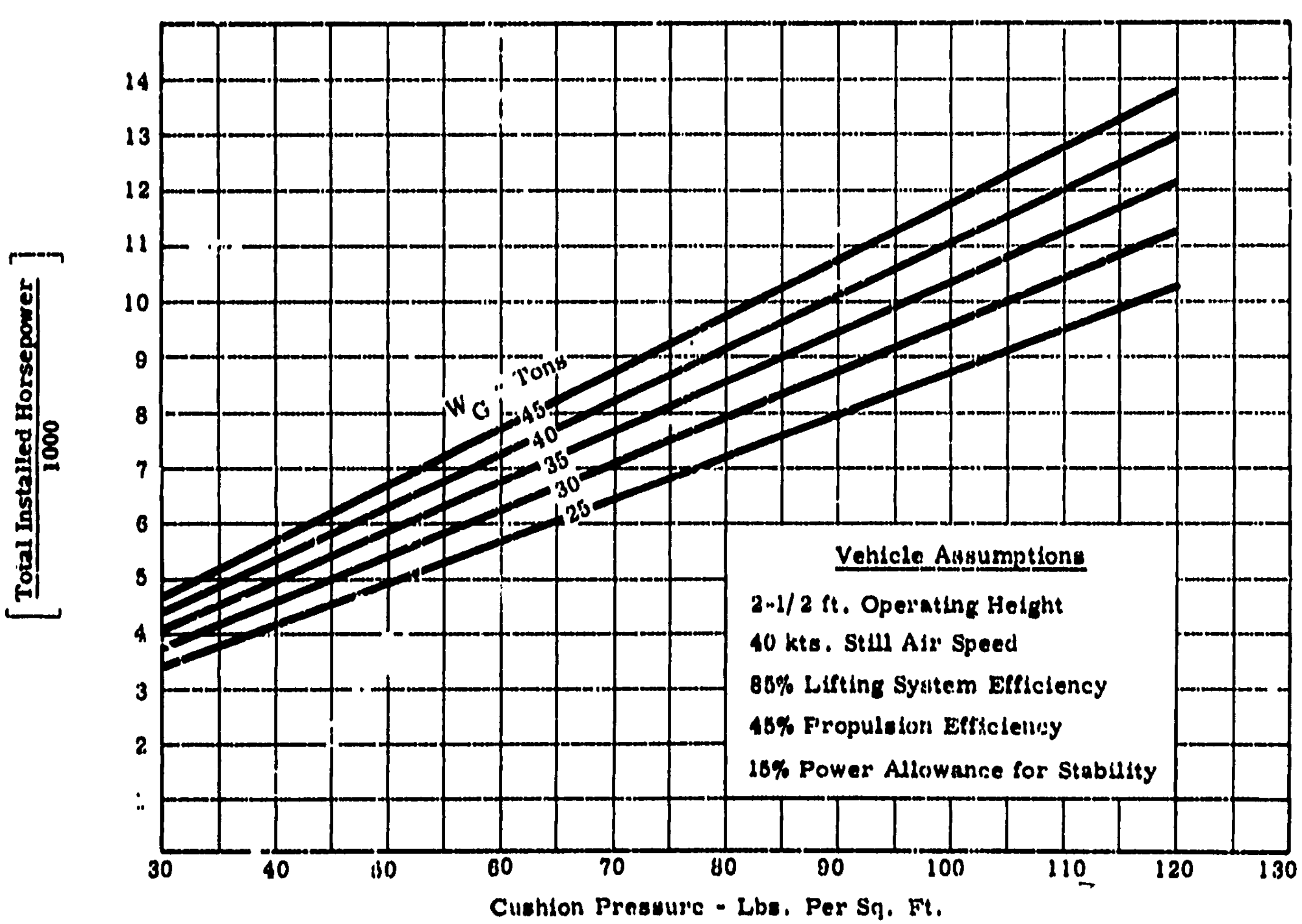


Fig.III-2. Installed Horsepower Vs. Cushion Pressure and Gross Weight

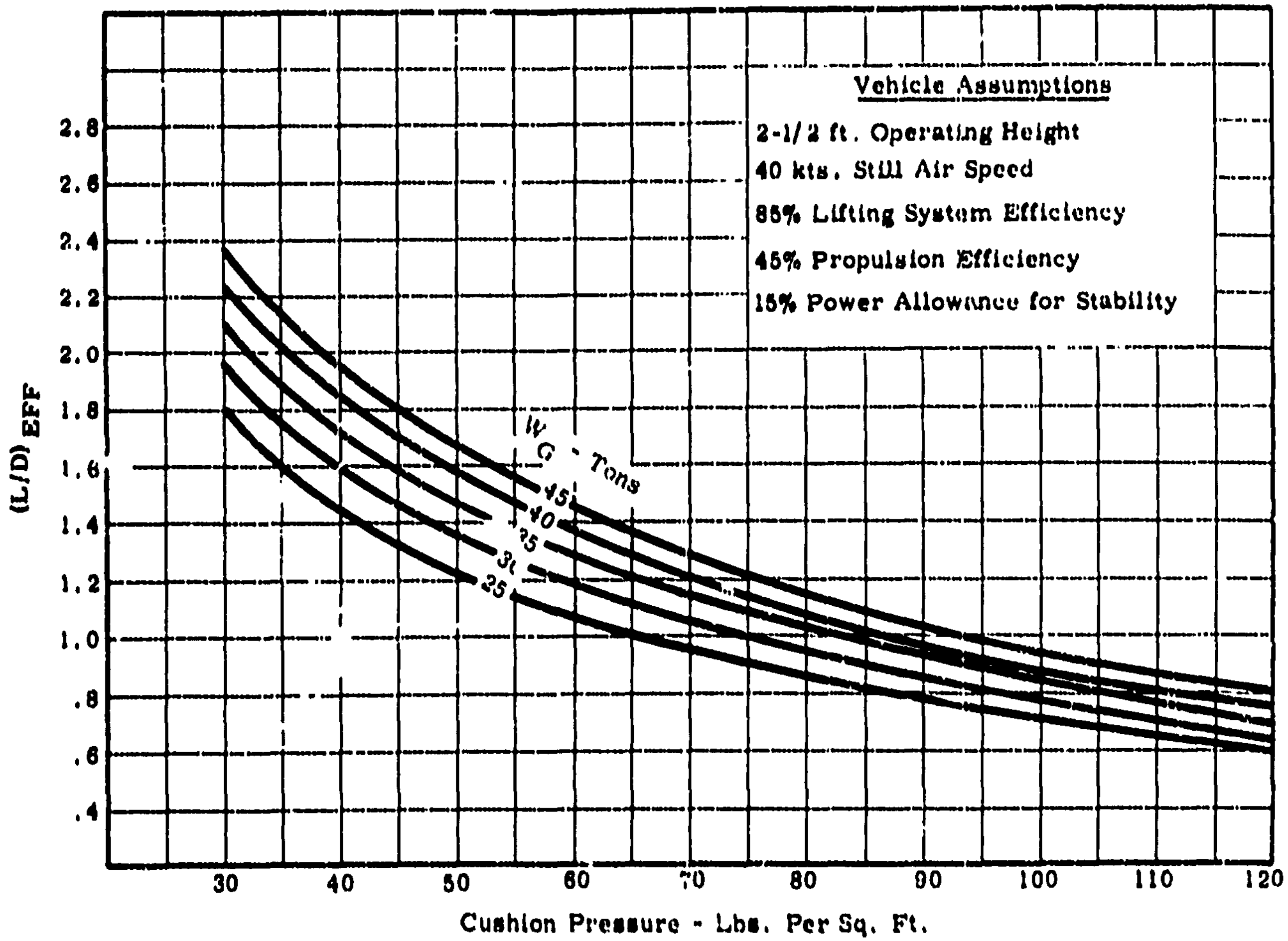


Fig.III-3. Effective Lift/Drag Ratio Vs. Cushion Pressure And Gross Weight

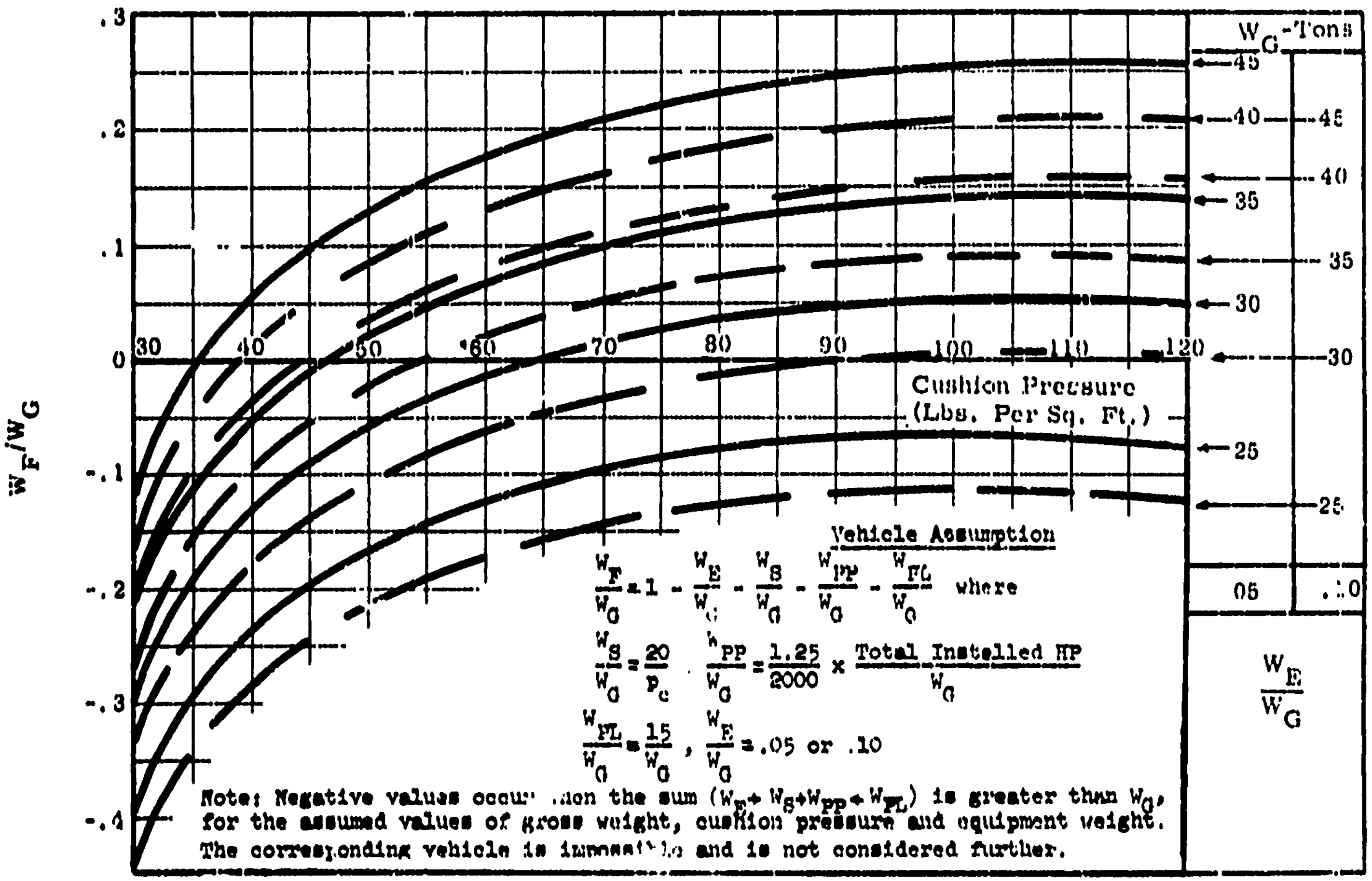


Fig.III-4. Fuel Weight Vs. Cushion Pressure And Gross Weight

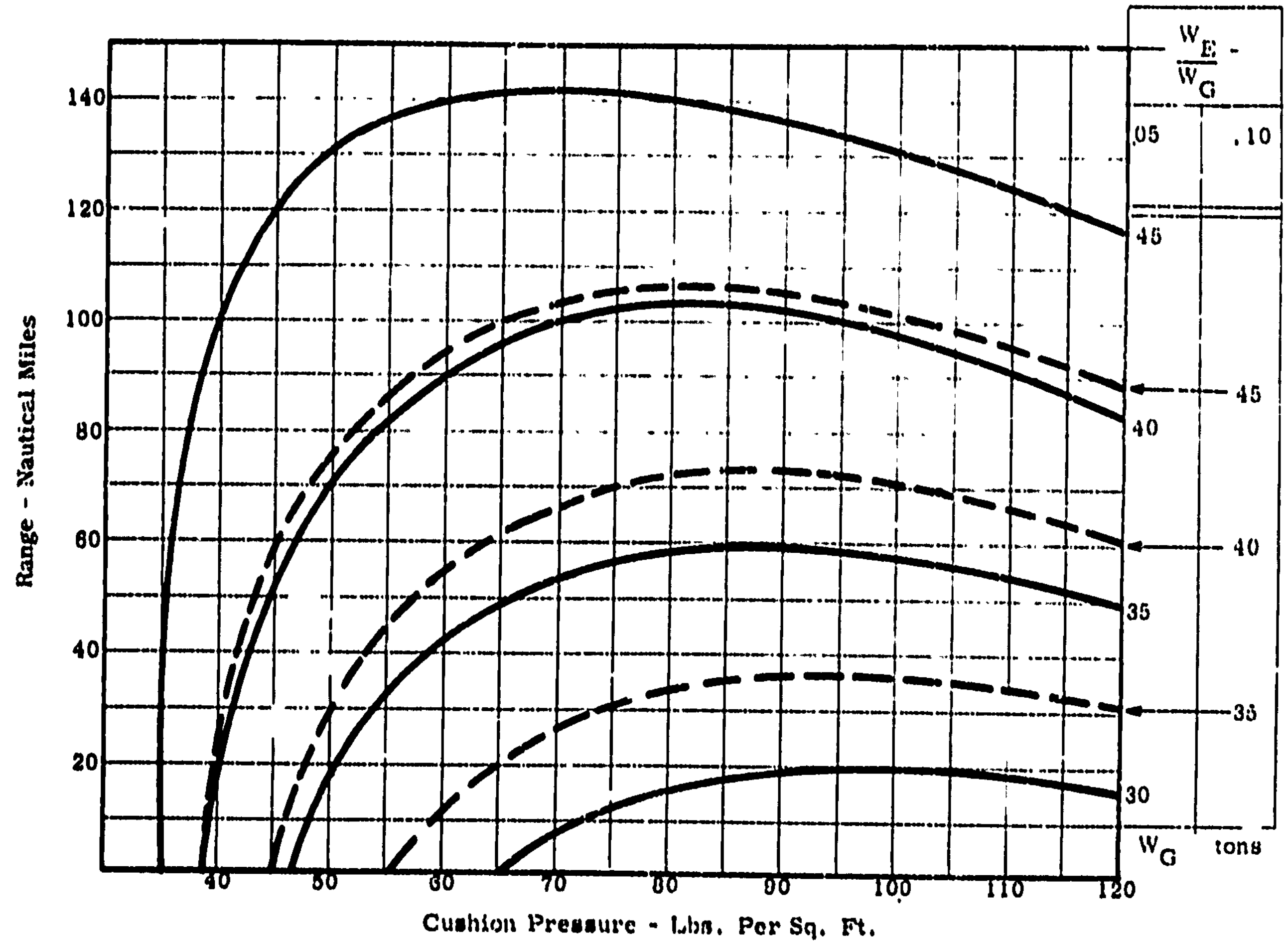


Fig. III-5. Range Vs. Cushion Pressure And Gross Weight propulsion will be required. It is considered that a satisfactory layout will utilize six lift engines and two propulsion engines.

In the overload condition, when the airjets and fans are operating "off design", the gross weight will be 78 tons, at a cushion pressure of 120 pounds per square foot. Under these conditions, for the same power supplied to the fans the operating height will be approximately six inches, assuming off-design fan operation and increased duct losses.

3.1.4.2.2.4 Fan Size

Fan size is determined by utilizing the optimization methods of reference 26; effects of ram recovery will be ignored. Using the previously determined values of

cushion area $S = 1,300$
cushion perimeter $C = 2(48 + 27) = 150$
operating height $h = 2.5$

$$\frac{Ch}{4S} = \frac{150 \times 2.5}{4 \times 1,300} = .0722$$

from reference 25, jet thickness = .25 h;

then $\frac{Ct_e}{4S} = .01805$.

For this value of $\frac{Ct_e}{4S}$, and $p_c = 60$, and augmentation efficiency $\eta_a =$ .8 and $S = 1,300$, the volume flow Q is obtained from

$$\frac{\left(\eta a\right)^{1/2}(Q/S)}{p_c^{\;1/2}} = 2.13,$$

i.e.,

$$Q = 24,600 \text{ cubic feet per second.}$$

Similarily the total head at the jet exit, H_e, is obtained from:

$$\frac{\eta a \left(H_e - p_o\right)}{p_c} = 1.51,$$

i.e.,

$$H_e - p_o = 113 \text{ pounds per square foot.}$$

The ideal horsepower, HPi, is obtained from:

$$\left(\eta_a\right)^{3/2} \left(\frac{HPi}{W}\right) \frac{103}{(p_c)^{1/2}} = 5.9,$$

i.e.,

$$HPi = 5,060.$$

The fan characteristic is given by

$$\frac{\sigma}{n} = 2.105 \left(\frac{Q}{N}\right)^{1/2} \left(\frac{H_e - p_o}{\rho}\right)^{-3/4}$$

therefore

$$\frac{\sigma}{n} = .04460.$$

For a range of ω (rotational speed in revolutions per second), the following total lift HP's are obtained:

ω	10	30	50	70
HP	5,650	5,940	6,150	6,320

Of the engines examined in reference 22, the "Gazelle Junior" comes closest to meeting this requirement, since it has an output speed of 50 revolutions per second and a continuous HP rating of 300. Hence, six "Gazelle Junior" engines are considered in this design. Although at their current state of development they would result in a slightly underpowered vehicle, it is considered that future development would make up this deficiency.

Having selected the revolutions per second at which the fan is to operate, the optimum fan diameter is selected from:

$$\frac{D_{opt}}{\left\{1 - \left(\frac{D_H}{D_{opt}}\right)^2\right\}^{3/7}} \cdot \left(\frac{30}{H_e - p_o}\right)^{1/2} = 2.7$$

where D_H = hub diameter = $1/3\ D_{opt}$

This results in D_{opt} = 5.5 feet for this vehicle. As the details of the fan design are not important to this study, they are not developed here.

3.1.4.2.5 Over-all Dimensions

Having decided on the cargo area dimensions, the cushion dimensions, the jet thickness and angle, the power plant requirements, and the fan size, the over-all vehicle layout is determinable. A preliminary sketch of the vehicle plan is shown in Figure 3-6.

3.1.4.3 Maneuvering Capability

3.1.4.3.1 Side Forces

All available side thrust on one side will provide the maximum maneuver g's in the lateral direction, neglecting aerodynamic side forces when the vehicle is sideslipping.

On the assumption that all modules will produce the same thrust, the front and rear modules may have different louver geometry when compared to the center modules.

$$\text{Total maneuver g's} = \frac{\text{total side thrust}}{W_G}$$

$$= \frac{3 \times T_L}{W_G}$$

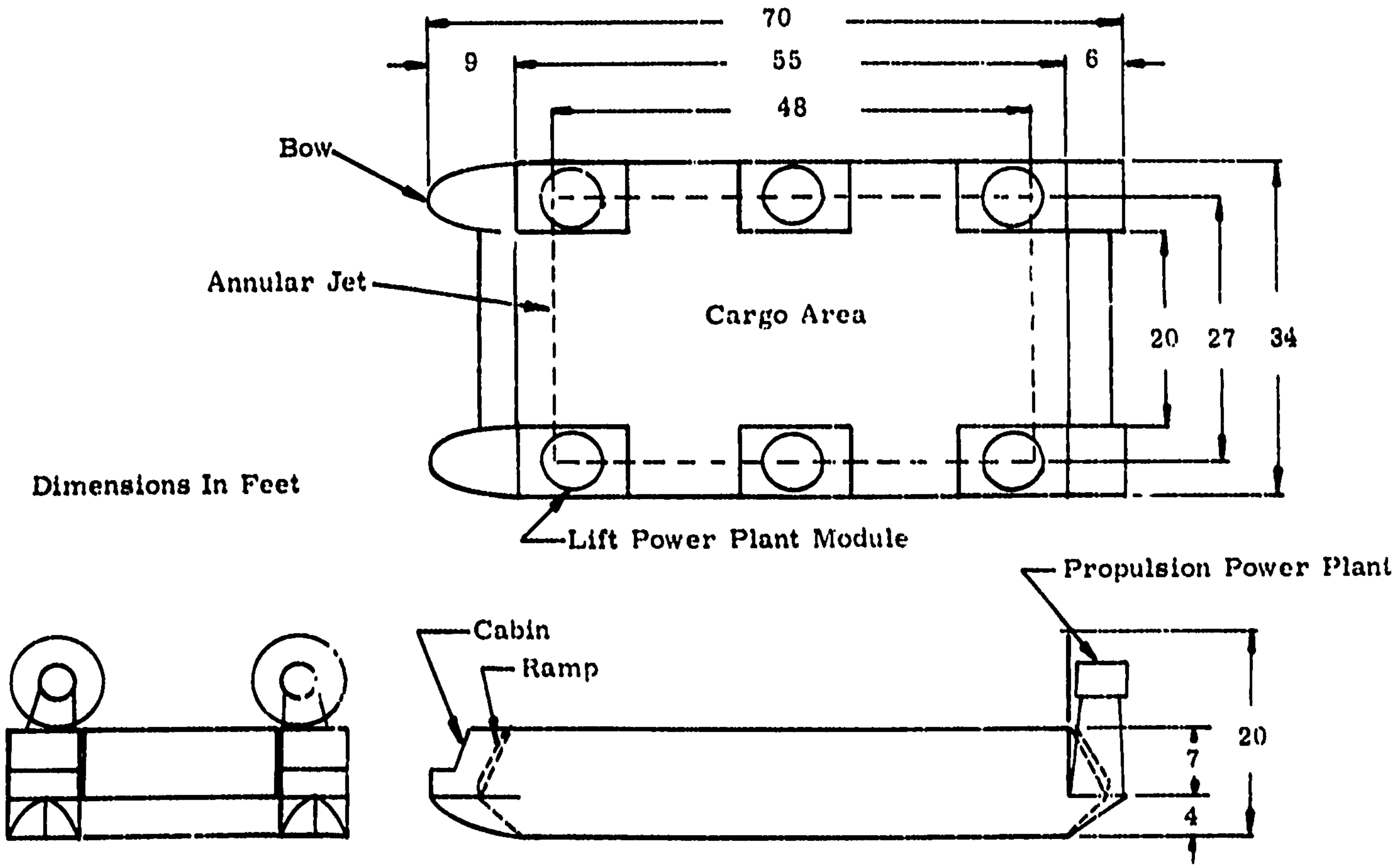


Fig.III-6. Vehicle Preliminary General Arrangement

Louver thrust,

$$T_L = \eta_L \ \rho \ Q_L \ V_L = \eta_L \ \rho \ Q_L \sqrt{\frac{H_L - p_o}{1/2\,\rho}}$$

$$= \eta_L \sqrt{2\rho(H_L - p_o)}$$

Louver area $\qquad A_L = Q_L \left[\frac{1/2\,\rho}{H_L - p_o}\right]$,

where $\quad Q_L$ = volume flow through louver

V_L = velocity of flow through louver

H_L = total head of flow through louver

η_L = louver efficiency factor.

$$T_L = \eta_L \ \rho \ \frac{(24,000 - Q_E)}{6} \sqrt{\frac{H_E - p_o + 1/2 \ \Delta H_D}{1/2 \ \rho}}$$

where Q_E and H_E are the values required to maintain the desired operating height, ΔH_D is the loss in total head in duct between the fan and the annular jet.

Therefore,

$$T_L = \eta_L \sqrt{\frac{2\rho}{6}} \left(24,000 - \frac{[V] \times 1300 \times (60)^{1/2}}{(.8)^{1/2}}\right) \sqrt{[H] \times \frac{60}{.8} + 1/2 \ \Delta H_{LDL}}$$

where V , H are the volume flow and total head parameters from reference 24.

Therefore, $T_L = \eta_L \ (2430 - 1140 \, [V]) \sqrt{[H]}$

and $A_L = \dfrac{15.68 - 7.34 \, [V]}{\sqrt{[H]}}$.

The results of this analysis are developed in Table 3-2, and show that this version will have poor lateral maneuverability by virtue of its low installed power.

3.1.4.3.2 Braking and Accelerating Forces

Maximum accelerating force will be at zero forward speed; with the contemplated propeller design this will be approximately .065 g's.

Maximum braking force will be at maximum forward speed of 40 knots and will be approximately .095 g's.

In actual practice it may be necessary, to supplement both of these capabilities by deflecting the thrust from the side louvers in a fore and aft direction. Average values over the speed range are:

.04 g acceleration
.06 g deceleration.

3.1.5 FINAL STUDY CONFIGURATION

The considerations outlined here and in reference 24 have led to the configuration shown in Figs. 3-7 and 3-8 consisting of a minimum performance vehicle suited to the basic LOTS mission, utilizing modular power plants and interchangeable units as far as possible. Flush intakes to the lift system power plants are utilized to facilitate loading and unloading while alongside a ship or dock in heavy weather. It is considered that the low-speed and short range requirements for the LOTS carrier make more sophisticated intakes unnecessary. The complexity of an integrated lift-propulsion system cannot be justified without considerable design refinement, so a simple separate propulsion system is chosen in which the thrust may be oriented as desired for firm and additional control.

An attempt has been made to provide maneuver control with the aid of side thrusting controllable louvers, but maneuvering can be conducted satisfactorily only at low speeds, or with the aid of the differentially controllable propulsion propellers. Hence, this vehicle is restricted to over-water missions in most cases.

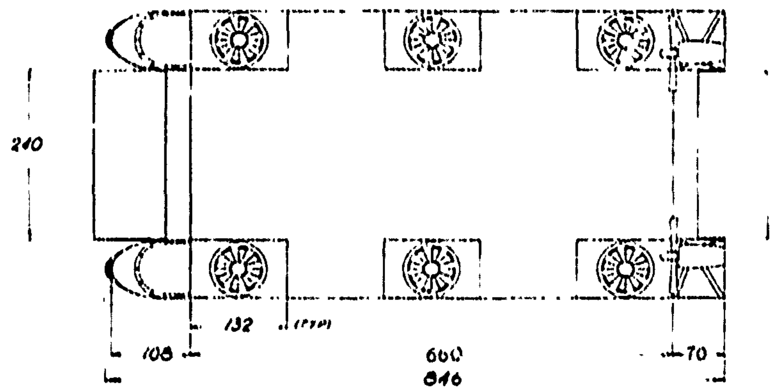
To offset the limited maneuvering capability of the vehicle, ramps have been provided at both ends for loading and unloading purposes. Twin bow cabins accommodate a crew with dual controls to facilitate close hauled

Table III-2
Maneuverability Analysis

$\frac{t_e}{h}$	.25	.35	.45	.55	∞
h	2.5	1.788	1.39	1.138	0
H	1.51	1.29	1.17	1.10	.96
$[V]$	2.13	1.75	1.55	1.30	0
2430-1140 $[V]$	0	430	660	950	2430
T_L/η_L	0	489	714	996	2380
15.68-7.34 $[V]$	0	2.83	4.28	[illegible].15	15.68
A_L sq. ft.	0	2.49	3.96	5.84	16
Total Sideforce lbs. (η_L = .8)	0	1170	1710	2390	5710
Lat. acc g's	0	.015	.022	.031	.073
Turning radius feet					
5 kts	∞	148	[illegible]	72	30
15 kts	∞	1330	905	642	271
25 kts	∞	3700	2520	1790	760
Percent installed LHP used for maneuver	0	28.6	44.5	54.5	100.

CHARACTERISTICS

LENGTH (RAMP RETRACTED)	70.5 FT
WIDTH	34.0 FT
HEIGHT	21.0 FT
WEIGHT EMPTY	40464 LB
A.U. WEIGHT	78044 LB
CUSHION AREA	1296 FT²
CUSHION PRESSURE	60 LB/FT²
SPEED (MAX)	40 MPH.
RANGE	60 M.
OPERATING HEIGHT	2.5 FT
POWER (LIFT)	5400 SHP
POWER (PROPULSION)	1750 SHP
CARGO (NORMAL)	15 TON
CARGO (OVERLOAD)	45 TON
CARGO AREA	1100 FT²



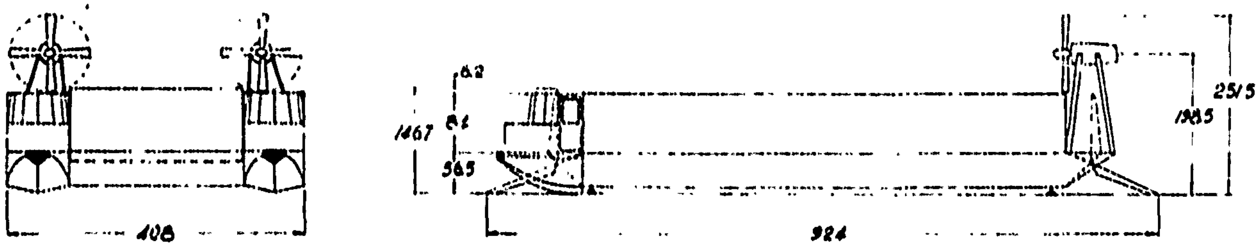


FIG III-7 GENERAL ARRANGEMENT OF VEHICLE

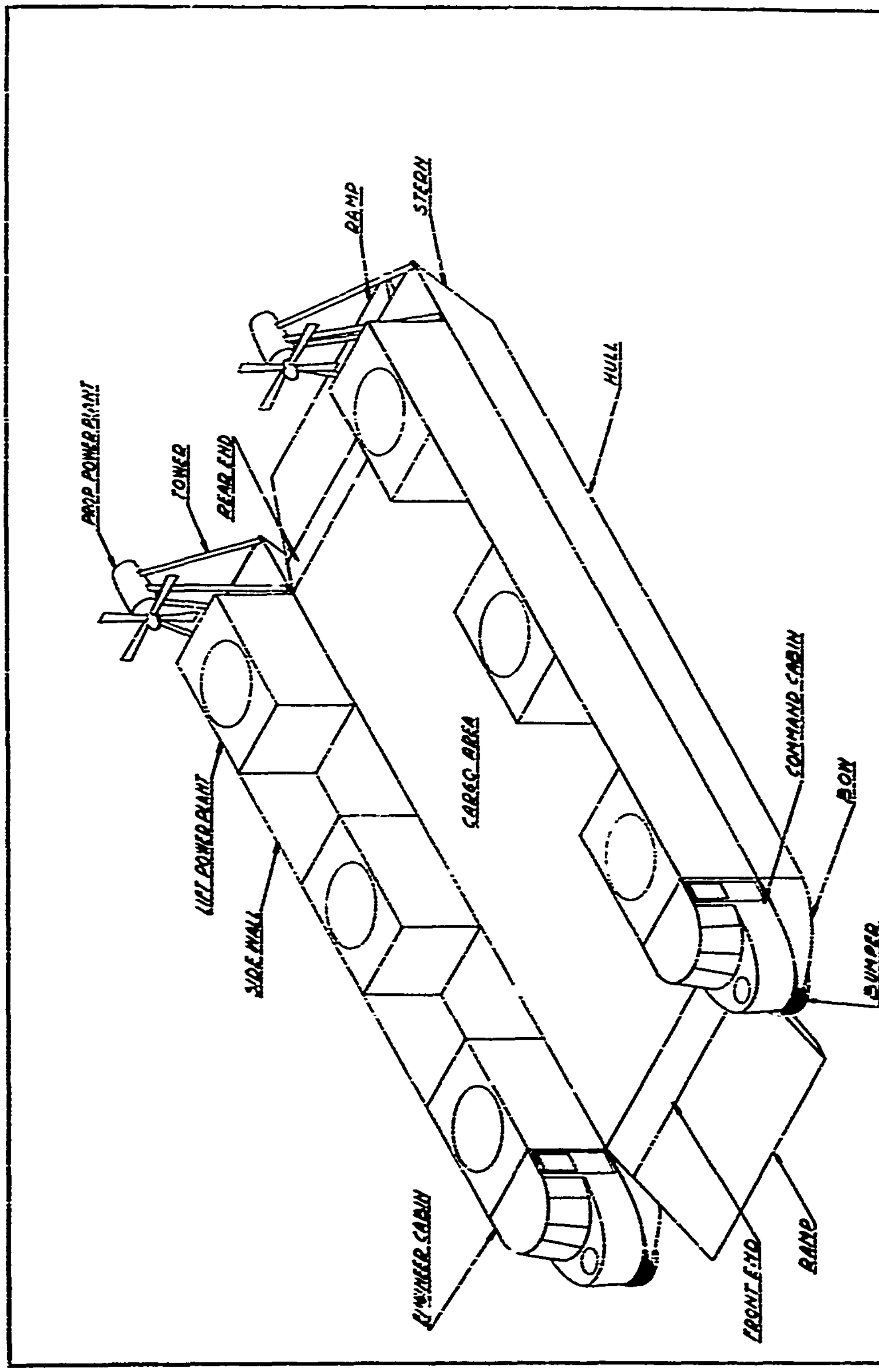


FIG III-8 ISOMETRIC VIEW OF VEHICLE

handling. Bow sections have been designed to absorb most of the shock of wave impact, and contain the necessary mooring gear. The stern sections contain auxiliary power units for water maneuverability.

The lift power modules, a typical section of which is shown in Figure 3-9 utilizes the simpiest possible "straight-through" flow system with engine cooling taking place automatically. The choice of the "Gazelle Junior" engine is ideally suited since this engine can be mounted vertically.

3.1.6 PRELIMINARY WEIGHT AND BALANCE ESTIMATE FOR INFLATABLE STRUCTURE

By definition it is assumed that certain prime components are attached as hardware to the inflated structure; they are:

1. All six lift modules;
2. The propulsion units;
3. The equipment and systems.

The remainder of the machine is constructed, in convenient sections, as a series of inflatable structures. The weights for the hardware items are known from the analysis in reference 23. An approximate estimation of the inflated structure weight follows.

3.1.6.1 Inflated Structure Weights for Stressing Purposes

3.1.6.1.1. Basic Assumptions for Preliminary Estimates

It will be assumed that, since an inflation pressure of 10 poundsper square inch for a 4-foot deep beam (40 feet long by 20 feet wide) requires a skin strength of approximately 220 pounds per inch, and the same beam loaded with a 50-pounds per square foot uniformly distributed load requires a skin strength of 30 pounds per inch (with a 3g load factor), then a skin strength of three (30 ÷ 220)--i.e., 750 to 800 pounds per inch--will be typical of the major part of the structure, allowing for adequate structural life.

The weight of this skin is in the region of 30 ounces per square yard, when fully coated and protected for air retention and environment. When no coatings or protection are necessary, as in internal webs and shape retention members, the weight reduces to 12 ounces per square yard. This gives approximately 0.2 pounds per square foot for the uncoated fabric.

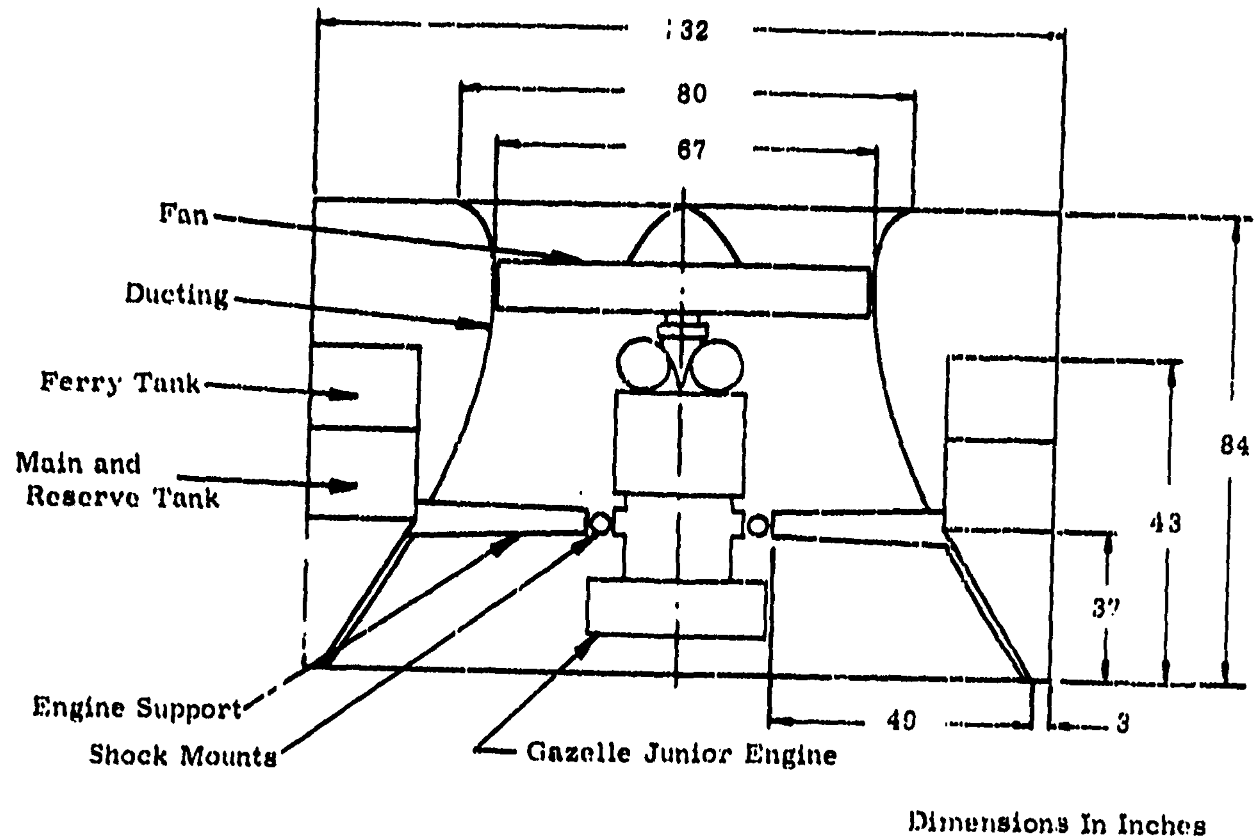


Fig. III-9. Lift Power Plant Module Cross-Section

3.1.6.1.2 The Air-Jet Sections

Assuming that the air-jet sections are made up of double-walled ducts, with webs in the walls to provide shape retention, the approximate coated fabric area is 50 square feet per foot of air-jet, and with one web every foot the uncoated area is 25 square feet. Total fabric weight per foot then is

$$(50 \times .2) + (25 \times .08) = 12 \text{ pounds per square foot of jet;}$$

allowing for additional cement and fastenings, this will result in an approximate weight of 25 pounds per foot of jet.

3.1.6.1.3 The Cargo Deck

With top and bottom skins at .2 pounds per square foot, the total skin weight is approximately two (820 x .2) = 328 pounds for a cargo deck of 20 feet by 41 feet.

With 4-foot deep webs at 1-foot intervals, the uncoated fabric weight will be 20 (42 x 4 x .08) = 280 pounds.

Total weight, including additional cement and fastenings, etc., = 1,200 pounds, or about one pound per square foot.

Since the cargo deck will be highly susceptible to deflections, more material will be required to maintain a reasonable stiffness; it is assumed that four pounds per square foot will be adequate for this purpose.

3.1.6.1.4 The Front Hulls

The hull bottoms must be designed to withstand the maximum impact pressures and be highly abrasion resistant. The surface area for each hull is approximately eight square yards.

The internal structure for each hull will use about 25 square yards of normal coated material, since it will be inflated to support the outer skin.

Over-all hull structure weight becomes

$$\frac{8 \times (30 + 70)}{16} + \frac{25 \times 30}{16} = 100 \text{ pounds approximately per hull,}$$

using a 70-ounce per square yard abrasion coat on the bottom skin.

3.1.6.1.5 Cockpit Weights

Total weight of everything other than structure is taken from reference 23 to be about 1,040 pounds, including crew, communication equipment, instruments, controls, furnishings, and glazing. In the inflatable version, the glazing will be replaced by a transparent flexible plastic of much lower weight. The additional structure weight will be approximately 100 pounds, so that total cockpit weight should be approximately 1,140 pounds per side.

3.1.6.1.6 The Propulsion Platforms

These can be adequately constructed of inflatable fabric of 800 pounds per inch strength, and require approximately 15 square yards of material, for a weight of 30 pounds each.

Hence, total propulsion platform weight, including propulsion units, fuel tanks and supports will be not more than 2,100 pounds each, again utilizing the weights from reference 23.

3.1.6.1.7 Forward and Rear Ramps

Using the same strength fabrics as the propulsion platforms, approximately 90 square yards each of coated material with 40 square yards of uncoated material is required for a total weight of approximately 250 pounds for each ramp.

3.1.6.1.8 Weight and Balance Summary

Utilizing the weight and balance of the rigid vehicle of reference 22, in combination with the inflatable structure weights estimated above, a summary is drawn up for the vehicle in Table 3-3, following this page.

Table III-3

Preliminary Weight and Balance for Inflatable Vehicle

		W	x	$Wx/10^6$	$Wx^2/10^9$
Cabin 1		803	+ 370	+ .336	.103
Bow 1		100			
Cabin 2		803	+ 370	+ .336	.103
Bow 2		100			
Front End		100	+ 346	+ .0346	.012
Ramp 1		250	+ 346	+ .0865	.030
			+ 264		.1395
			+ 132		.0349
Main Structure (equilavent to Cargo Deck and Air-jet sections)		10,000	0	0	0
			- 132		.0349
			- 264		.1395
Wall	1		+ 11		
	2	30		0	0
	3		- 11		
	4				
Power Plant (lift) Mod	1	1,850	+ 264		.258
	2	1,850			
	3	1,850	0	0	0
	4	1,850			
	5	1,850	- 264		.258
	6	1,850			
Rear End		100	- 346	- .0346	.012
Stern 1		30	- 353	- .0106	.0037
Stern 2		30	- 353	- .0106	.0037

Table III-3 (Contd.)

		W	x	$Wx/10^6$	$Wx^2/10^9$
Tower 1		307	- 365	- .1121	.041
Tower 2		307	- 365	- .1121	.041
Power Plant (prop)	1	1,035	- 365	- .3775	.138
	2	1,035	- 365	- .3775	.138
Ramp 2		250	- 346	- .0865	.030
GEM Empty		26,380		- 1.0004	1.5202
Weight Empty		26,380	- 125	- .3284	1.5202
Crew 1		340	+ 370	+ .126	.0466
Crew 2		340	+ 370	+ .126	.0466
Fuel	1				
	2				
	3	1,800	+ 264		.1255
	4				
	5				
	6				
	7	1,800	0	0	0
	8				
	9				
	10	1,800	- 264		.1255
	11				
	12				
	13	750	- 365	- .274	.100
	14	750	- 365	- .274	.100
Cargo (normal)		10,000	+ 108		.1166
		10,000	0		0
		10,000	- 108		.1166
Normal Useful Load		37,580		- .296	
Normal Gross Weight		63,960	- 9.76	- .6244	2.2976

K = 15.8 ft.

3.1.7 DESIGN LOADING CASES AND LOADS ANALYSIS

3.1.7.1 Basic Assumptions

Based on the vehicle characteristics, size, and weight distribution developed, the structural design of the vehicle can be determined. In this respect the work of reference 24, which has reviewed a wide variety of loading cases on projected and existing GEMs, is used to develop major loading cases for the purposes of this study. The most severe cases may not have bee covered, but the character of the inflatable structure can be establishec quite adequately without recourse to a detailed analysis, such as would be required in any thorough examination for a specific vehicle.

3.1.7.2 Loading Cases

The loads imposed on GEM structures as a result of these loading cases are examined and the greatest values utilized in further analysis of inflatable structures for a range of GEM sizes.

The loading cases examined, together with the magnitudes of the maximum bending moments, shear forces, and torsions determined from them, are quoted in the following summary. The unit bending moment and shear force diagrams with which they are associated are shown in Figs. 3-10 and 3-11.

3.1.7.2.1 Summary of Un-Factored Design Cases For Main Structure

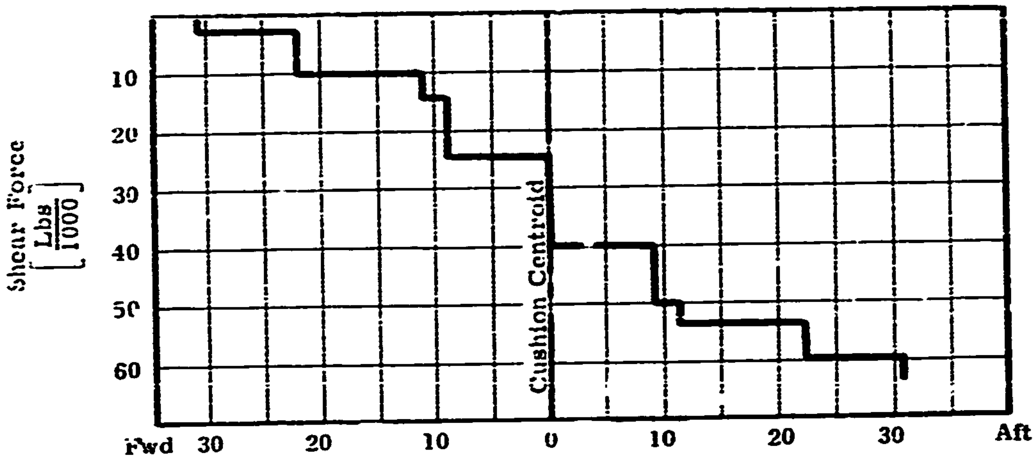
3.1.7.2.1.1 Bending--Longitudinal

8g vertical acceleration at the bow;
8g vertical impact at opposite corners of the cushion.

The worst case of these two is the 8g landing case reacted at opposite corners of the cushion. Maximum bending moment = 4,240,000 pounds feet; with impact points located approximately 28 feet forward and aft of the cushion centroid.

3.1.7.2.1.2 Bending--Lateral

8g impact at opposite corners of the cushion gives rise to
Maximum positive bending moment = 560,000 pounds feet
Maximum negative bending moment = 360,000 pounds feet



NOTE: These distributions do not represent equilibrium conditions since no reactions are included, thus enabling bending moments to be minimized by appropriate choice of reaction location.

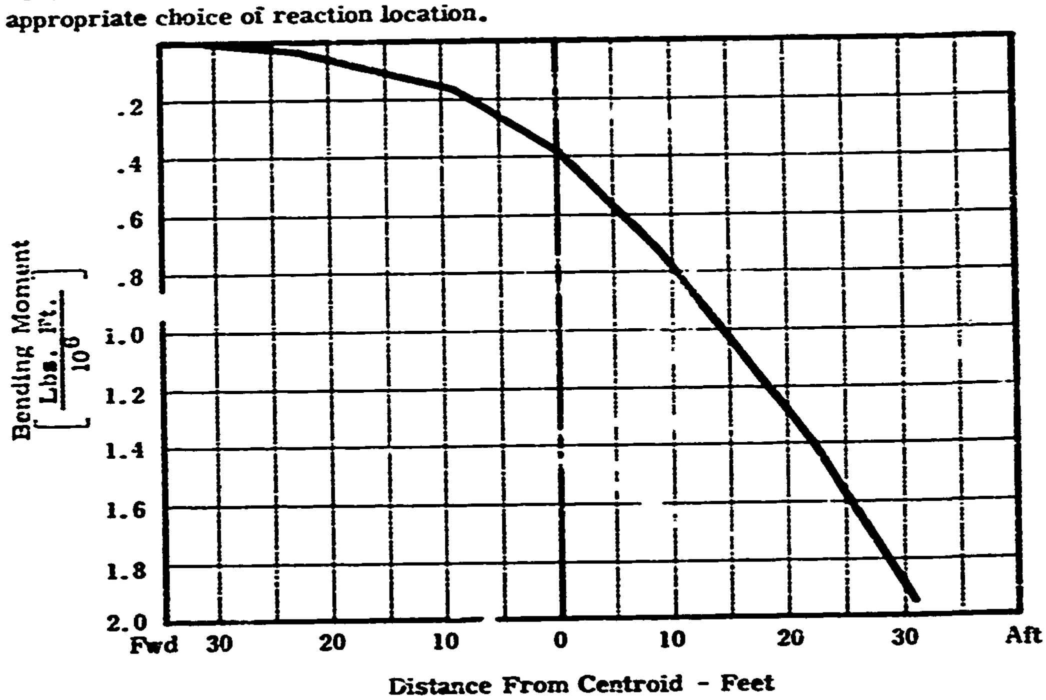
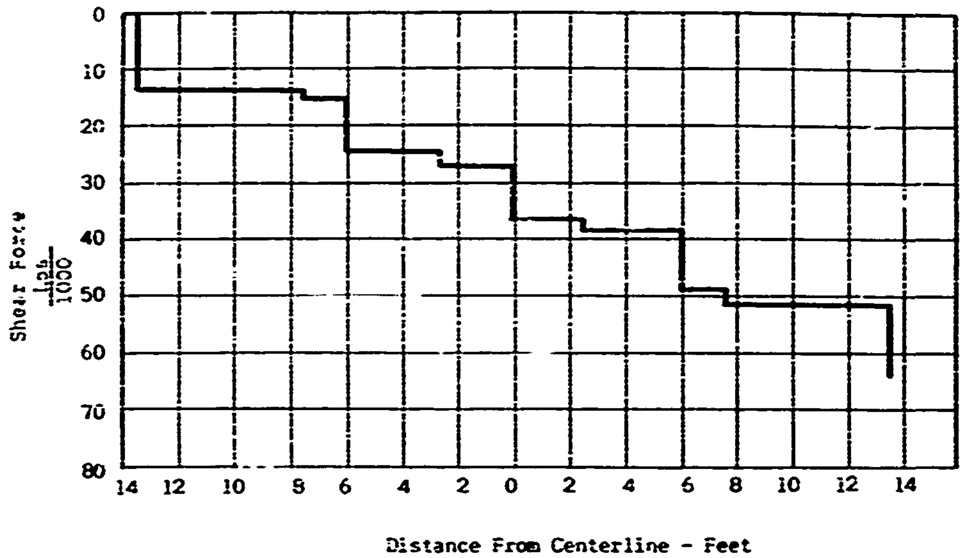


Fig.III-10. Longitudinal Shear Force And Bending Moment Distributions Due To 1G Vertical Acceleration (No Reactions)

NOTE: These distributions do not represent equilibrium conditions since no reactions are included, thus enabling bending moments to be minimized by appropriate choice of reaction location.



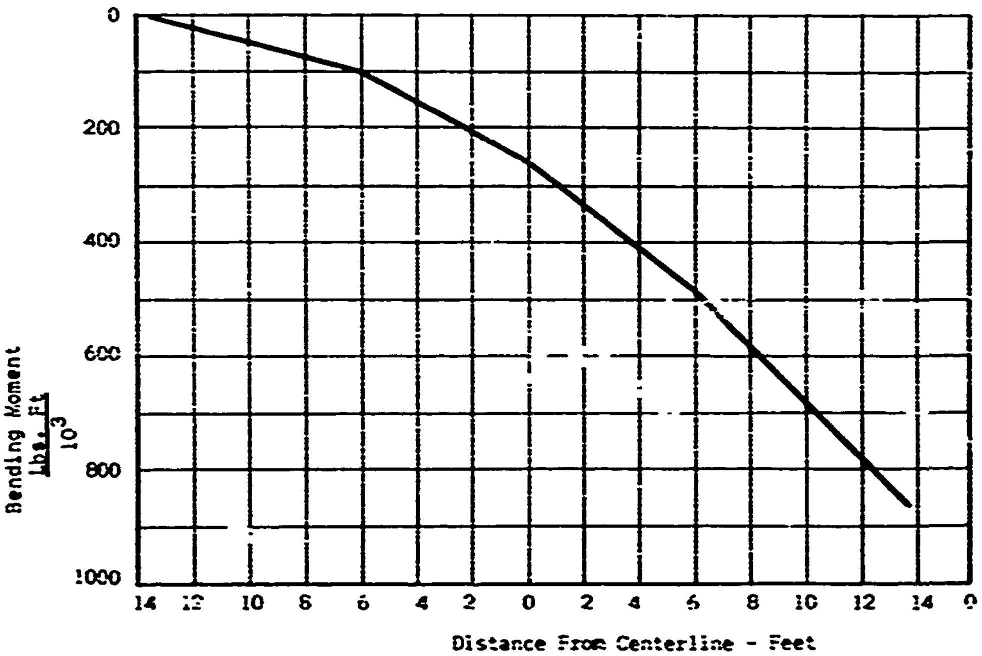


Fig. III-11. Lateral Shear Force And Bending Moment Distributions Due To 1G Vertical Acceleration (No Reactions)

Note: the negative case arises because the lift system modules extend beyond the cushion area, causing negative bending moments.

3.1.7.2.1.3 Torsion

Lateral and longitudinal torsion cases are determined from appropriate components of the diagonal torsion arising from the 8g landing case reacted at diagonally opposite corners, and are concurrent with the above bending cases.

Torsion component about longitudinal axis = $\pm$ 5,120,000 lbs. feet

Torsion component about lateral axis = $\pm$ 14,400,000 lbs. feet

3.1.7.2.1.4 Buckling

Arising from vehicle weight applied uniformly over the front or the side of the vehicle:

Longitudinal buckling loads	=	64,000 lbs. at ends of vehicle
		40,750 lbs. at center of vehicle
Lateral buckling loads	=	64,000 lbs. at sides of vehicle
		37,000 lbs. at center of vehicle

3.1.7.2.2 Application of Design Loads to Strength and Vehicle Life Estimation

The primary structure must be capable of withstanding the factored design loads without permanent deformation. In other words, the factored design loads must not stress the primary structure beyond the elastic limit.

A factor of 1.5 will be utilized on the design loads to allow for inaccuracies in estimation.

To provide the required vehicle life, a factor is applied to the material strength utilized in strength estimations. This factor is determined by the estimated utilization of any one vehicle. Due to the potential usefulness of inflatable GEMs, it is expected that they will incur much greater year-round utilization than will rigid construction machines. Utilization for an inflatable machine might be 3,000 hours per year, with a desirable service life of three years, giving a total life of 9,000 hours or approximately one year. For Dacron or nylon materials, the design material strength is taken as one-fourth of the quick-break strength for inflation stresses, and one-third of the quick break strength for design load stresses applied to the inflated structure.

3.1.8 EVALUATION OF INFLATABLE STRUCTURE TO MEET DESIGN LOADING CONDITIONS

3.1.8.1 Approach

The applicability of inflatable structures in the primary structure role can be reasonably evaluated by utilizing the critical loading conditions to determine the necessary inflation pressures, geometry, and material size for such loading conditions.

It is assumed that for this structure, wrinkling and collapse occur very close together; therefore, the critical loading conditions are utilized to determine the inflation pressure necessary to prevent wrinkling at these loads.

From Chapter II, one value for the inflation pressure is given by the solution of

$$p = \frac{16\, M_w}{\pi h^3} \left[\frac{1 + \frac{2b}{\pi h}}{\left(1 + \frac{4b}{\pi h}\right)^2} \right]$$

where M_w is the maximum bending moment, and torsion is negligible.

Hence, for a given M_w and $\frac{b}{h}$, p can be found as a function of h.

3.1.8.2 Variation of Inflation Pressure with Depth of Structure

The pure bending case is used here for simplicity because it provides conservative answers, and the complexity involved in an analysis of all cases where shear and bending, and possibly even torsion, occur together puts such an analysis beyond the scope of this study. The purposes of the study are considered met with the simpler expression.

For the vehicle under examination:

$$M_w = 6,360,000 \text{ pounds feet - factored}$$

$$(b + h) = 21 \text{ feet}$$

$$\left(\frac{b}{h}\right) = \left(\frac{21}{h} - 1\right)$$

$$p = \frac{32.4 \times 10^6}{h^3} \left[\frac{1 + .636\left(\frac{21}{h} - 1\right)}{\left(1 + 1.272\left(\frac{21}{h} - 1\right)\right)^2} \right]$$

Figure 3-12 is the solution to this expression. It shows that this vehicle cannot be built to withstand the design load with an inflation pressure of less than about 25 pounds per square inch. To be able to utilize seven pounds per square inch, the loading would have to be reduced to about 1,820,000 pounds per foot, factored.

For a vehicle with a 4-foot beam depth, the required inflation pressure is about 320 pounds per square inch; if the vehicle is required to react the applied load on opposite corners, the torsion introduced increases the necessary inflation pressure to about 440 pounds per square inch, an increase of 37 per cent.

In order to accommodate both seven pounds per square inch and a 4-foot depth, the applied loading would have to be reduced to 139,000 pounds per foot without torsion, or to 100,000 pounds per foot with torsion. This is equivalent to a 97.8 to 98.5 per cent reduction in design load applied to the inflatable structure.

This indicates that the majority of the vehicle structure under consideration must be of rigid construction, the remainder being constructed of inflatable material. <u>It is therefore clear that the predominant use of inflatable construction for the primary structure of vehicles of this configuration and size is not feasible until it has been demonstrated that high inflation pressures</u>

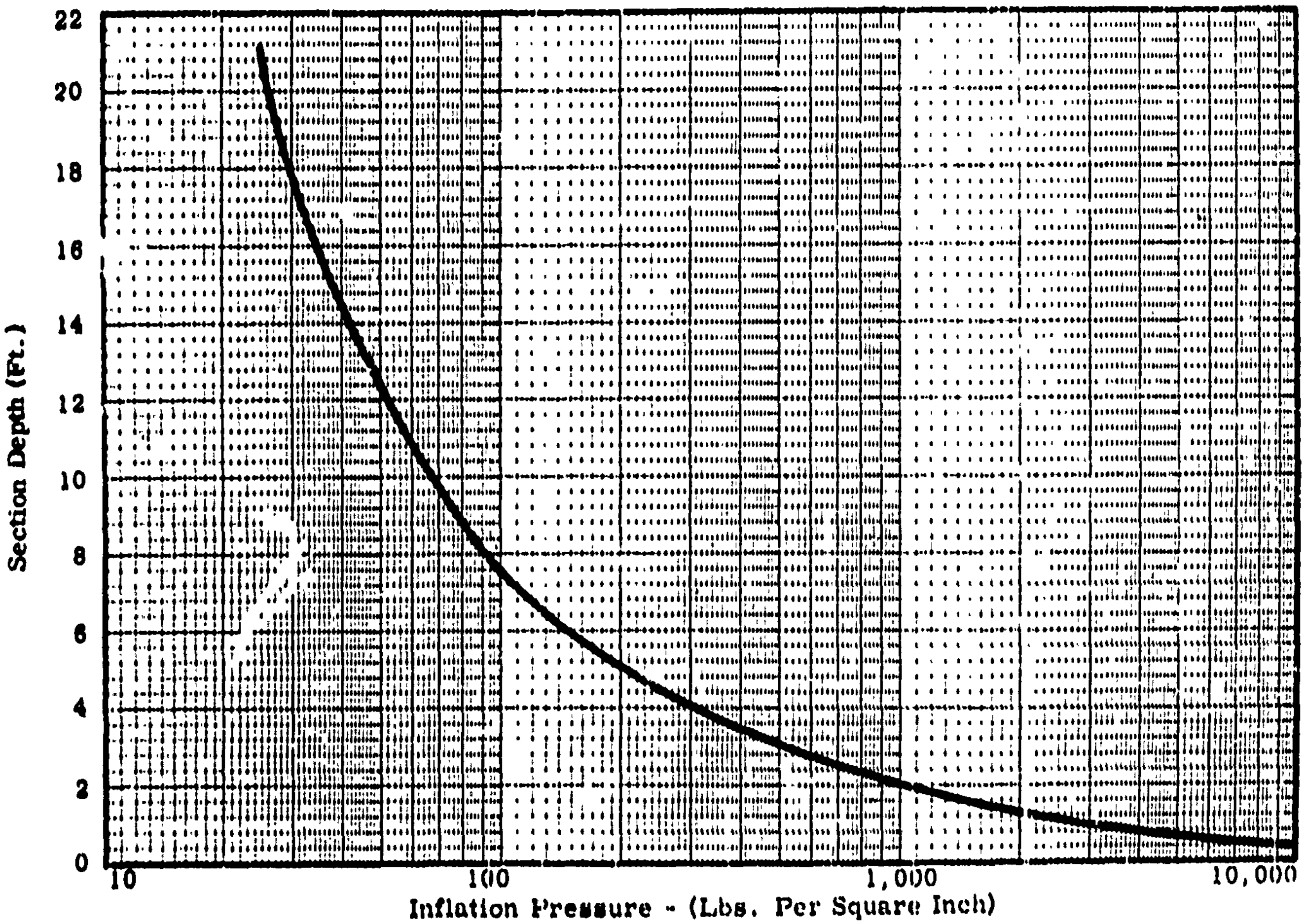


Fig.III-12. Characteristic Of Inflatable Structure To Meet The Design Load can be safely utilized--and then other factors need to be fully evaluated before this application is practicable.

To demonstrate the kind of loads that are feasible or nearly so for different configurations and sizes, Figure 3-13 presents the parameter $\frac{M}{ph^3}$ plotted against a section thickness parameter. For a typical current range of inflation pressures, 1,000 to 2,000 pounds per square feet, $\frac{M}{h^3}$ increases from a minimum of 200 to 400 for a circular section, and to 5,000 to 10,000 for a section having geometric characteristics similar to an aerofoil section. Hence, where depth is a constraint, $\frac{M}{h^3}$ of 5,000 to 10,000 is probably typical; but where weight is limiting, 500 to 1,000 is probably a better compromise. Note that in inflatable aircraft these two construction requirements conflict, since the low thickness/chord ratios inherent in aerofoil design result in low strength/weight ratios.

3.1.9 APPLICATION OF INFLATABLE STRUCTURES TO OTHER STRUCTURE COMPONENTS

A study of Figure 3-14, showing the change in the design bending moment with section size and configuration for the simple beam section under consideration, permits assessment of the utility of these structures in other areas.

The total GEM structure can be subdivided into typical areas of structural importance, as follows:

Primary Structure, including

- Cargo Deck
- Air-jet Assembly
- Power Module Assemblies
- Ramps
- Propulsion Assemblies

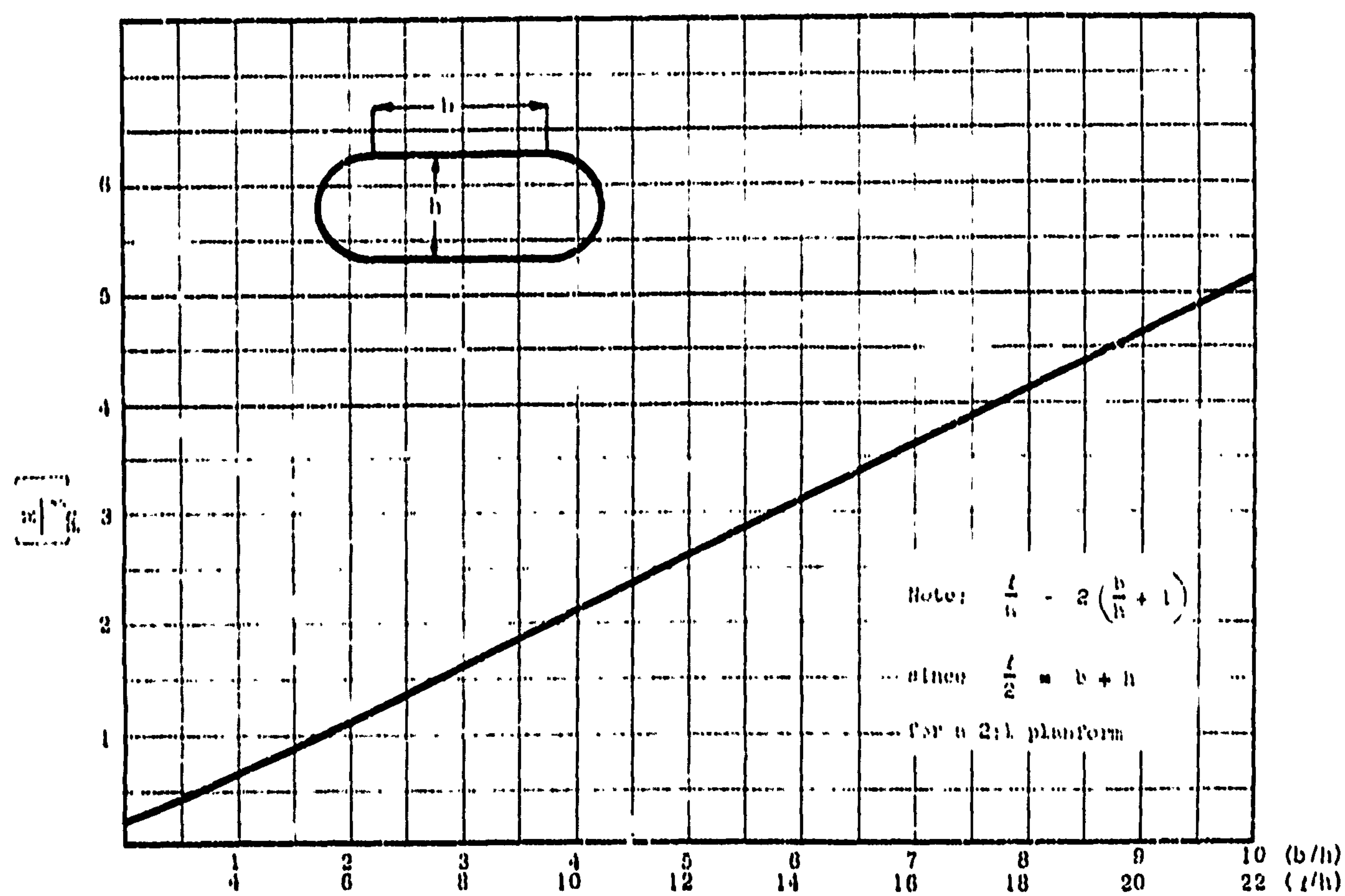


Fig. III-13. Requirements For An Unwrinkled Section In Pure Bending

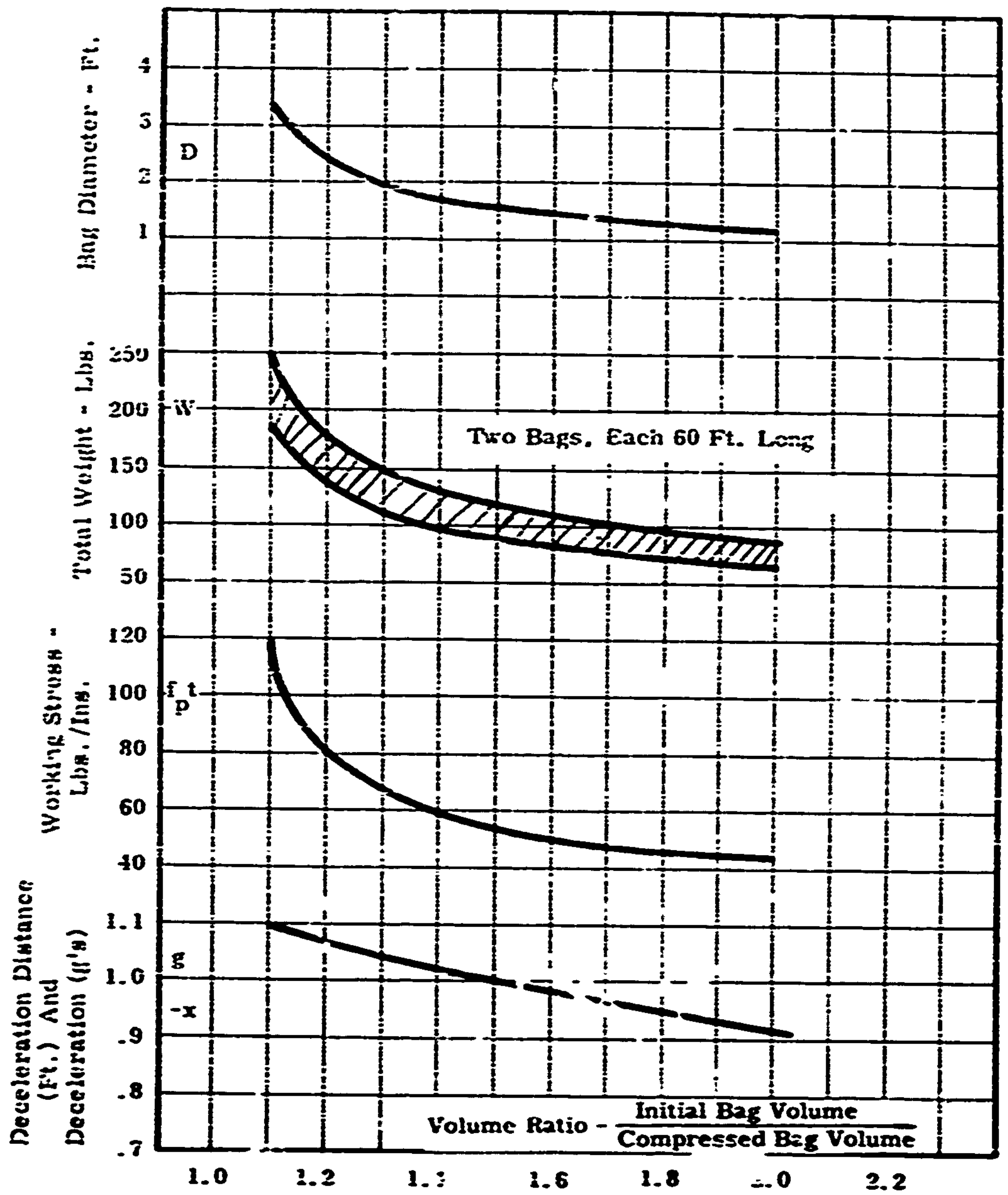


Fig.III-14. Inflatable Fender Parameters

Secondary Structure, including

- Cargo Superstructure
- Cabins
- Personnel Seats and Benches
- Aerodynamic Surface

Auxiliary Structures, including

- Fenders
- Landing Gear
- Flotation Units
- Intakes
- Skirts or Trunks

Each of the above areas can be considered as inflatable structures.

In the Primary Structure area, the previous analysis has indicated that, in this size vehicle, inflatable structures are not at present applicable.

In the Secondary Structure area, it may be that the lower loads required will permit inflatable structures. A probable application, in which several functions may be combined, is in the construction of the peripheral jet ducting and extended skirts or ducts. These units could be so constructed as to provide adequate shock absorbing qualities, laterally and vertically, in the event of impact with the surface or other objects. In so doing, the severity of the design loading conditions can be reduced and some design refinements considered as a consequence.

In the Auxiliary Structure area, many possible applications come to mind. The main characteristics of auxiliary structures are that they are not organic to the operation of the craft and are essentially detachable items. Any auxiliary structure that can be added to the craft and that will enhance its performance is worthy of consideration.

Some of the items cited earlier fall into this category, although just how good an exchange can be made will depend on the particular design. Inflatable structure is the natural choice for:

Fenders, which can pay for their weight in the reduced vehicle stressing requirements and better shipside and dockside handling capabilities.

Landing bags, which can claim the same trade-off, coupled with lower ground pressures in the power off condition.

Skirts or trunks, extensions to the annular jet, which can provide an effective operating height at much lower installed power than a conventional GEM.

Intakes, which can provide efficient inlet conditions for the lift fan system at forward speeds, and yet can be retracted to a safe stowage in a minimum of space when loading or unloading cargo.

Aerodynamic surfaces, which provide directional stability at high speed but are unwieldy, ineffective, and in the way in the hover or low-speed regime.

Collapsible seating accommodation for transporting personnel over long distances.

Environmental protection for operation in severe weather conditions, utilizing inflatable walls or an inflatable roof over the required area; this can be configured to suit any desired streamline form.

As an example of the type of auxiliary structure that could be used, the following data have been developed for a bumper along each side of the LOTS carrier considered in this study. It is considered that the energy that would be present at impact between the GEM and a large cargo ship, if the GEM were on the water in a Sea State 4, would be on the order of 500,000 feet pounds, and that this energy must be absorbed by the bags. Further, currently acceptable initial inflation pressures have been utilized and perfect damping assumed.

The independent variable chosen for this analysis is the volume ratio between the initial air bag volume and the volume when the GEM has been brought to rest. The following air bag parameters are shown in Fig. 3-15.

The bag diameter, D feet.
Deceleration distance, x feet.
Deceleration, g's.
Maximum principal stress in bag, pounds/inch.
Approximate weight for two bags, one each side, for the full length of the vessel.

3.1.10 Discussion of Inflation Systems for Each Application

The range of available and usable systems for inflation has been discussed in Chapter I, indicating the current trends in weight, capacity, and power requirements for desirable inflation pressures. The choice of a particular system for a specific application is best made during the preliminary design stage for that application. In this study, it is sufficient that all inflated structures are designed for a specific purpose, and that means for maintaining the pressure must be included in the design even though the structure may be auxiliary rather than primary.

The data in Chapter I show that a wide range of air supply requirements can be met at the present time with acceptable weight penalties, by small engine-driven pumps. The use of gas turbine bleed air may be feasible, but is probably only suitable for large vehicles, where the cost of cooling the bleed air can be offset in deicing equipment, or cabin heating equipment that utilizes the heat extracted.

Where there is a mix between primary, secondary, and auxiliary structures, it will undoubtedly prove efficient and feasible, in some cases, to utilize higher inflation pressures for some secondary and auxiliary structures than are used in the primary structure. This might be possible in high performance vehicles that require thin aerodynamic surfaces, the design of which could be most efficiently conducted at higher than usual pressures.

Under these circumstances, a system of relief valves in the inflation set-up would provide the necessary control, together with cutoff valves where needed.

CHAPTER IV

RESULTS, DISCUSSIONS, AND SECONDARY APPLICATIONS

4.1 EFFECT OF GEM SIZE ON THE UTILITY OF INFLATABLE PRIMARY STRUCTURE

Since some doubt was raised in the previous chapter concerning the suitability of inflatable structure for primary structures in GEMs, this section attempts to delineate the effect of GEM size (taking typical GEM characteristics) on the utility of such structure, at the same time indicating the effect of changing some of the primary parameters of a GEM.

It has been necessary to choose a single loading condition, one that was found severe in the previous work, and apply it throughout the range for a typical practical developed GEM. The result is qualitative rather than quantitative; the trends indicated are a rational guide to the effect of GEM size on primary inflatable structures.

4.1.1 VEHICLE CRITICAL LOADINGS AS A FUNCTION OF VEHICLE TYPE

From the analysis of the 15-ton vehicle, the critical bending loading occurs on vertical impact with the surface, when the support or contact points are taken to be near the ends of the vehicle, and no cushion pressure exists. This critical bending moment is a function of the mass distribution of the vehicle, which in turn depends on the power requirements and operating conditions.

A generalized analysis is made of the magnitude of the critical bending moment as a function of the following vehicle parameters:

- Cushion pressure
- Operating height
- Forward speed
- Gross weight

To simplify the computational problem a typical utility vehicle of good design, having the following characteristics, is considered:

- Structure weight - 15 pounds per square foot of cushion area
- Power plant weight - 1.0 pounds per installed HP
- Cushion area proportions - 2.1
- Over-all lift system efficienty - .60
- Propulsion system efficiency - .45 .65 .78 at V feet per second - 50 100 150
- Drag coefficient - .09 based on cushion area
- Equipment weight - 5 per cent of gross weight.

Within these limitations, the following expressions are used.

4.1.1.1 Lift Horse Power, Including Stability Allowance

$$\frac{LHP}{W_G} = .191 \left(1 + 1.5\left(\frac{h}{D_e}\right)\right)\left(\frac{h}{D_e}\right) \times p_c^{\;1/2} \sim \frac{HP}{lb.}$$

4.1.1.2 Propulsion HP

$$\frac{PHP}{W_G} = \left(\frac{.195}{10^6} \times \frac{V^3}{\eta_p} \times \frac{1}{p_c}\right) + \left(\frac{1.63}{10^4} \times \left(\frac{h}{D_e}\right) \times \frac{V^2}{\eta_p} \times \frac{1}{(p_c)^{1/2}}\right) \sim \frac{HP}{lb.}$$

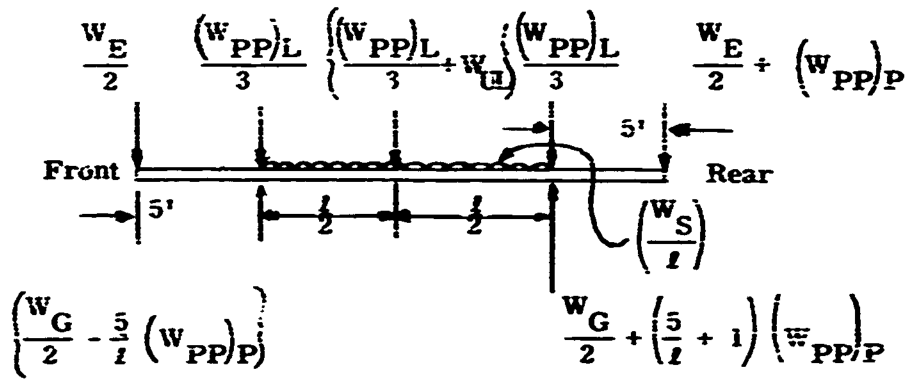
4.1.1.3 The Vehicle Loading is developed as follows:

The vehicle size is determined, assuming a 2:1 cushion planform.

Hence, cushion length, $\ell = \sqrt{\frac{2W_G}{p_c}}$

It is assumed that:

- The structure weight is uniformly distributed over length ℓ .
- The lift power plant weight is divided into three equal parts, located at the center and either end of the cushion.
- The equipment weight is divided into two equal parts which are concentrated at two points, each 5 feet beyond the cushion at each end of the vehicle.
- The fuel and payload are concentrated essentially at the center of the cushion.
- The propulsion power plant weight is located at a point 5 feet behind the rear of the cushion.
- The impact is reacted at each end of the cushion. This arrangement is illustrated in the sketch.



The following maximum bending moments result:

Maximum positive bending moment, occurring at center,

$$= \left(\frac{W_G}{2} - \frac{W_{PP\,L}}{3} - \frac{W_E}{2} - \frac{W_S}{4}\right)\frac{\ell}{2} - 2.5\left[W_E + (W_{PP})_P\right]$$

Maximum negative bending moment, occurring at rear reaction

$$= -2.5\left[W_E + 2(W_{PP})_P\right].$$

These expressions, in the form $\frac{BM}{W_G}$ have been utilized for a range of p_c, $\frac{h}{D_e}$, V and W_G to determine the order of magnitude of maximum 1g bending moment that a GEM structure should experience.

4.1.1.4 Load Factors

These being 1g bending moment values, it is necessary to know the order of magnitude of load factor that can be anticipated in the case considered. In the 15-ton machine, it was considered that a maximum of 8g was not unreasonable for impact into typical beach surfaces, with a rigid machine. These impact values may be appreciably less for an inflated machine, except close to the impact points, as a direct result of the flexibility of the machine.

4.1.1.5 Wrinkling Load of the Inflatable GEM

In order to establish the order of magnitude of inflated vehicle dimensions required to withstand the typical loads indicated in the previous section, a simple form of inflated structure is considered. Treating the longitudinal direction of a GEM as a beam, loaded according to the previous section, then the cross-section of the beam is determined from the requirement that the vehicle be able to take at least the critical load without collapsing. The collapse load is less determinate than the wrinkling load, it can be, at most, 2 x the wrinkling load, but never less than the wrinklin load. Therefore, for preliminary analysis, it is assumed that the vehicle must carry the critical load without wrinkling; thus a factor of safety of between 1.0 and 2.0 at collapse is mandatory.

For a typical inflated beam of depth h formed of flat top and bottom sections, separated and restrained by membranes or threads, and sealed at the ends by circular sections, the wrinkling load is derived from the expression for inflation pressure obtained in section 2.1.2.7.

$$M_W = \frac{\pi}{16} \cdot ph^3 \frac{(1 + 4b/\pi h)^2}{(1 + 2b/\pi h)},$$

where p = inflation pressure
b = width of flat part of section
h = depth of section.

Note, however that this is true only where shear and torsion are negligible at the section. As discussed in Chapter II for simple beam structures, the maximum bending moment, which coincides with zero shear, designs the inflation pressure in the absence of torsion. With torsion present, the inflation pressure has to be increased. For the GEM structures considered here, having a length-to-beam ratio of 2:1, the inflation pressure is increased approximately 40 per cent by the torsion created as a result of impacting on diametrically opposite corners of the machine.

4.1.1.6 Variation of Inflation Pressure with Vehicle Size and Type

The analysis outlined in the previous section results in the relationship illustrated in Fig. 4-1. This shows "p", the required inflation pressure (for pure bending moment) per "g" of applied vertical loading, and "h", the depth of the inflated section, plotted against the vehicle length to depth ratio $\frac{\ell}{h}$ for a range of cushion pressure, gross weights, speeds, and operating height ratios. The inflation pressure is divided by the number of "g's" applied in the critical loading case, simply because it has not yet been decided just what does constitute the critical loadings for a range of GEM sizes.

Examination of the figures clearly indicates the areas in which current technology will permit construction of the simple types of GEM structure discussed in this study.

The closer the section comes to be circular in cross-section, $\frac{\ell}{h} = 2$, the lower the inflation pressure required per g of loading; this is to be expected from a structural standpoint. From the point of view of GEM configurations, the vehicles that can most quickly profit from inflatable structure as a primary structure are those utilizing low cushion pressures. As far as physical dimensions are concerned, the smallest vehicles are those utilizing high cushion pressures, but of low gross weight.

Note that to a first order, the inflation pressure depends only on the "g" loading, the cushion pressure, and the (length/depth) ratio, for the 2:1 length-to-breadth ratio vehicle example.

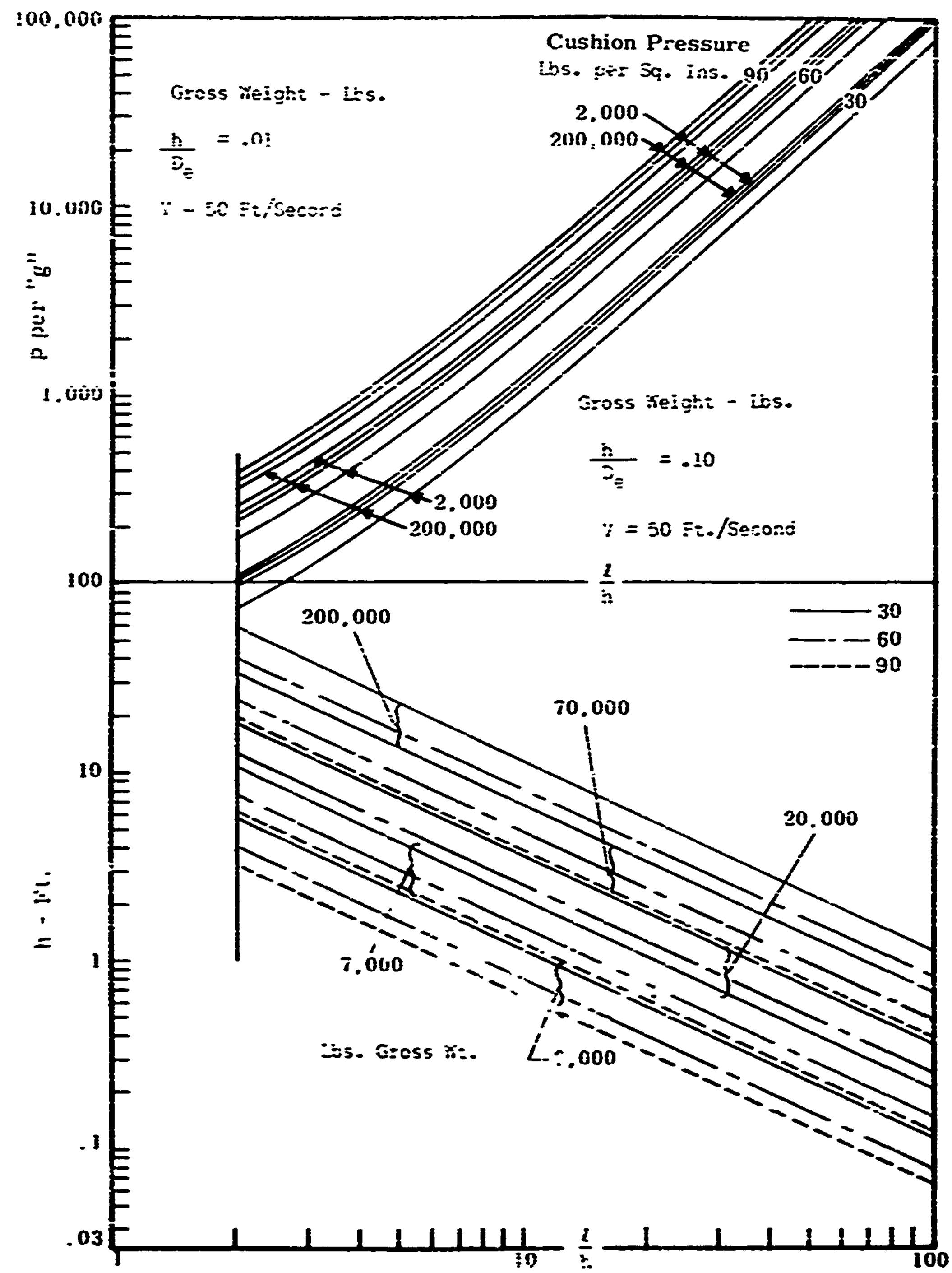


Fig.IV-1. Inflation Pressure Vs. Size For A Range Of GEM Parameters

4.1.2 PRIMARY STRUCTURE APPLICATION

From a study of the structural capabilities of inflatable structures as applied to GEMs, it is evident that, with inflation pressures that are currently practiced and accepted, too great a primary structure depth, in proportion to vehicle length, is required for vehicles designed to withstand operational loadings.

To enable practical use of inflatable structures for the configuration of GEM considered in this report, several alternatives must be considered; they are to:

- Increase inflation pressures to 100 to 140 pounds per square inch.
- Decrease permissible load factors to about 1.2 to 1.5g.
- Mix inflatable and rigid construction such that at least 80 per cent of the maximum bending moment is taken by the rigid structure--in which case the inflatable part serves the purpose of a flexible skin that will absorb impact loading and provide flotation capability. This will probably require a simple form rigid framework to be designed, thus providing adequate strength with rapid assembly capability.

The capability of an inflatable structure is radically altered by the presence of a permeable rigid foam core. Tests conducted at NASA-Langley with foam-cored fabric cylinders show an appreciable increase in structural stiffness--the increase in stiffness being on the order of 1,000 per cent in torsion and 1,000 to 5,000 per cent in bending.

The characteristics of rigid foam-filled inflated structures are appreciably different from those of a simple inflated structure. The presence of the core will permit a given geometry of structure to maintain a much higher load before wrinkle or collapse or a smaller structure for a given load; thus the unsupported inflatable structure. A third very practical possibility is the use of lower inflation pressures to permit the structure to hold a given load. The penalty involved of course is twofold; first, a weight increase, and second, complete loss of the unique transportability aspects and packaging capabilities of inflatable design. However the application may have fairly immediate practicability in short-life vehicles where high material stresses are tolerable, and where foaming-in-place is practical--as in small, one-piece vehicles. Foaming-in-place has met with indifferent success where large amounts of foaming are necessary due to the difficulty of developing a good structural bond between the foam and the skin, and of obtaining a uniform density core. Such a development in a small-scale application suitable for transporting one or two soldiers could be worth while and feasible, but would require further research and development.

4.1.3 SECONDARY APPLICATIONS FOR INFLATABLE STRUCTURES IN GEMS

4.1.3.1 Secondary Structure

Such items as aerodynamic surfaces, cabin structures, roofs, seats, and benches are all amenable to inflatable construction, many examples being in existence today. Considerations of safety in the event of a crash or failure to a main structural component will impose a more severe requirement on these structures in a GEM than has been the case in most applications to date. This in turn means that the structural walls of the components have to be thicker than current practice requires to cater for the increased load, or higher inflation pressures must be utilized.

4.1.3.2 Auxiliary Structures

The items under this category may be fabricated and attached to existing hardware in order to improve their utility of performance. They constitute the most immediate and direct application of inflatable structures to GEMs.

By definition these are neither primary nor secondary structures, and by this token are relatively lightly loaded, or are loaded in some transient fashion that will permit much lower design loads to be considered.

As discussed in the previous chapter, these items inclu , but are not necessarily limited to,

- Peripheral bumpers, to reduce the peak loadings on impact between GEM and ship or sho e.

- Extendable landing mats or pads to provide a lightly loaded support capable of adapting itself to the ground variations, and als. capable of providing adequate flotation on the water.

- Retractable intakes to permit greater intake efficiency during normal operation and yet to provide flush intakes for use during overhead loading and unloading operations, thus eliminating inlet damage.

- Skirts or trunks around the peripheral jet. This use of inflatable structures has promise of the greatest return on its investment in terms of vehicle performance improvements. The use of the skirts will mean either an increase in vehicle operating height or a reduction in vehicle power required, and hence a greater operational utility. The skirts or trunks themselves are operating in a region of static pressure on the order of 15 to 30 pounds per square foot for many vehicles, and they may be just the order of pressure required for normal operation. However, in the event that an increase in operating height is desired, inflation of the skirt or trunk will make it rigid, thus effectively extending the annular jet the full length of the trunk before the curtain air issues and curves in reaction to the cushion pressure. In any event, high inflation pressures are unnecessary since the skirt or trunk will be required to deflect under load and to offer little resistance to obstacles, and yet to maintain its shape in the presence of the cushion pressure of 30 to 90 pounds per square foot.

4.1.4 SELECTION OF MATERIALS

The selection of materials that go to make up an inflated structure skin or web is based on:

- Strength in tension
- Life under load
- Pressure-holding capability
- Environmental consideration (including temperature, humidity, ultraviolet, ozone, abrasion, etc.)

4.1.4.1 Basic Fiber

From the strength aspect, Fortisan, Dacron, and nylon are far superior to all other man-made fibers except glass, and cotton is the natural fiber that approaches these. When the creep characteristics of these fibers are examined, Fortisan is demonstrably poor, and cotton is poor by comparison to nylon and Dacron.

Examination of glass, Dacron, and nylon for resistance to other conditions indicates that they are generally not attacked by acids, alkalis, organic solvents, or mildew. Effect of heat is such that strength reduces approximately linearly to zero at about 300° to 450° F for nylon and Dacron, with Dacron being superior, while glass will retain up to 50 per cent of its strength at 650°F, losing it all by 1,350°F.

4.1.4.2 Coating or Bonding

This must be selected to provide the desired pressure seal, and at the same time to withstand the conditions imposed by the natural and operational environment, including abrasion, ozone, temperature extremes, and solar radiation.

Practically all useful coatings or bondings are members of the family of rubbers--natural and synthetic--that require treatment during or after application to develop their full capabilities. Among the total family, some of those most suited as bonds through a wide temperature range are neoprene, butyl, thiokol, and silicone rubbers, while for abrasion resistance, ozone resistance, general weather resistance, fuel resistance, etc., neoprene butyl, hypalon, polyurethane, and fluorocarbon rubbers are best.

Neoprene and butyl appear to combine the required properties except insofar as very low temperature flexibility and high abrasion resistance are concerned. This question of the right combination of desired qualities has led manufacturers of inflatable products to utilize several coats of different materials to provide the desired properties. Typical of this is the use of neoprene as a pressure seal followed by a thin coat of hypalon to add to the abrasion resistance of the cloth. One of the least satisfactory characteristics of most currently used inflatable fabrics is their low temperature flexibility--no substances are currently available that are suitable in their other characteristics and yet can retain their flexibility down to -65°F.

4.1.4.3 Fabric Weave

Generally speaking, the simple, commercially available "plain weave" is quite adequate where a woven fabric is desirable. The only purpose of a woven fabric is for ease of manufacture, except in the case where a special weave may be desirable in construction, such as a "rip-stop" fabric to minimize the length of a puncture. A cloth, in general, loses strength by comparison to the individual fabrics, although several sets of individual fibers bonded together in layers may also suffer from delamination due to porosity in the material bond. This may also happen to woven fabrics, where several layers are required to carry the load. Bonding or coating a fabric to provide adequate strength and sealing is an area that needs considerable attention, primarily in the actual technique of bonding.

4.1.4.4 Inflation System Power Requirements

From the discussions in Chapter II, it appears that the use of an independent air pump system is most suitable for inflatable structures, either driven from the engine accessory system, or independently by some other means. The use of the separate pumping system will provide the least cost in both power and weight in supplying the low pressure, high volume flow air at the ambient temperatures that are currently required.

The power requirements depend on the desired inflation time, but the approximate HP required can be estimated by taking a figure of 0.5 HP per cubic foot per minute of average inflation rate, i.e.,

$$\frac{\text{Inflation Volume (cubic feet)}}{\text{Inflation Time (minutes)}} .$$

The pumping system should have an emergency capability for operation at high flow rates for short periods to maintain structural integrity, regardless of damage inflicted by enemy action. Since this has to be a continuous capability, emergency bottles of compressed air would not be satisfactory. Mixing engine bleed air at high pressure, and temperature and low volumetric rate with normal inflation system air at low pressure and temperature and high volume could provide the desired emergency capability.

4.1.4.5 Range of Weights and Comparison with Semi-Rigid and Rigid Structures

In order to provide a feeling for the relative weight of an inflatable structure and corresponding conventional structure, the following estimate is made of the weight per square foot of cushion area of the inflatable primary structure developed in the early part of this chapter, excluding the weight of air at the necessary inflation pressure.

The weight estimate is based on the fabric strength required at the section governing the required inflation pressure, and on an approximation to the weight per unit area of suitable fabrics discussed in Chapter I.

$$\text{Weight per square foot} = \frac{\text{(Surface Area) x (Weight per Unit Area)}}{\text{Plan Area}}$$

where the weight per unit corresponds to the fabric strength.

4.1.4.5.1 Weight per Unit Area

From Chapter I, an approximation to the weight-strength relationship for suitable fabrics is as follows:

$$\text{Wt. (ozs. per sq. yd.)} = 10 + \left[\text{Strength, (lbs per inch)} \times \left(\frac{60}{2400}\right)\right]$$

$$= \left(10 + \frac{\text{Strength (lbs. per ft.)}}{480}\right)$$

$$\therefore \text{Wt. (lbs. per sq. ft.)} = \left(10 + \frac{\text{Strength (lbs. per ft.)}}{480}\right) \times \frac{1}{144}$$

$$= \frac{4800 + S \text{ (lbs. per ft.)}}{69,120}$$

4.1.4.5.2 Surface Area $\doteq$ Perimeter x Length (neglecting end caps)

$$= (\pi h + 2b)\ell$$

$$\text{Now } b = \frac{\ell}{2} - h$$

$$\therefore A_s = (\pi h + \ell - 2h)\ell$$

$$= \left(\frac{\pi - 2}{2}\right)\frac{\ell^2}{\frac{\ell}{2h}} + \ell^2$$

$$\therefore \frac{A_s}{\ell^2} = \left[0.57 \Big/ \left(\frac{\ell}{2h}\right)\right] + 1 .$$

4.1.4.5.3 Plan Area $= l \times \frac{l}{2} = \frac{l^2}{2}$

$$\therefore \frac{\text{Surface Area}}{\text{Plan Area}} = 2\frac{A_s}{l^2} = 1.14\sqrt{\frac{l}{2h}} + 2.$$

4.1.4.5.4 Stresses

To a close approximation. the maximum stress will occur when the longitudinal pressure stress and the maximum bending stress are additive

$$f_{max} = f_{x_p} + f_{x_B}$$

Now the required fabric strength, pounds per foot

$$= (f_{max} \times t)$$

$$(f_{max} \times t) = f_{x_p} \times t + f_{x_B} \times t.$$

The maximum bending stress that can occur will just cause wrinkling at the compression face of the section, i.e., at $y = + h/2$.

So, from Chapter II, the magnitude of maximum bending stress

$$f_{x_B} = \frac{Mh/2}{I} = \frac{Mh}{2 \times \frac{\pi th^3}{8} \cdot \left[1 + \frac{4b}{\pi h}\right]}$$

$$\therefore f_{x_B} \times t = \frac{16}{\pi} \cdot \frac{M}{l^2} \cdot \frac{\left(\frac{l}{2h}\right)^2}{\left(1 + \frac{4b}{\pi h}\right)}$$

The maximum pressure stress, from Chapter II, is

$$f_{x_p} = \frac{ph}{4} \cdot \left[\frac{1 + \frac{4b}{\pi h}}{1 + \frac{2}{\pi} \cdot \frac{b}{h}}\right]$$

$$= f_{x_B}, \text{ when the required value for } p \text{ is used.}$$

$$\therefore (f_{max} \cdot t) = 2\left(f_{x_B} \cdot t\right) = \frac{32}{\pi} \cdot \frac{M}{\ell^2} \cdot \frac{\left(\frac{\ell}{2h}\right)^2}{\left(1 + \frac{4b}{\pi h}\right)}$$

4.1.4.5.5 Weights

Hence, inflated structure weight per square foot of planform area is:

$$F_1\left(\frac{\ell}{2h}\right)\left\{10 + F_2 \frac{\ell}{2h} \cdot \frac{M}{\ell^2}\right\}$$

where

$$F_1\left(\frac{\ell}{2h}\right) = \frac{1}{144}\left[\frac{1.14}{\left(\frac{\ell}{2h}\right)} + 2\right]$$

$$F_2\left(\frac{\ell}{2h}\right) = \frac{1}{15\pi}\left[\frac{\frac{\ell}{2h}}{1 + \frac{4}{\pi}\left(\left(\frac{\ell}{2h}\right) - 1\right)}\right]^{2}.$$

This results in the following data for 1g bending moments, covering a range of cushion pressures from 30 to 60 pounds per square foot, vehicle gross weight from 1 ton to 100 tons, height-to-effective diameter ratios from .01 to .10, and speeds from 50 to 150 feet per second. The structure weights lie close together, varying mainly with cushion pressure. For this reason the tabulation is given in average values, with variations; the figures are also plotted on Fig. 4-2, indicating the effects of the various parameters.

ℓ/h		2	3	4	6	10	14	20	30	40	60	100	140	200
Wt. lbs. per sq. ft.	Average	.24	.22	.21	.20	.21	.23	.25	.300	.35	.44	.64	.84	1.14
	±	.01	.01	.015	.02	.03	.04	.05	.08	.105	.15	.255	.36	.51

It is clear from these data that the weight of such structural envelopes will not be a limiting factor; in fact, such structure has an excellent weight-saving potential. Whether this potential can be reached in an actual design is a question for further study, and in the long run depends, as in rigid structures, on the careful and efficient design of joints, attachments, etc. Note that, although the above figures are evaluated for 1g bending moment data, even very severe "g" loadings require weights that are still low by comparison to rigid structures. Almost no change is required for $\frac{\ell}{h}$ from 2.0 to 10.0; when $\frac{\ell}{h}$ is in the region of 50 to 100 or more, the weight increase is approximately in proportion to the increase in "g" loading. In the most heavily loaded case in the table, the weight increases to approximately 12 pounds per square foot, which is still not heavy by conventional structure standards.

Note that the analysis does not take into account the additional weight due to bonding or to drop threads or diaphragms. The increase in weight due to these should be on the order of 10 to 20 per cent, since no air-sealing or environmental protection is generally required.

4.1.4.6 Packaging and Storing

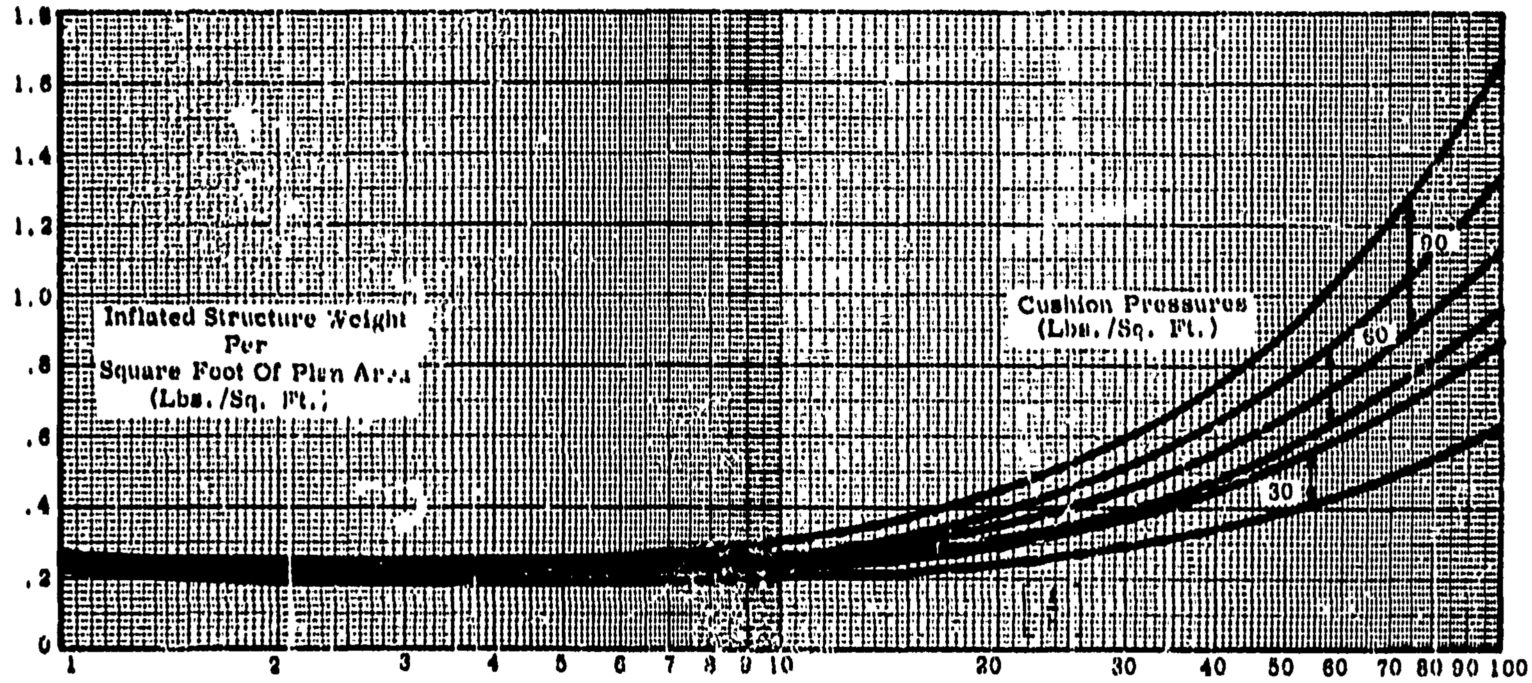


Fig.IV-2. Inflated Structure Weights - Evaluated For Simple Beam Construction Of A Range Of GEMs.

4.1.4.6.1 Packaged Volumes

The packaged volume of an inflated structure is dependent on a large number of factors, the most important being:

- Inflated size
- Configuration
- Material characteristics
- Amount and location of fixed hardware
- Number of modules of which structure is composed.

It is very difficult to generalize on this subject, although some guidance is available from past structures that have been built. Some examples are:

	Storage Volume (cubic feet)		
	Inflated	Packaged	Ratio
Inflated plane	600	42	$\frac{1}{15}$
Inflated boat	800	40	$\frac{1}{20}$
Parachute	2,000	1	$\frac{1}{2000}$
Toroid	12,500	250	$\frac{1}{50}$

In the development of any particular item of inflatable construction where packaging is important, a careful study will be necessary to determine the best means for packaging; this requirement can quite conceivably dictate the choice of configuration and the degree of breakdown in the end product, which deserves thorough investigation. The ease with which the item may be packaged will determine to a large degree its utility in the field and its service; poor packaging may result in rapid fabric and elastomer deterioration. Reference 27 discusses an analytical approach to this problem for fairly simple shapes.

4.1.4.6.2 Storage Environment

Practical materials of the nylon or Dacron family, when combined with a selected elastomer and possibly an additional coating, are quite capable of withstanding most storage environments. The conditions that give rise to some concern for the utility of such structure are mainly at very low temperatures, -40°F to -65°F, when the elastomer may become brittle and crack when flexed. The material will lose its air-retaining qualities as a result even when normal temperatures are required. If the vehicle is stored at these temperatures, and is packed and unpacked at temperatures higher than -40°F, no particular problems should arise. Proper treatment and additives to the elastomer will prevent deterioration due to sunlight, ozone, mildew, fungus, insects, etc., and cure of the structure and appropriate elastomers at temperatures on the order of 180°F to 190°F should provide adequate protection against loss of strength in the tropics.

4.1.4.7 Effects of Operational Environments

4.1.4.7.1 Operations Over Water

Operations over water, as far as the structural material is conce[illegible]ec defines operations in a humid environment, probably with a high sa[illegible] content, through a wide range of temperatures.

As far as current environmental information on fabrics is concerned, all indications are that, again, the nylon and Dacron fabrics show to advantage, with a fairly wide range of elastomer materials that can be married to these fabrics. The actual life of these fabrics under load has not yet been established, although it is known to be well in excess of eight months of complete immersion in salt water, which will correspond to a considerable number of operational hours in a saline spra[illegible] environment--a number appreciably in excess of 10,000 hours.

4.1.4.7.2 Operations Over Land

The most important material aspect consideration resulting from operation over land (or over any exposed hard surface, such as coral) is the ability of the inflated material to withstand applied forces tending to damage the fabric locally, thus creating weak spots susceptible to "blow-out", puncture, and tearing. Such forces can arise easily during passage through unprepared areas, due to impact with sticks, roots, fence posts, damaged buildings, coral heads, and so on.

Although this problem cannot be finally resolved without recourse to tests, at least it can be argued that in many such impact loadings, an inflated structure will show to advantage when comparing flexible structures to rigid. If the energy absorbed on impact is the same for the initial conditions of essentially identical vehicles, then the energy absorbed depends first on the distribution of load at the impact point and then on the energy-absorbing capabilities of the structure. The skin of a metal structure can absorb energy in the form of strain energy, until a physical limit is reached, dependent on the dimensions and material of the region in which the load is applied. This region, generally speaking, is small by comparison to the size of the structure, in contrast to an inflatable structure, in which the effect of the load is transmitted throughout the structure by the air under pressure inside. In an inflatable structure, the local skin at the impact point has very little shear or bending stiffness; consequently, the energy of the impact is absorbed in tensile strain energy in the skin and pressure strain energy in the interior. The distribution between tensile and pressure strain energy will depend very much on the intensity of the impact load. A very high intensity load will cause a deep local indentation and high tensile stresses; this in turn means relatively high tensile strain energy and the tensile strength of the skin may quickly be exceeded. A low intensity load, well distributed, will give rise to low tensile stresses, high pressure strain energy, and no rupture, since the reservoir for pressure strain energy is very large.

Hence, except for high intensity impact loadings where tearing may result in loss of pressure for inflated structure, but relatively no loss in strength for the metal structure, the inflated structure will show a better capacity for absorbing the impact loads without damage and with reduced accelerations during impact.

4.1.4.8 Effects of Combat Environments

The primary question in combat becomes "how much battle damage can the structure withstand before the vehicle is useless?" This question poses the biggest problem when the utility of inflatable structures in military vehicles has to be established. No satisfactory arguments have yet been developed to show how a simple inflated structure is better than a rigid structure in a battlefield. More sophistication can be introduced for limited applications that may very well provide a militarily useful vehicle, such as foaming the inflated interior with a rigid foam to provide some measure of structural strength if the pressure cannot be maintained, but such methods tend to defeat the purpose of an inflatable structure in most applications, since they introduce more weight and more complexity and less reliability into what could be an otherwise simple concept.

Other aspects of a combat theatre, "radar reflectivity", "infrared radiation", maintenance, repair, and so on, are mainly functions of detail design. A few comments should suffice to point out any important factors.

4.1.4.8.1 Radar Reflectivity

A normal inflated structure material is radar transparent, so that the radar target that remains depends mainly on the power plants, fans, and propellers. However, it is well known that a large part of the radar image of a propeller airplane is due to the return from the exposed propeller. In the case of an inflatable structure (GEM), the lift system fans are, in effect, exposed and will quite probably give rise to a substantial radar return; if propulsion propellers are used, an even worse situation may exist.

4.1.4.8.2 Infrared Radiation

This should be about the same as for ordinary vehicles with similar power plant.

4.1.4.8.3 Maintenance and Repair

Cold repairs in the field should be feasible and rapid, somewhat similar to cold repairs to aircraft and truck tires, but with different types of equipment. Vulcanizing may also be quite feasible, for field repair to small areas. Large areas of damage may have to be dealt with at a repair depot, where adequate floor space should be available. Little maintenance will be required; regular checks to inspect for deterioration will be essential, as will fairly regular washings to remove accumulations of grit, oil, fuel, etc.

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

5.1 CONCLUSIONS

5.1.1 USE OF INFLATABLE STRUCTURES IN GEMs

The present utility of inflatable structures in GEMs is primarily confined to application in add-on or auxiliary structures, where loading conditions are not too severe, and where GEM utility would be appreciably increased.

5.1.2 MATERIAL TYPE AND STRUCTURAL FORM

5.1.2.1 Materials

From a strength point of view, creep-rupture characteristics and inertness of Dacron and nylon fibers are most suitable. They are workable, reliable, and much experience has been gained in handling them.

These fibers can be woven preferably in a simple weave for ease of handling. Plied fibers or fabrics are essential in some heavily stressed areas, and are satisfactory, provided adequate bonding techniques are utilized to prevent delamination. Such techniques have been developed in specific applications.

5.1.2.2 Structural Form and Characteristics

The structural form may be adjusted to suit almost any shape provided the loading conditions are such as to allow the desired dimensions for currently acceptable inflation pressures. The most efficient forms are the same as for normal structures--tubes for torsion, deep beams for bending, large cross-sections for compression. The notable difference in behavior for a given loading and geometry, assuming the inflation pressure is acceptable, is the increase in structural deflection.

Where enclosed structures are desirable, as in cabins, etc., inflated wall structures may be satisfactorily developed, but unless the wall thickness is appreciable, such structures provide only shelter.

5.1.3 EFFECT OF INFLATABLE STRUCTURES ON GEM PERFORMANCE

5.1.3.1 Effect of Structure Weight

Where an inflatable structure is usable in place of an equivalent rigid structure, an appreciable weight saving should result, which can be reflected either in increased range, increased endurance, or reduced gross weight.

5.1.3.2 Effect of Auxiliary Structures

The use of inflatable structures as items that do not directly provide the vehicle strength, but are responsible to a large degree for determining the strength that must be provided in the main structure is apparent. Such items as bumpers, landing mats, inflatable skirts and trunks, impact-attenuations or "shock-absorber", and so on, all provide for a more efficient vehicle at a low weight penalty.

5.1.4 INFLUENCE OF OPERATING ENVIRONMENT ON INFLATABLE STRUCTURE UTILIZATION IN GEMs

With currently developed fabric constructional methods and coatings there need be very few restrictions imposed by the operating environment. Some unknown factors still exist, notably the effect of sudden punctures by enemy action or by obstacles, the likelihood of snow and ice accretion, susceptibility to fire, and the resistance to puncture and tearing. Aside from these, the only real restriction that appears to exist is in the elastomer embrittlement at very low temperatures, below -40°F.

5.1.5 EFFECT OF VARIATION IN GEM SIZE ON UTILIZATION OF INFLATABLE STRUCTURES AND OTHER APPLICATIONS

5.1.5.1 Primary Structures

Increase in GEM size, without change in gross weight, implying as it does, a reduced cushion pressure, is beneficial for utility of inflatable structures, since it reduces the inflation pressure required on the structure.

If GEM size implies an increase in gross weight, as well as physical dimensions, no advantage accrues, and a higher inflation pressure may well be demanded.

Change in GEM configuration has a marked effect on the utilization of inflatable structures, and is about twice as powerful as a change in cushion pressure. Doubling the structural depth, for instance, will result in approximately one-fifth of the inflation pressure, whereas, doubling the size will require approximately two-fifths. Note that a hundredfold increase in gross weight at constant cushion pressure only requires approximately a 15 per cent increase in inflation pressure.

5.1.5.2 Secondary Structures

Insofar as secondary structures carry load continously but at a lower level than the primary structure, and the loads and size of structure will vary in accordance with the loads and size of the main structure, the affect of GEM size on inflatable secondary structures will be similar to that in primary structures, but less critical or demanding--lower inflation pressures may be possible with desired configurations, or more reasonable configurations with acceptable pressures.

5.1.5.3 Auxiliary Structures

These structures are designed to loading criteria that do not reflect the major design cases for the primary GEM structure, and hence are relatively independent of GEM size. This is a very useful aspect of such structures, since their utility is enhanced by the fact that for each auxiliary application, a range of modular units would be developed to cater all GEM sizes.

5.2 RECOMMENDATIONS

5.2.1 FUTURE DEVELOPMENTS REQUIRED TO IMPROVE THE UTILIZATION OF INFLATABLE STRUCTURES IN GEMs

First and foremost, an increase in acceptable inflation pressures, by a factor of 10 or more, will enable construction of inflatable primary structures to withstand the most severe loading conditions, while still retaining an acceptably slim configuration. In general, the ability to confidently design to higher inflation pressures is an all around gain.

Development of higher strength-weight materials with high Young's modules, and high ultimate strength.

Development of strongly abrasion resistant coatings of light weight, to provide the environmental protection required for overland and amphibious use.

Development of elastomers that will meet more completely all of the environmental conditions discussed in Chapter I, in particular the low temperature condition discussed.

5.2.2. SPECIFIC AREAS OF INVESTIGATION REQUIRED

Investigation of the application of currently acceptable inflatable structures in the auxiliary GEM structure area, and allied applications, including the following:

- Inflatable fenders for mounting on GEMs along the side, at the bow or underneath to reduce the initial loading conditions to which GEMs need be designed.

- Inflatable trunks or skirts, to permit GEMs to operate over rough terrain or in heavy seas with low lift power and greater maneuverability.

- Inflatable bumper pads for use alongside or inside LSDs or LPDs, to minimize the impact loading on GEMs and other amphibious craft inherent in operations in heavy seas.

- Inflatable landing pads as an alternative to metal plates, enabling satisfactory operation in much softer terrain for a given weight penalty.

- Variable shock-absorber systems, adjustable to suit the roughness of the operating terrain and the vehicle speed. This is of particular value in overwater operation when lightly damped GEMs will be severely limited in speed due to resonance in heave and pitch.

- Inflatable air intakes to provide efficient operation at high speed, while still providing efficient operation at low speed with no intakes to obstruct loading and unloading.

- Inflatable stabilizing surfaces that may be deployed at will.

- Inflatable bow sections to reduce water impact loads transmitted to the GEM structure.

- A combined bumper, skirt or trunk, bow, and landing pad assembly that can be constructed in modular form for attachment to various sizes and types of vehicles.

- Several minor applications, such as seats, benches, canopies, etc.

Investigation of the application of currently inflatable structure in the primary structure role, in vehicles of appreciable depth (of the same order as vehicle width). Possible applications are in the drone vehicle field, the stretcher carrying field, equipment carriers, and possibly one-man vehicles.

5.2.3 GENERAL RECOMMENDATIONS

As soon as the utility of inflatable structures in the GEM is established, it is strongly recommended that a program be initiated to collate all available data on materials, materials testing, structural testing, for use in the formulation of criteria for the design of inflatable structures for GEMs.

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Unclassified Report

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2. Transportability--Ground Effect Machines
3. Logistic over the shore Carrier (LOTS)
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5. Contract DA 44-177-TC-726

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