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AMMRC TR 84-27

FEASIBILITY STUDY ON THE DESIGN OF REINFORCED PLASTIC COMPONENTS FOR THE LVTP (7) VEHICLE SHAFTS

DECEMBER 1984

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ABSTRACT

This document pertains to a feasibility study in the design and manufacture of a composite drive shaft and propeller shaft for the LVTP (7) amphibious vehicle. The report contains an analysis of the present steel shaft design and the two design approaches to a composite drive shaft. The first is a two piece drive shaft; the second is a combined single drive shaft eliminating several other parts. The design section includes the design of a joint to interface between the steel couplings and the composite shaft. The second part of this study is a manufacturing analysis of the production of 2,000 shafts for the actual retrofitting of the vehicle. It includes individual sections on fabrication, machining, and assembly of the proposed composite drive shaft.
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FOREWORD

Contract DAAG46-83-C-0025 was awarded by the Army Materials and Mechanics Research Center (AMMRC) to Brunswick's Defense Division, Lincoln, Nebraska. The Phase I contract covered the design and production feasibility analysis of a reinforced plastic drive shaft and propeller shaft for the LVTP(7) Amphibious Armored Personnel Carrier.

Work was conducted from May, 1983 to December, 1984. The work resulted in a recommendation for a single composite shaft that offers significant weight savings over the existing steel shafts. This assembly would replace the existing two steel shafts and requires only a minor modification to the vehicle bulkhead to allow insertion of a larger diameter shaft. An alternative which would not require vehicle modifications would be a composite two-shaft system; however, the weight savings over the present steel shafts would not be significant.

Mr. Peter Dehmer served as COTR for this contract and monitored progress, provided technical information, and attended the progress review meetings. Technical personnel from Brunswick Corporati n included Mr. Jim McGee, who served as the project and manufacturing engineer coordinating this project, and Mr. David Shy, who served as the design engineer. Other technical personnel that contributed are as follows:

Mr. B.E. Spencer	Program Manager for Composite Shaft Development
Mr. A.R. Cederberg	Research Consultant
Mr. W.D. Humphrey	Product Development Manager

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1.0 INTRODUCTION AND SUMMARY

Brunswick was awarded the Phase I Contract DAAG46-83-C-0025 to study the design and manufacturing feasibility of replacing the metal shaft assembly of the LVTP(7) Amphibious Armored Personnel Carrier with a composite drive shaft produced from reinforced plastics. The object of this study was to maintain or increase service life and reliability of the assembly while reducing weight and costs.

1.1 Introduction

The LVTP(7) is an amphibious armored personnel carrier that can be propelled by treads for land travel or by a water jet propulsion system when operating in a marine environment. Figures 1 through 8 show the vehicle and its drive shaft components that were the subject of this study. Figure 1 illustrates a side view of the vehicle where the shafts are hidden from view. Figure 2 is a view of the thrust directing system that redirects the flow from the jet pumps for maneuvering and reversal. The output of the jet pump is exhibited in Figure 3 where the thrust directing system has been retracted for forward movement. Figure 4 is viewed from the vehicle underside and shows the propeller shaft, one of the shafts being studied. Figure 5 depicts the drive shaft from inside the vehicle with the protective cover removed, while Figure 6 shows the protective cover in place. Figure 7 illustrates the location of the water seal in the vehicle viewed from the vehicle underside, with Figure 8 showing a replacement water seal.

There are two identical shaft sets in this vehicle which rotate in opposite directions. Each set is composed of a drive shaft, a universal joint coupling, and a propeller shaft that is connected directly to the jet drive. Presently, all the drive train components are steel except the seals. Phase I

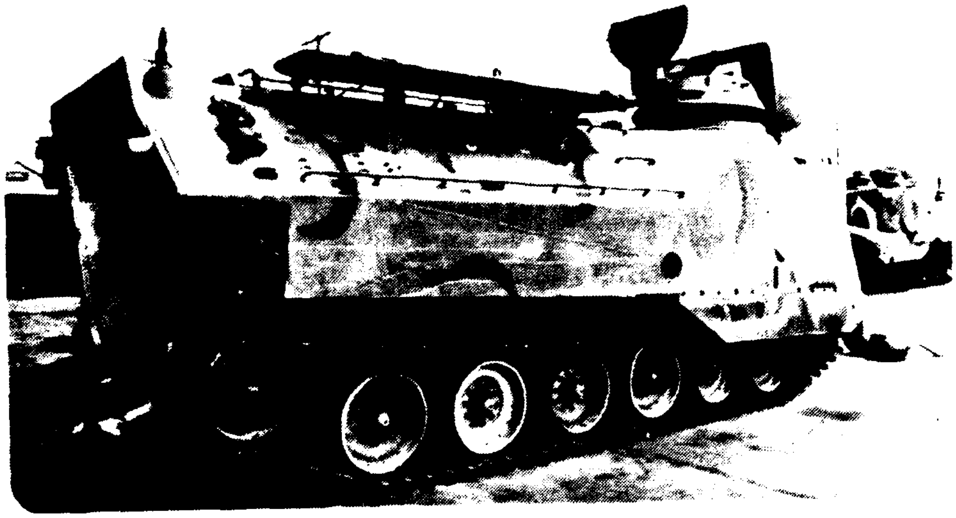


Figure 1. LVTP (7) Side View

Figure 2. LVTP (7) Aft View, Deflector Extended

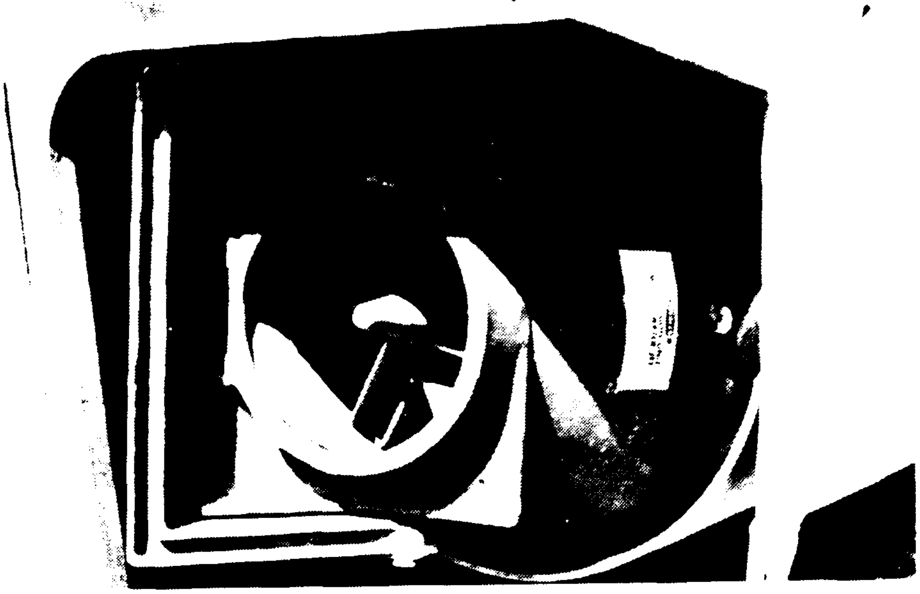


Figure 3. LVTP (7) Aft View, Deflector Retracted

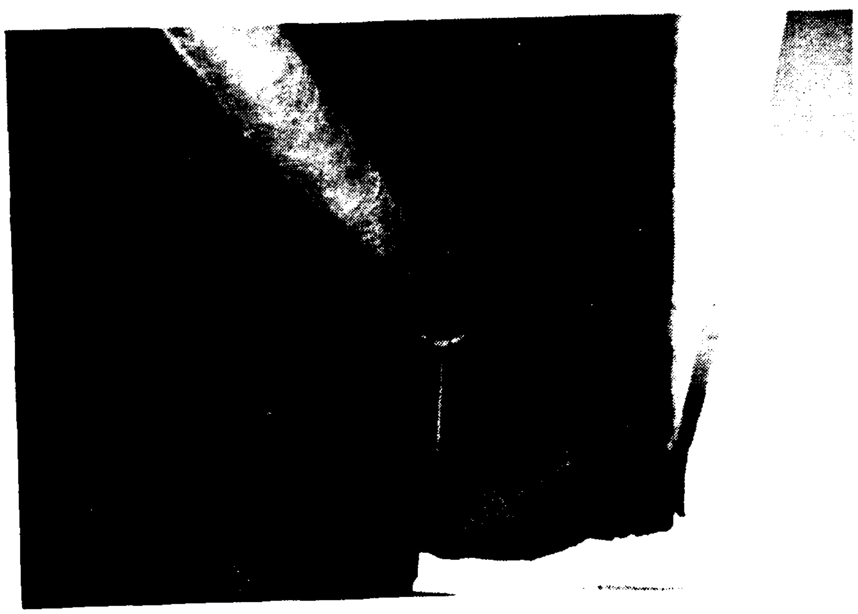


Figure 4. LVTP (7) View From Underside of Propeller Shaft

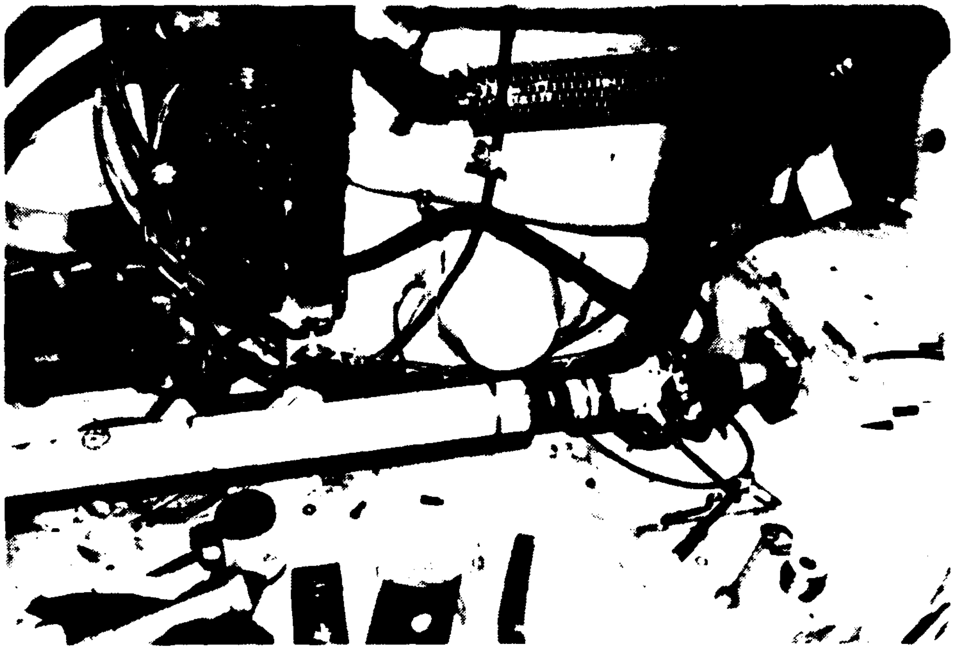


Figure 5. LVTP (7) Inside View, Drive Shaft to Propeller Shaft Connection

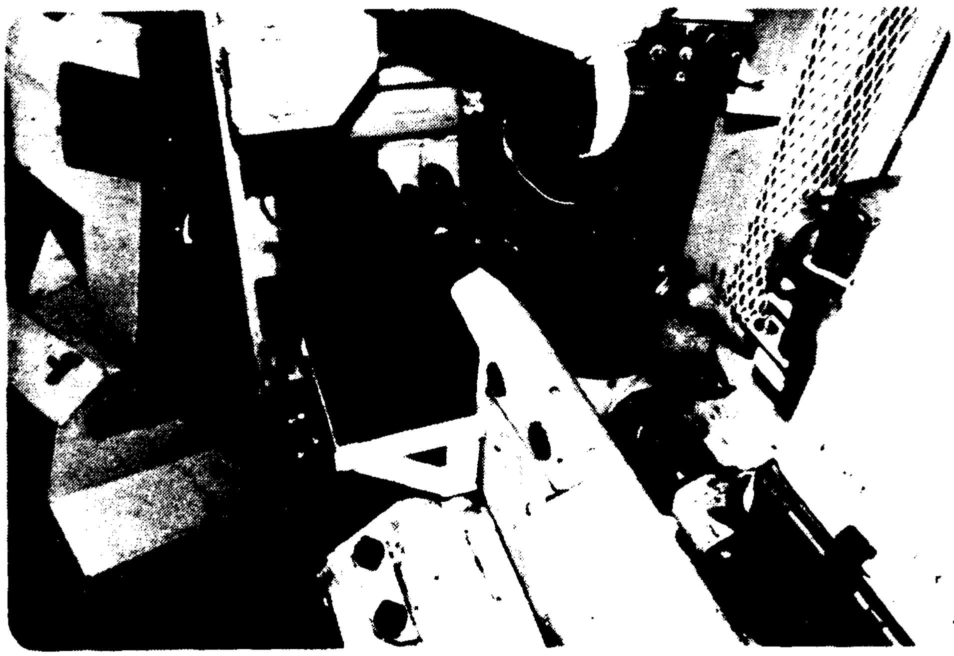


Figure 6. LVTP (7) Inside View, Drive Shaft Cover in Place

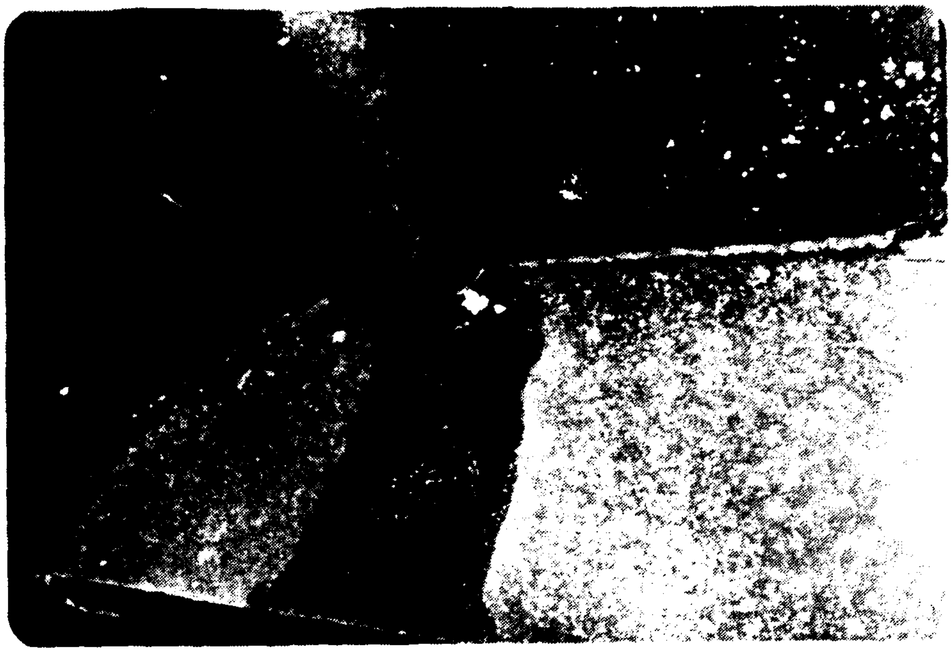


Figure 7. LVTP (7) View of Seal From Outside

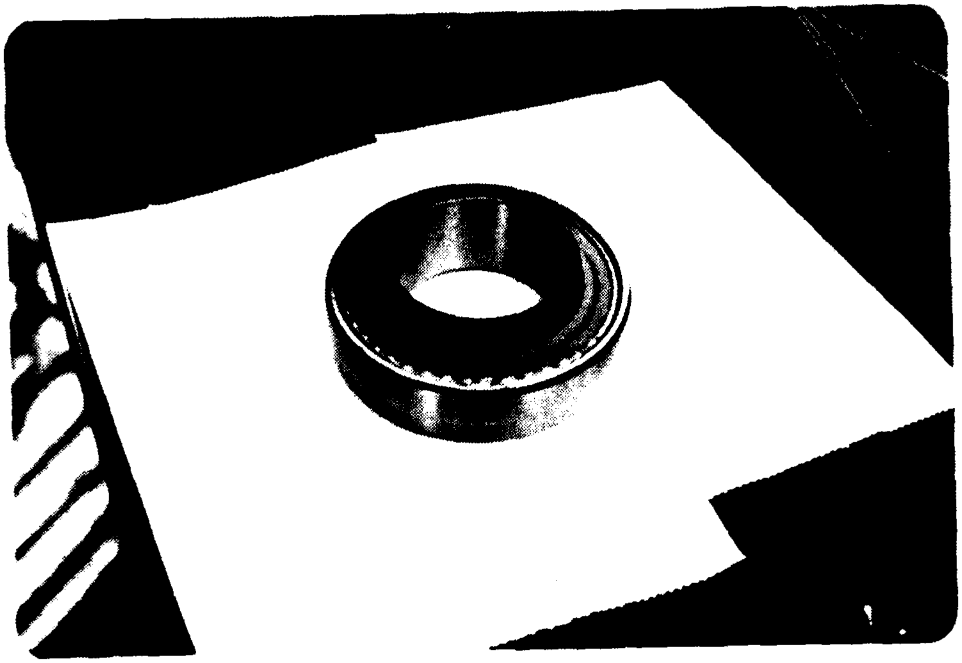


Figure 8. LVTP (7) Closeup of Seal

investigated the feasibility of producing a laminate composite drive shaft which would replace the existing two steel drive shafts, with reduced weight and a reduction in the number of total parts required.

A design was recommended and a manufacturing sequence selected as part of the feasibility study. A prototyping and qualification demonstration program for five shafts has been bid but has not been funded to date. Price and production methods were proposed for a 2,000 shaft production run. These recommendations have also been included as part of this feasibility investigation.

1.2 Summary of Design Selections

There were two design configurations for the composite shaft studied. The first was a single-piece shaft 160 inches long and the second was a two-shaft design similar to the present system. These design configurations and the present steel shaft are shown in Figure 9. The design selection process employed a computer modeling process where different design configurations (see Table VIII in Section 4.3) of materials and material thicknesses were calculated and plotted as the design curves shown in Figures 11 through 17 in Section 4.0. From these curves, a baseline and alternate design were selected. The factors of weight, cost, and critical speed were considered in the selection. All of the design configurations were selected to meet the torque requirement and required safety factor.

The baseline design selected by this study was a single-piece shaft 160 inches long. This design is very weight efficient, saving 36 pounds over the current two-piece metal shaft. It consists of approximately 40 percent,

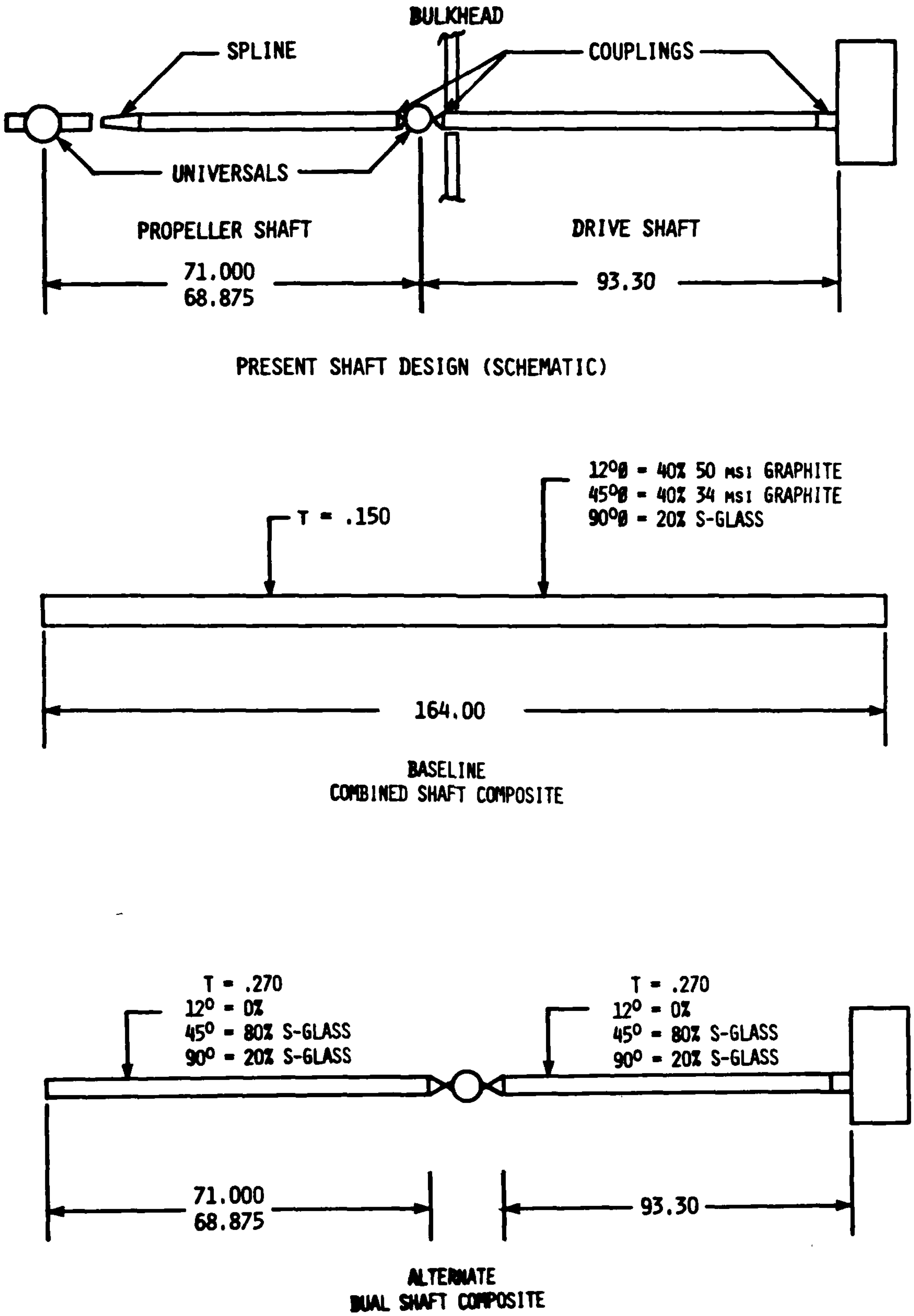


Figure 9. Shaft Designs (Schematic)

50 msi (msi refers to the modulus of the fiber x 10^6 psi) graphite, 40 percent 34 msi graphite, and 20 percent S-2 glass reinforcement. The laminate thickness in the midshaft area is .150 inch, and the construction is shown in Table I. At the joint sections, the thickness is reinforced with 50 msi graphite to provide .280 inch composite thickness for the bearing loads. The shaft would be manufactured by filament wet winding using Lincoln Resin Formulation (LRF) 092 resin system for both graphite and glass sequences. Curing would be performed using a steam heated mandrel. A single steel shaft is not feasible since its critical frequency would fall within the vehicle's normal operating range and thus, possibly cause mechanical failures. It is projected that the combined composite shafts could be produced on a 2,000 unit purchase order for approximately $1,700 per assembly in 1985 dollars in accordance with the plan in Section 6.0.

The alternate design selected by this study is a two-shaft design. One shaft is 80 inches long, and the other is 60 inches long. Only the longer shaft was analyzed since the same design can be used for the shorter shaft. This two-shaft concept is lowest in cost; however, the weight saving is negligible since the two center steel couplings are not eliminated.

The two-shaft design, shown in Table I, uses all S-2 glass with a .270 inch wall. The manufacturing would be similar to the baseline design, except only one reinforcement material would be used in the basic tube making it more efficient to manufacture.

A direct comparison of the present system, the baseline, and the alternate recommendation is presented in Table II. The baseline design has a greater margin of safety in torque carrying ability; however, its critical speed is nearer the maximum operating speed than the original shaft design.

Table I

LAMINATE CONSTRUCTION FOR SELECTED DESIGNS

Combined Shaft Baseline 160 inch Length

Layer	Orientation	Composite Thickness	Material
1	90°	.0075	S-2 Glass
2	±45°	.020	34 msi Graphite
3	±12°	.030	50 msi Graphite
4	90°	.0075	S-2 Glass
5	±12°	.030	50 msi Graphite
6	90°	.0075	S-2 Glass
7	±45°	.020	34 msi Graphite
8	90°	.0075	S-2 Glass
9	±45°	.020	34 msi Graphite
	Total	.150	

Dual Shaft Alternate 80 inch Length

Layer	Orientation	Composite Thickness	Material
1	90°	.018	S-2 Glass
2	±45°	.072	S-2 Glass
3	90°	.018	S-2 Glass
4	±45°	.071	S-2 Glass
5	90°	.019	S-2 Glass
6	±45°	.072	S-2 Glass
	Total	.270	

Fiber thickness is 60 percent of composite thickness.

Table II

SUMMARY OF RESULTS

	Present Design Steel	Combined Shaft	Dual Shaft
CHARACTERISTICS			
No. of Shafts	2	1	2
Type of Material	Steel	Hybrid Composite	Composite
Grade of Material	1020	High Modulus and Standard Graphite, Glass, and Epoxy	of Glass and Epoxy
Wall Thickness (inches)	.094	.150	.270
PROPERTIES			
Torsional Load Carrying Capabilities (ft.-lb.)	4056	4225	4225
WEIGHT			
Shaft 1 and 2 Weight (lbs./in.)	.248	.0767	.164
Estimated Relative Weight with Joints (lbs.)	52	22	43
Approximate Cost		See Appendix B	
Maximum Operating Speed	1020 rpm	1020 rpm	1020 rpm
Natural Frequency			
Propeller Shaft Steel	4613 rpm		
Drive Shaft Steel	3045 rpm		
Drive Shaft, Composite (80")			2360 rpm
Single Combined Propeller and Drive Shaft, Composite		1100 rpm	
Maximum Shear Stress	33.2 ksi	N/A	N/A
Margin of Safety (%)	+.16	+.25 min.*	+.25 min.*

*Based on allowable fiber stress in Table VII.

The design of the composite tube center section is only part of the total design. The attachment methods and stiffening of the composite at the ends of the shaft are also very important in providing a complete shaft. The chosen coupling design consists of a steel hollow tube fastened to the composite shaft by two rows of 10 rivets at each end. The calculated angular deflection between the coupling and the composite are matched such that the two rows of rivets carry nearly the same load. To accomplish this, the shaft ends have been stiffened by the inclusion of five layers of 50 msi graphite. The reinforcing layers are staggered and distributed evenly through the laminations. The total composite thickness at the end of the shaft increases by .070 inches to compensate for the bearing loads.

1.3 Summary of Phase I Study

This section addresses the work performed in addressing the statement of work in the Phase I effort. Design through manufacturing operations are discussed including several configuration trade-offs. The following discussion will summarize the contents of this report, section by section.

1.0 INTRODUCTION

Provides background information on the vehicle and an overview of the report.

2.0 WORK STATEMENT AND REQUIREMENTS

Defines the work to be accomplished under this contract.

3.0 PRESENT STEEL SHAFT

The existing steel shaft is analyzed and the loading criteria established for: 1) torsional buckling, 2) maximum shear stress, 3) lateral natural frequency (critical speed), and 4) torsional stiffness. Table II summarizes the present steel shaft design.

4.0 DESIGN OF THE COMPOSITE TUBE

The available materials are reviewed, and a decision is made to use 50 msi and 34 msi graphite, S-2 glass, and LRF-092 epoxy resin in the baseline tube design. The percent content of each material is defined and the orientation selected. Two designs are investigated: a two-shaft (dual) exact replacement and a single-shaft (combined) are analyzed. The single-shaft configuration is selected as baseline based on a decided weight advantage over the two-shaft unit.

5.0 JOINT DESIGN

Three joint concepts are identified and analyzed. The riveted joint is selected with two rows of ten rivets each. The composite construction and the metal coupling are analyzed for strength and fatigue.

6.0 MANUFACTURING TRADE STUDY

The basic trade-off decisions for selection of the manufacturing processes are identified and made. There are three steps defined: fabrication, machining, and assembly.

7.0 PHASE II DEVELOPMENT

The Phase II development is defined and a schedule established to produce prototype shafts and perform a testing program.

8.0 PRODUCTION OF 2,000 UNITS

The manufacturing processes for a production run of 2,000 shafts are defined; and the required tooling, capital equipment, and total labor hours are summarized.

Following Section 8.0 are two appendices. Appendix A covers the design of large thickwall shafts for a naval application. This appendix supplies some of the basis for the feasibility analysis of this report. Appendix B provides cost data and projected pricing for Phase II and the production of the 2,000 proposed units.

2.0 WORK STATEMENT AND REQUIREMENTS

This section defines the work statement defined under Contract DAAG46-83-C-0025, Items 0001AC and 0001AD.

According to the contract, the work statement for Phase I was defined as follows:

1. Perform structural analysis of the current existing shafts to establish the design criteria for the composite shafts. (Refer to Section 3.0.)

2. Perform feasibility analysis of redesigning shaft components with composite to optimize weight reduction, while maintaining or increasing service life, and reliability. Determine if the shaft is stiffness or strength critical. (Refer to Section 4.0.)

3. Special attention should be given to the design of a single composite shaft to replace the drive/propeller shaft combination presently used. (Refer to Section 4.0.)

4. Address potential problem areas and the secondary manufacturing operation needed for the drive and propeller shaft. (Refer to Sections 5.0 and 6.0.)

5. Redesign the shaft using composite material properties and design criteria specified. (Refer to Sections 4.0 and 5.0.)

6. Perform a manufacturing analysis and recommend techniques to produce demonstration and production units. (Refer to Sections 6.0, 7.0, and 8.0.)

7. Project per part cost for production quantities using an economic analysis. (Refer to Appendix B.)

3.0 STEEL SHAFT DESIGN ANALYSIS

This section analyzes the existing steel shafts in the LVTP(7) vehicle using the design loads provided.

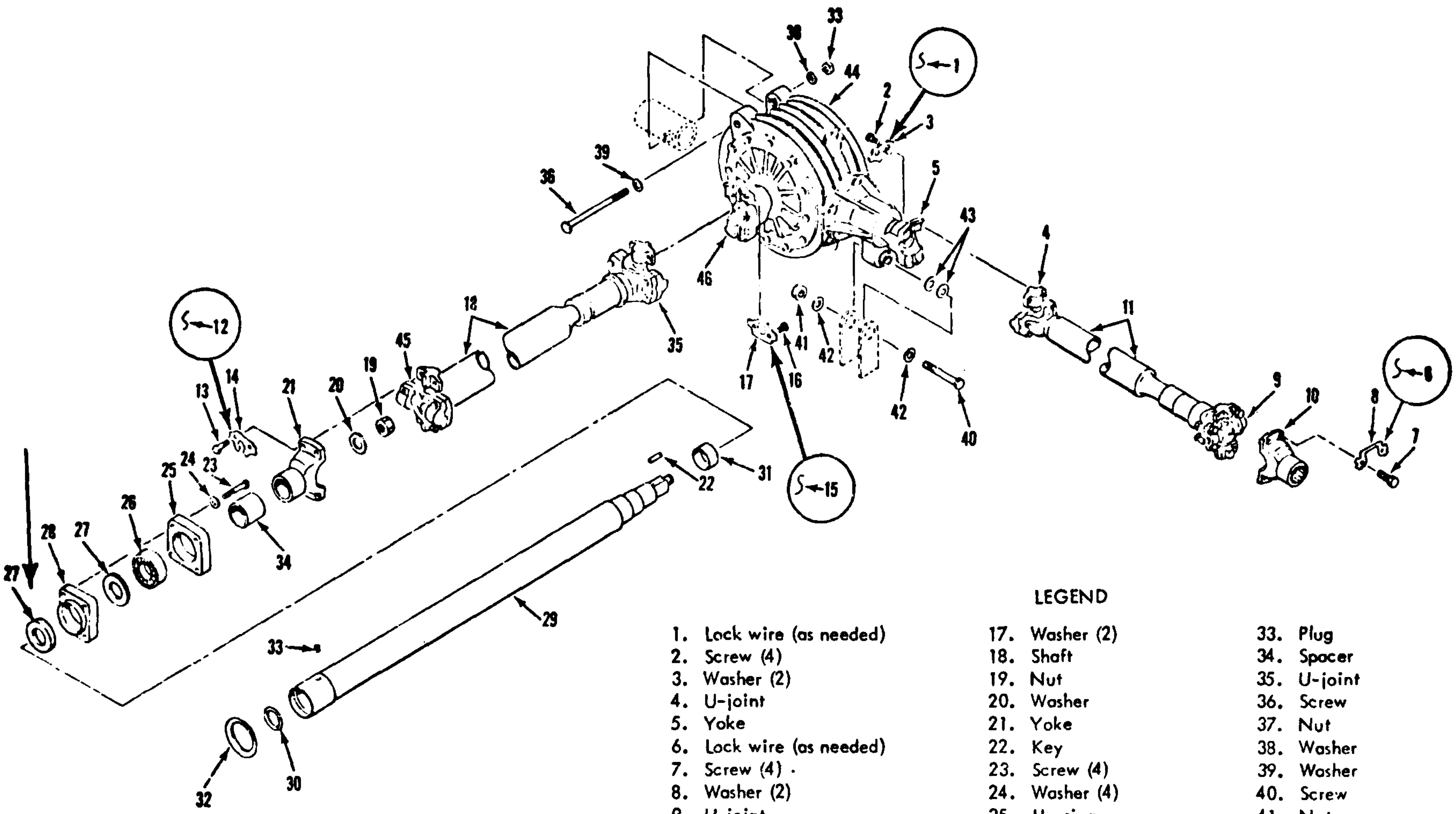
The analysis was performed for: 1) torsional buckling, 2) maximum shear stress, 3) lateral natural frequency (critical speed), and 4) stiffness. The results of this investigation are summarized in Table II under the heading "Present Design."

The current shaft design includes a propeller shaft and drive shaft, as shown in Figure 10. One end of the drive shaft is connected to the impeller, and one end of the propeller shaft is connected to a U-joint which is linked to the main drive train. The two shafts are joined together by a U-joint with the propeller shaft passing through a water-tight bulkhead to the impeller. Since the shafts are separated by U-joints, it is reasonable to analyze the shafts separately. The basic dimensions for the drive shaft are 80 inches in length, 3 inches in outside diameter, and .095 inches in thickness. The propeller shaft has the same dimensions except the length is 65 inches. The material for the current shaft is 1020 steel. The properties of interest for this material are presented in Table III.

Design details from Solicitation No. DAAG46-83-R-0011 are shown in Table IV.

3.1 Torsional Buckling Analysis

The torsional buckling shear stress can be determined by the following formula for a thin-walled circular tube with hinged ends:



LEGEND

1. Lock wire (as needed)
2. Screw (4)
3. Washer (2)
4. U-joint
5. Yoke
6. Lock wire (as needed)
7. Screw (4)
8. Washer (2)
9. U-joint
10. Yoke
11. Shaft
12. Lock wire (as needed)
13. Screw (4)
14. Washer (2)
15. Lock wire (as needed)
16. Screw (4)
17. Washer (2)
18. Shaft
19. Nut
20. Washer
21. Yoke
22. Key
23. Screw (4)
24. Washer (4)
25. Housing
26. Bearing
27. Seal (2)
28. Housing
29. Shaft
30. O-ring packing
31. Bushing
32. Slinger ring
33. Plug
34. Spacer
35. U-joint
36. Screw
37. Nut
38. Washer
39. Washer
40. Screw
41. Nut
42. Washer (2)
43. Shim (as needed)
44. Right angle drive
45. U-joint
46. Yoke

Figure 10. Present Shaft Design (Assembly View)

Table III

STEEL SHAFT DESIGN SUMMARY

Physical Constraints

Shaft length = 80" (drive shaft), 65" (propeller shaft)
Ends hinged, therefore, wall free to change angular direction

Material Properties

Weight (weight per inch) = .248 lbs./in.
Present shaft constructed of 1020 steel

Tensile stress to yield σ_y = 66 ksi

Tensile stress to ultimate failure σ_{ult} = 78 ksi

Shear modulus G = 11.0 msi
Tensile modulus E = 29.0 msi
Shear stress allowable = 39.6 ksi

Design Properties

Critical shear buckling stress* = 5,673.9 ksi (drive shaft)

Margin of safety on maximum shear stress = .16

Critical speed* = 3,800 rpm (propeller shaft)
3,000 rpm (drive shaft)

Stiffness EI = 26.6×10^6 lbs. x in.^{2}
JG = 20.1×10^6 lbs./in.^{2}

*Roark, Raymond J., and Warren C. Young, Formulas for Stress and Strain, 5th edition, McGraw-Hill, 1975, New York.

Table IV

DESIGN CRITERIA

Normal Operating Speed	0 - 940 rpm
Operating Temperature Range	-40°F to +200°F
Over Speed Capacity	1020 rpm for 1 hour without loss of life rating
Operating Life Rating	145 hours min. when operated with the following duty cycle and without lubrication

Duty Cycle Torque (ft.-lb.)	Speed (rpm)	Time (hours)
935	940	120
1,870	940	25
3,380	0	Momentary Static W/O Yielding

$$T' = \frac{E}{1-\nu^2}(t/\ell)^2\ (1.27 + \sqrt{9.64 + 0.466H^{1.5}})\ \ *$$

where, $H = \sqrt{1-\nu^2}\ \ (\ell^2/rt)$ for $r/t > 10$

T' = torsional buckling stress
E = elastic modulus
t = wall thickness
l = length of tube
H = substitution term
r = radius of tube
ν = Poisson's ratio

This formula assumes the ends are hinged, the wall of the shaft is free to change angle, and circular sections remain circular. Tests indicate that the actual buckling stress is from 60 to 75 percent of this theoretical value, with the majority of the data points nearer to 75 percent. A conservative 60 percent factor will be used in the analysis.

For 1020 stainless steel (E = 29.0 msi and ν = 0.3), the critical torsional buckling shear stress for the drive and propeller shaft is 5,674 ksi and 8,594 ksi, respectively. The maximum allowable shear stress is 39.6 ksi. This shows a very high margin of safety for torsional buckling.

3.2 Maximum Shear Stress Analysis

The maximum shear stress can be determined by the following formula:

where, T = maximum torque = 3,380 ft.-lb.
r_o = outside radius of the tube = 1.5 in.
r_i = inside radius of the tube = 1.405 in.
J = polar moment of inertia
S_{max} = maximum shear stress

$$S_{max} = T\ r_o/J$$

---

*Roark, Raymond J., and Warren C. Young, Formulas for Stress and Strain, 5th edition, McGraw-Hill, 1975, New York.

$$J = 1/2\ \pi\ (r_o^4 - r_i^4)$$

$$= 1/2\ \pi\ (1.5^4 - 1.405^4) = 1.831^4 \text{ in.}$$

$$S_{max} = \frac{3,380 \times 12 \times 1.5}{1.831} = 33,226 \text{ psi} = 33.2 \text{ ksi}$$

Shear stress allowable = 39.6 ksi

The margin of safety is calculated below:

$$\text{M.S. (Margin of Safety)} = \frac{39.6 - 33.2}{39.6} = + .16$$

3.3 Lateral Natural Frequency Analysis (Critical Speed)

The lateral natural frequency can be determined by the following formula:

$$fn = \frac{98.7}{2\pi} \sqrt{\frac{E\ I\ g}{\omega\ \ell^4}}\ {**}$$

fn = natural frequency, rev/sec
E = Young's modulus
I = moment of inertia
g = gravity, 386.4 in./sec.^{2}
ω = weight per inch (for a 3-inch shaft with .095 inch thickwall)
$\omega = .286 \times \pi/^4 \times (3^2 - 2.81^2)$
$= .248$ lbs./in.
ℓ = length of the shaft, in.

This formula assumes that the tube is a uniform beam with both ends simply supported. The lowest natural frequencies (critical speed) of the drive and propeller shafts are 3,000 rpm and 3,800 rpm, respectively. The maximum

**Thomson, William T., Theory of Vibration, Prentis Hall, 1972, p. 273.

operating speed of the shaft is 1020 rpm which is much lower than the critical speed. Therefore, the shaft is considered to be safe from a critical speed consideration.

3.4 Stiffness Analysis

The lateral stiffness of the shaft can be calculated from EI ($E = 30$ msi, $I = .9156$ in.^{4}).

$$EI = 26.6 \times 10^6 \text{ lb.-in.}^2$$

The torsional stiffness can be calculated from:

$$JG \quad (G = 11 \text{ msi}, \ J = 2I = 1.831 \text{ in.}^4)$$

$$JG = 20.1 \times 10^6 \text{ lb.-in.}^2$$

4.0 COMPOSITE TUBE DESIGN

This section presents the rational used in selecting the recommended composite shaft designs. The recommended designs for the combined-shaft and dual-shaft systems were previously presented in Figure 9 and Table II. This section first presents the considerations of resin and fiber selection. This is followed by the design analysis, including the development of a model with various laminate designs compared on the basis of weight, cost, thickness, and natural frequency. The results of the model, shown in Figures 11 through 17, were used to determine the design configuration that best met the design requirements. The baseline and alternate designs were selected based on these curves.

4.1 Considerations in Fiber Selection

The design of any composite component begins with identifying applied loads, service environment, dimensional constraints, material compatibility constraints, and performance requirements. The applied loads will dictate the composite construction. Some examples follow. Graphite is typically used to increase stiffness and reduce weight. Fiberglass has the advantages of low cost, high impact strength, ease of machining, and is relatively unaffected by sea water. The disadvantages of fiberglass are low modulus and high density. Kevlar is higher in modulus than fiberglass, has a lower weight, and offers better ballistic resistance. The disadvantages of Kevlar are very low compressive strength, marginal shear strength, and high water absorption. Kevlar is not recommended for use in torque carrying applications because of its lower strength in compression and low shear strength. Table V shows the material properties of commonly used fibers. Figure 18 presents the specific strengths and modulii of various fibers.

When designing a torque carrying structure, a ±45° fiber wind angle can carry the torque load more efficiently. The required stiffness dictates the

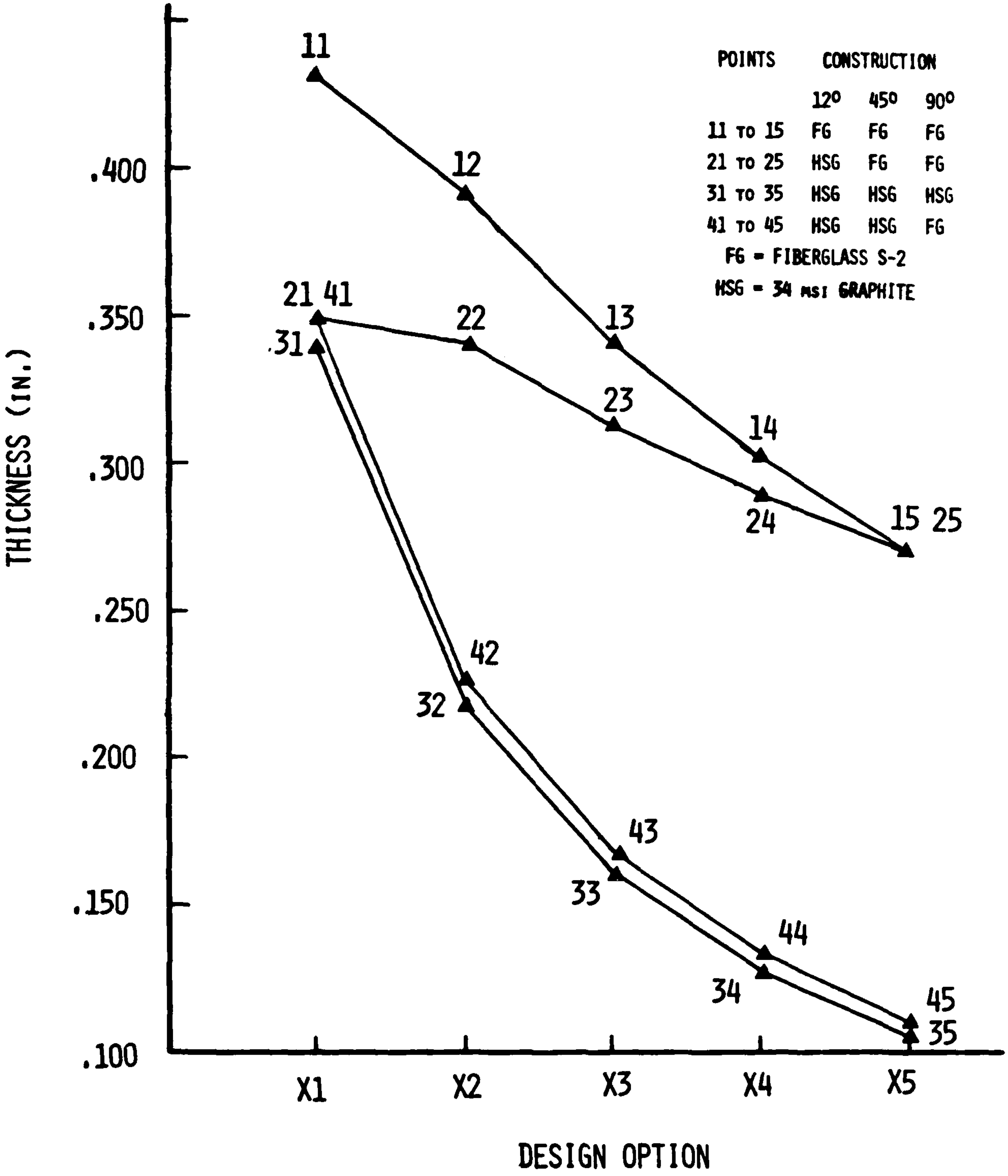


Figure 11. Design Option Versus Composite Thickness

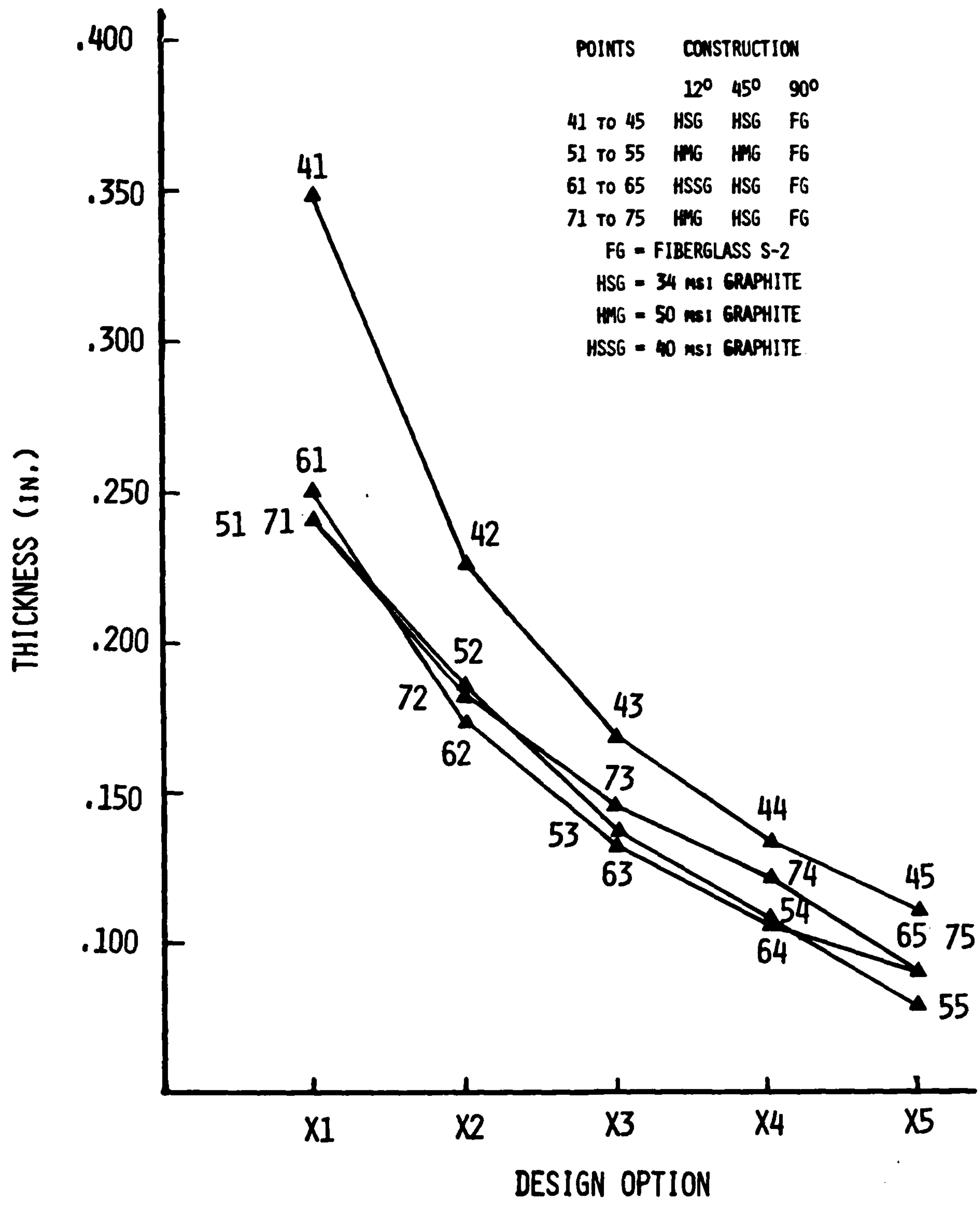


Figure 12. Design Option Versus Composite Thickness

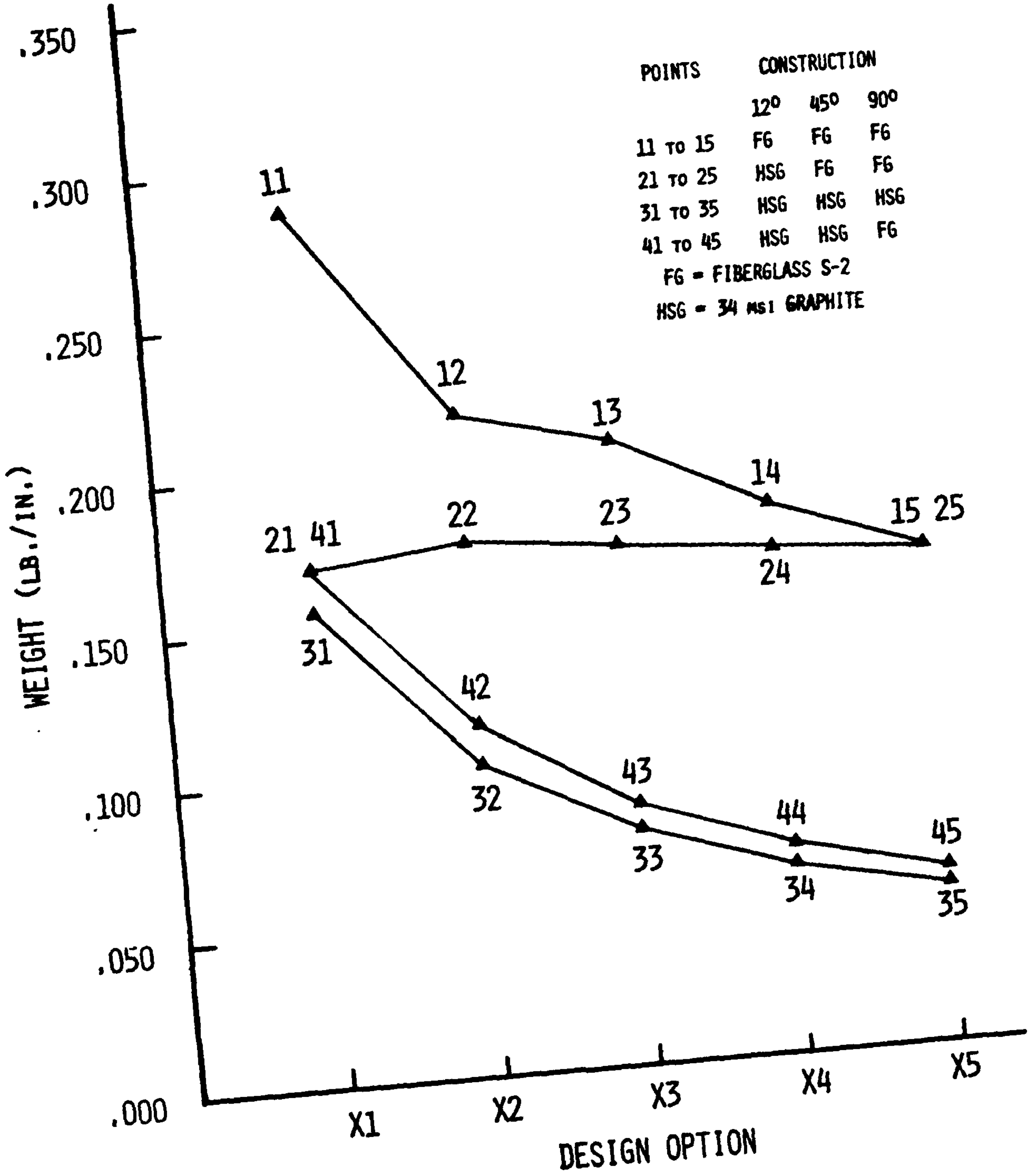


Figure 13. Design Option Versus Weight Per Inch

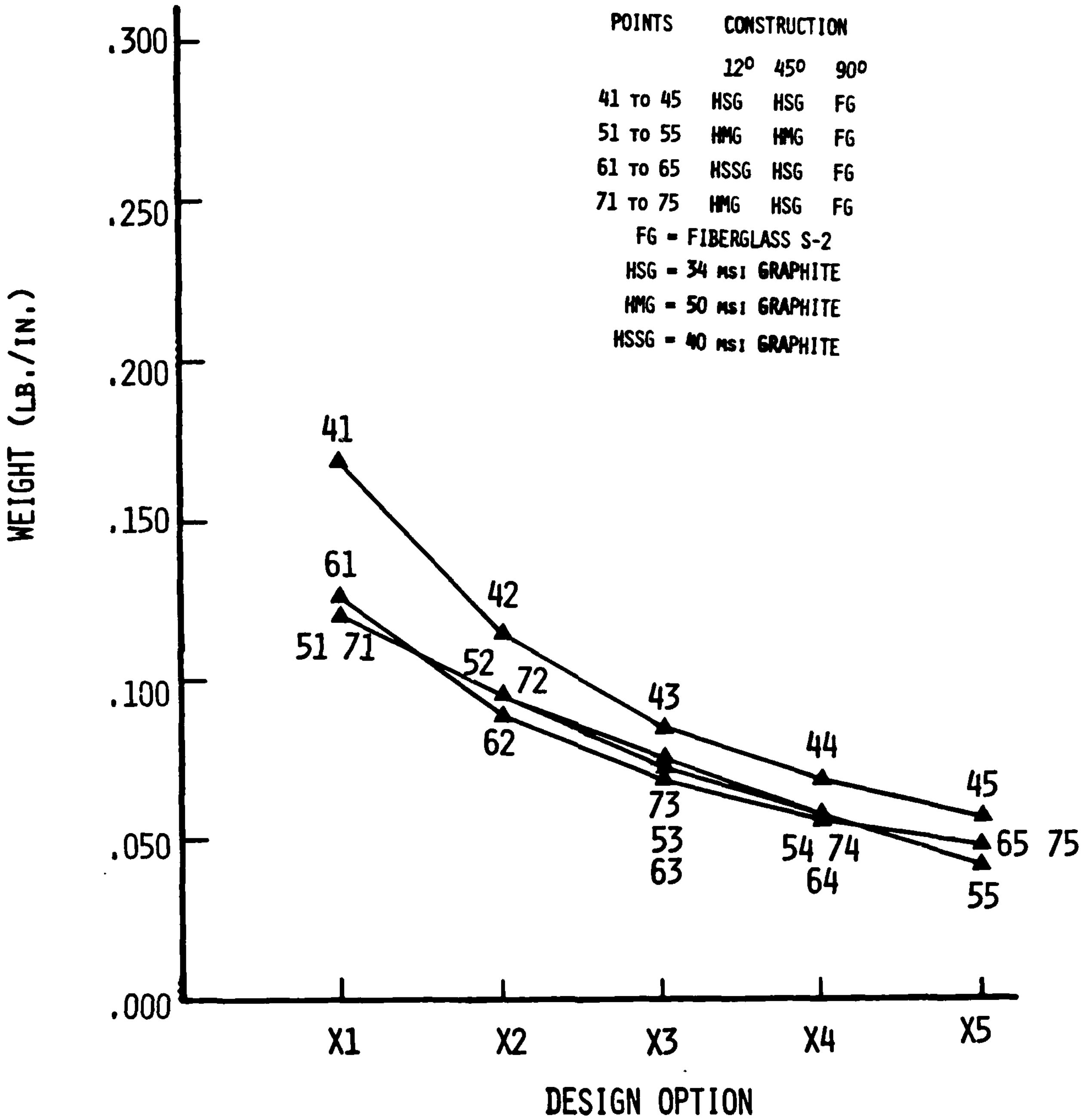


Figure 14. Design Option Versus Weight Per Inch

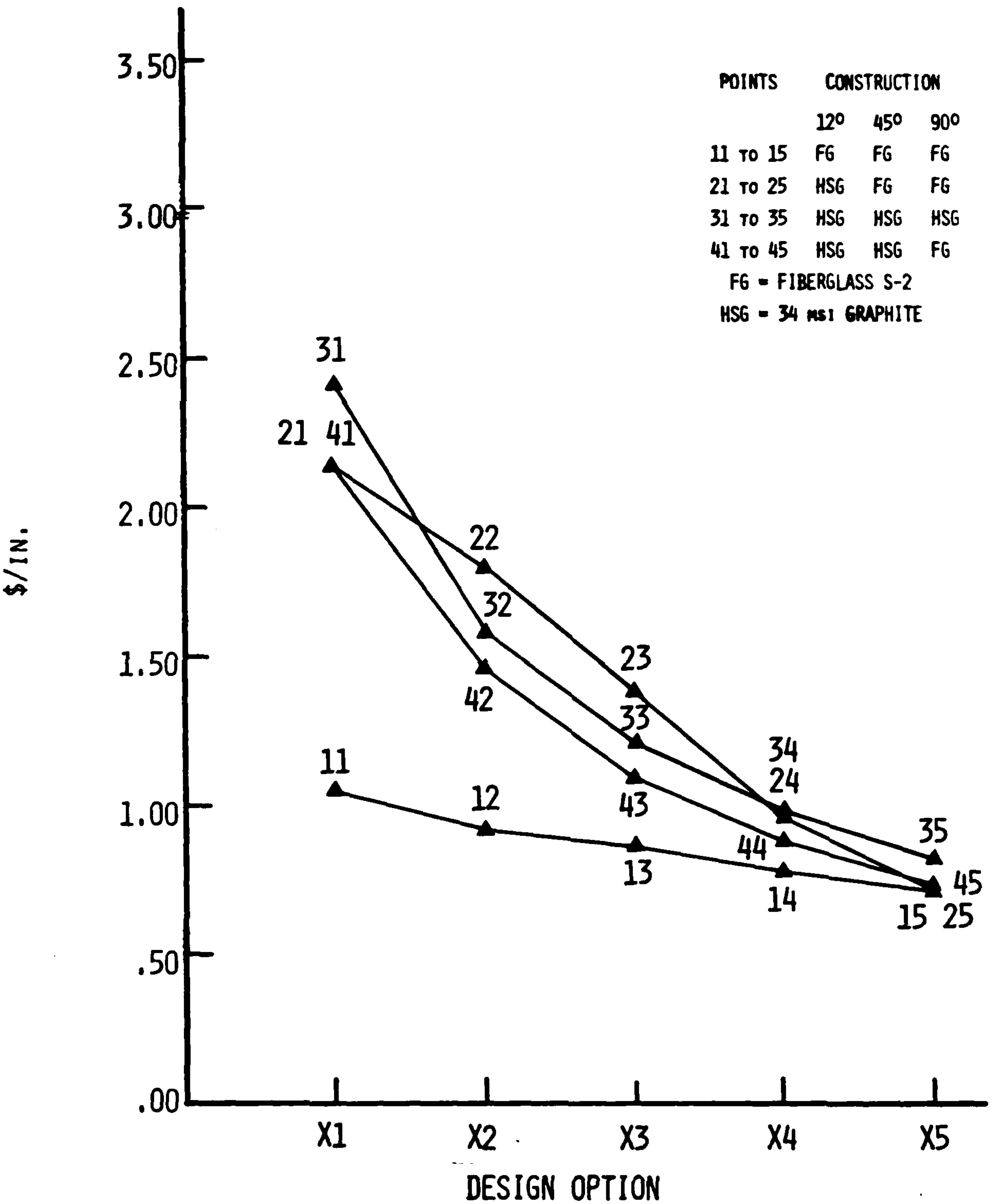


Figure 15. Design Option Versus Cost Per Inch

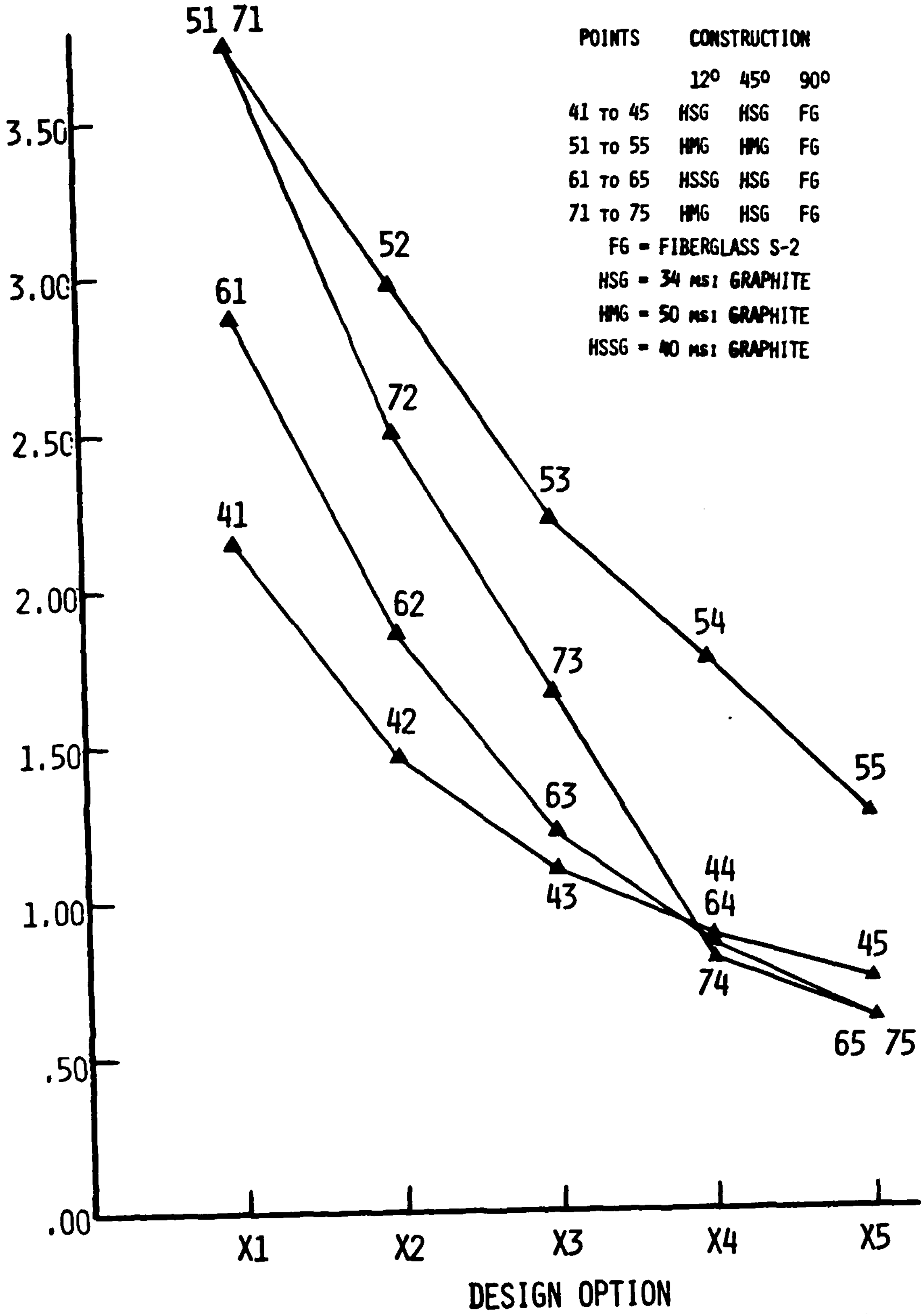


Figure 16. Design Option Versus Cost Per Inch

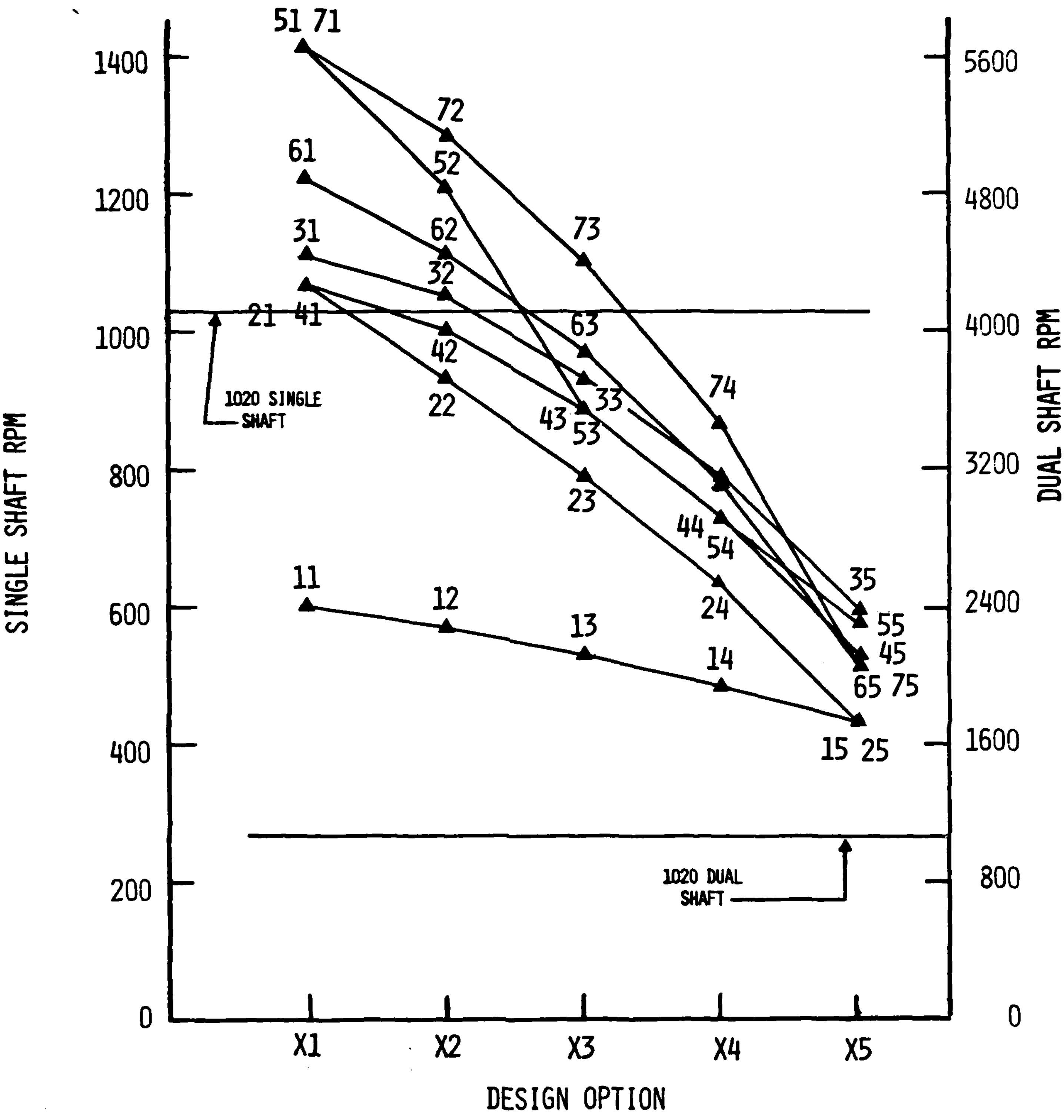


Figure 17. Design Option Versus Critical Speed

SPECIFIC* TENSILE STRENGTH AND SPECIFIC* TENSILE MODULUS OF REINFORCING FIBERS

* TENSILE STRENGTH OR MODULUS DIVIDED BY DENSITY

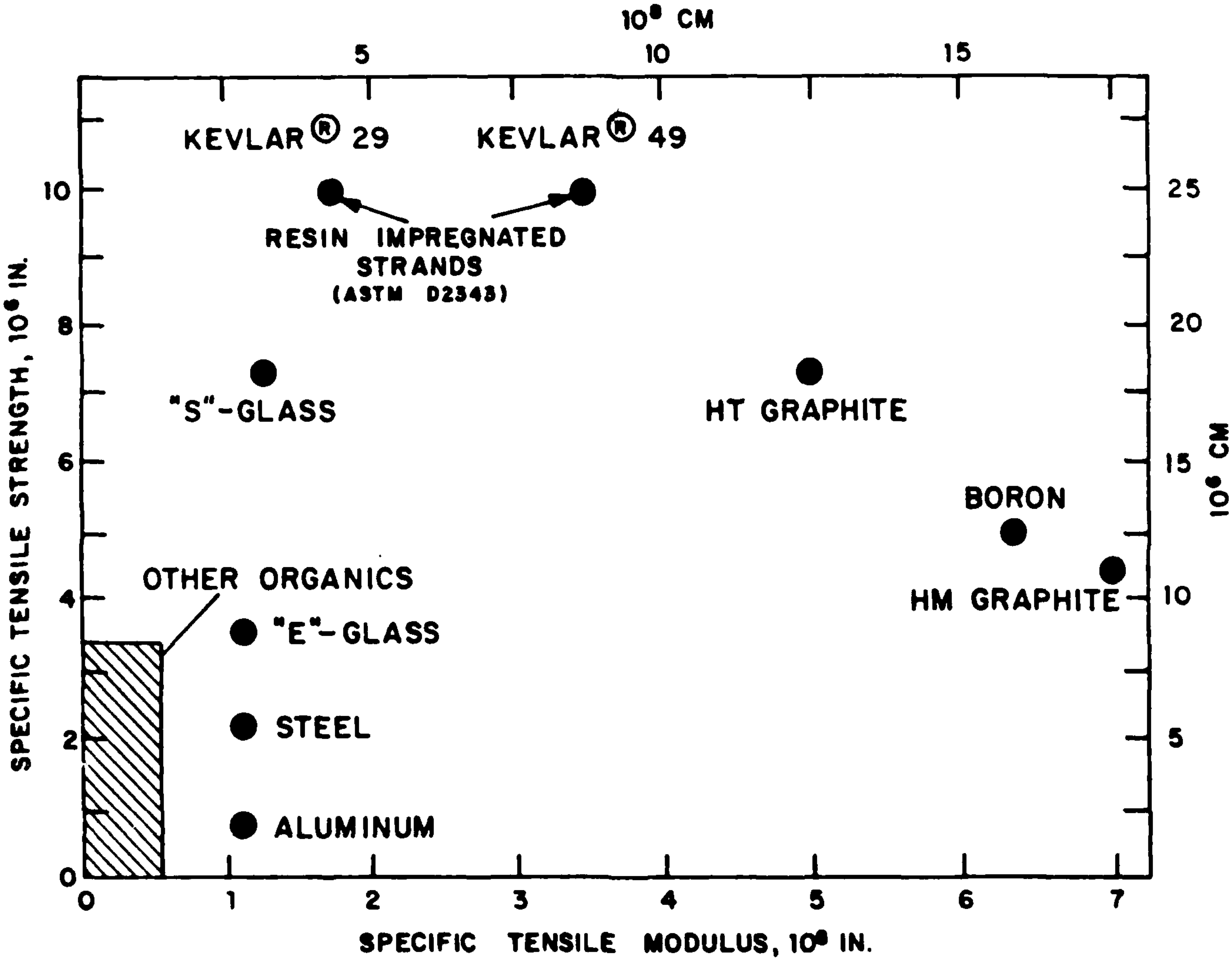


Figure 18. Specific Modulus Versus Specific Strength

Table V

FIBER PROPERTIES

	S-2 Glass[1]	34 msi Graphite[2]	40 msi Graphite[3]	50 msi Graphite[4]	Kevlar[5]
Density, lb./in.^{3}	.090	.065	.065	.065	.052
Strand Tensile Strength, 70°F, psi x 10^3	665	600	820	350	525
Modulus of Elasticity, 72°F, psi x 10^6	12.6	35	40	57	19
Coefficient of Thermal Expansion, in./in.-°F x 10^{6}	3.1	-.3	-.3	-.4	-1.1
Tensile Elongation at 72°F, %	5.4	1.7	2.0	.6	2.7

Source: Manufacturers Product Literature

[1]Owens Corning S-2 Fiberglass.
[2]Union Carbide T-600.
[3]Union Carbide T-40.
[4]Union Carbide T-50.
[5]DuPont.

msi = modulus x 10^6 psi

fiber type to be used and the percentage of low wind angle fibers needed. The smallest wind angle usually possible in filament winding is 10° off the shaft axis.

4.2 Considerations in Resin Selection

The consideration for the resin system selection is based on many factors. Some of the important factors in this design are material cost and moisture resistance in a marine environment.

Two Lincoln Resin Formulations (LRF's) that are well suited to the production of tubular structures were selected for consideration. They are LRF-092 and LRF-271. In a torsion loading condition there are two significant requirements for the resin system: first, it must be capable of withstanding the applied loads; and, second, it should have sufficiently high elongation to prevent resin crazing. For this application, there is also a third requirement, that the moisture gain be low. Shown below are the neat (pure) resin properties for LRF-092 and LRF-271.

Resin Casting Property (unaged)	LRF-092 Property Value	LRF-271 Property Value
Tensile Strength	8,480 psi	7,370 psi
Tensile Modulus	425 ksi	440 ksi
Percent Elongation	3.1	7.1
Shear Strength	7,860 psi	8,140 psi
Moisture Gain	.103%	.161%
Cost	1.28 $/lb.	1.36 $/lb.

LRF-271 is superior in these mechanical properties. However, LRF-092 was selected since it has the smallest moisture gain with satisfactory mechanical properties and it has been used for all torque shafts built at Brunswick to date.

LRF-092 was originally developed to filament wind the third stage of the Polaris III rocket motor case. At that same time, it was also used to filament wind radomes for the US Navy. The Handbook of Composites, by

George Lubin, relates how various aircraft components were tested after 20 years of service in the fleet with little degradation found from the as-fabricated properties. These parts were fabricated with a LRF-092 type resin system. LRF-092 is covered by MIL-C-47257. In addition, specifications for the components of LRF-092 are covered in Weapon Specification (WS) 21098 and the mixed resin with glass reinforcement is covered by WS 20538.

4.3 Composite Shaft Analysis

The design analysis of the composite shaft uses the classical lamination theory to predict material properties and stress/strain levels under specified load conditions. The stress/strain levels are compared against the material allowables shown in Tables VI and VII to determine design margins. The actual analysis computations are completed using computer software especially developed for solving the complex relationships that are required to describe the behavior of composite structure under specified loads. The Brunswick developed program used for the composite shaft analysis calculates the plane orthotropic material properties for composites, including moduli and Poisson's ratios of individual layers and of the total laminate. It also calculates strains in the laminate and stresses and strains in individual layers given the in-plane running loads. The theory on which this program is based is contained in the Primer on Composite Materials: Analysis, by Ashton, Halpin, and Petit.

In addition to calculation of the composite stress and strain, the natural frequency and critical buckling torque are also determined and presented in this section.

The following paragraphs examine several laminate constructions by first comparing critical speed, then thickness, weight, and cost. From these comparisons, the recommended designs for both the single and double-shaft are made. Lastly, the stress analysis for the recommended shafts is presented.

Table VI

TYPICAL UNIDIRECTIONAL PROPERTIES OF FIBER/EPOXY MATRIX
(FIBER VOLUME = .60)

Elastic Constants	S-2 Glass[2] Epoxy	Kevlar/[1] Epoxy
Longitudinal Modulus, psi x 10^6	7.7	11.6
Transverse Modulus, psi x 10^6	1.8	.8
Shear Modulus, psi x 10^6	.96	.42
Poisson's Ratio, V_{LT}	.32	.34
Strength Properties Allowables		
Longitudinal Tensile Strength, psi x 10^3	210	200
Transverse Tensile Strength, psi x 10^3	6	4.3
Longitudinal Compressive Strength, psi x 10^3	100	25
Transverse Compressive Strength, psi x 10^3	20	20
In-plane Shear Strength, psi x 10^3	9	Very Low
Ultimate Strains		
Longitudinal Tension, Percent	2.7	1.7
Transverse Tension, Percent	0.3	.54
Longitudinal Compression, Percent	1.3	0.2
Transverse Compression, Percent	1.1	2.5
Physical Properties		
Density, lb./in.^{3}	.072	.050
Longitudinal Coefficient of Thermal Expansion, in./in.-°F x 10^6	3.7	-1.1
Transverse Coefficient of Thermal Expansion, in./in.-°F x 10^6	19.5	19.5

Source: [1]Kevlar 49 Data Manual
[2]Brunswick Data Epoxy-Anhydride Resin

Table VII

TYPICAL UNIDIRECTIONAL PROPERTIES OF GRAPHITE/EPOXY MATRIX
(FIBER VOLUME = .60)

Elastic Constants	34 msi Graphite	50 msi Graphite	40 msi Graphite
Longitudinal Modulus, psi x 10^6	20.6	30.2	24.2
Transverse Modulus, psi x 10^6	1.5	1.6	1.5
Shear Modulus, psi x 10^6	.75	.80	.77
Poisson's Ratio, V_{LT}	.32	.32	.32
Strength Properties Allowables			
Longitudinal Tensile Strength, psi x 10^3	230	195	285
Transverse Tensile Strength, psi x 10^3	10	5	6
Longitudinal Compressive Strength, psi x 10^3	100	85	140
Transverse Compressive Strength, psi x 10^3	23	20	20
In-Plane Shear Strength, psi x 10^3	9	9	9
Ultimate Strains			
Longitudinal Tension, Percent	1.1	.65	1.2
Transverse Tension, Percent	.57	.29	.4
Longitudinal Compression, Percent	.48	.26	.58
Transverse Compression, Percent	1.3	1.2	1.3
Physical Properties			
Density, lb./in.^{3}	.057	.057	.057
Longitudinal Coefficient of Thermal Expansion, in./in.-°F x 10^6	-.21	-.35	-.21
Transverse Coefficient of Thermal Expansion, in./in.-°F x 10^6	19.5	19.5	19.5

Source: Calculated from fiber properties (Table IV) and resin properties of (Epoxy-Anhydride).

4.3.1 Selection of a Shaft Composite Design

The purpose of this study is to find an acceptable design given the constraints of weight, cost, design requirement, and producibility. Design criteria used in the selection study are indicated below:

1. The drive shaft tube, excluding universal joints and splines, has a 3-inch OD and 80-inch length and the propeller shaft tube has a 3-inch OD and a 60-inch length. The combined length is 160 inches.

2. The same design allowables are used to evaluate all the design options.

3. Failure occurs when the first ply fails.

4. Twenty percent of the thickness is glass hoop required for laminate compaction.

The physical constraints, such as outside diameter (O.D.) and length of the tube, and the material properties of modulus and density, dictate the calculation of the natural frequency (critical speed). Constraints such as weight, cost, impact strength, and environmental conditions, were considered during material selection. Also, from a manufacturing standpoint, compaction hoops are necessary to provide roundness control and a laminate with low void content. The theory used to size the laminate is the first ply failure theory, which states that the stresses in any ply of the composite laminate exceed the allowable for tension, compression, or shear, the whole laminate is considered to have failed. This is a conservative approach since the laminate structure has the capability of redistributing the load after the first ply fails. For a uniform and conservative approach in this particular study, the first ply failure theory served as the basis to size various design options.

The accepted method of selecting a design for a composite part is to perform a trade-off study which identifies the major factors effecting the performance of the part, then each factor is investigated and the result analyzed.

Table VIII defines the shaft laminate design combinations that were considered. Figures 11 through 17 compare these designs on a thickness, critical speed, weight, and cost basis. The thickness curves, Figures 11 and 12, were determined based on laminate stresses and the design allowables. These figures are valid for both the single and dual shaft concepts. The remaining curves, Figures 13 through 17 were then calculated from the laminate thickness curves in Figures 11 and 12. The weight and cost curves, Figures 13 through 16 again are valid for both the single and dual shaft concepts. The critical speed curve, Figure 17, shows the critical speed for a single shaft on the left ordinate and a dual shaft on the right ordinate. The dual shaft uses the 80 inch effective length of the drive shaft for the critical speed calculations and the single shaft uses the 160 inch effective length.

A review of the design curves show that all the design options for the dual shaft meet the critical speed requirement and only a few meet the critical speed requirement for the single shaft design (refer to Figure 17). These few designs for the single shaft that are above the 1020 rpm line on Figure 17 are then compared based on cost. The cost curves, Figures 15 and 16, yield design point 73 as the lowest cost point that will satisfy the design requirement. Therefore, combination point 73 is selected as the single shaft design baseline.

The design for the dual shaft was then selected based on minimum cost and weight, see Figures 13 through 16. Design option 15 was chosen because it was the lowest cost and saved 9 pounds when compared to the current steel configuration.

The specific laminate construction for these two recommended designs is shown in Table I. The ply stacking sequence was based upon sound manufacturing practices. The typical ply fiber thickness used in filament winding is .004 to .020 inches thick. The 90° plies are distributed through

Table VIII

DESIGN OPTIONS CONSIDERED (PART I)

Construction Type	Design Option	34 msi Graphite, % 12°	34 msi Graphite, % 45°	34 msi Graphite, % 90°	S-2 Glass, % 12°	S-2 Glass, % 45°	S-2 Glass, % 90°
All Glass	11				80		20
	12				60	20	20
	13				40	40	20
	14				20	60	20
	15					80	20
34 msi Graphite for Bending (12° only)	21	80					20
	22	60				20	20
	23	40				40	20
	24	20				60	20
	25					80	20
34 msi All Graphite	31	80		20			
	32	60	20	20			
	33	40	40	20			
	34	20	60	20			
	35		80	20			
34 msi Graphite for Bending and Torque	41	80					20
	42	60	20				20
	43	40	40				20
	44	20	60				20
	45		80				20

Table VIII, Cont'd.

DESIGN OPTIONS CONSIDRED (PART II)

Construction Type	Design Option	34 msi Graphite, %		40 msi Graphite, %	50 msi Graphite, %		S-2 Glass, %
		12°	45°	12°	12°	45°	90°
34 msi	41	80					20
Graphite	42	60	20				20
for Bending	43	40	40				20
and Torque	44	20	60				20
	45						20
50 msi	51				80		20
Graphite	52				60	20	20
for Bending	53				40	40	20
and Torque	54				20	60	20
	55					80	20
34 msi	61			80			20
Graphite	62		20	60			20
for Torque	63		40	40			20
40 msi	64		60	20			20
Graphite	65		80				20
for Bending							
34 msi	71				80		20
Graphite	72		20		60		20
for Torque	73		40		40		20
50 msi	74		60		20		20
Graphite	75		80				20
for Bending							

the laminate for compaction and roundness control. In addition, the goal is to have a symmetrical laminate with a low angle helix (12°) near the center of the laminate. This provides stability to the low angle helix.

The stress analysis for these two design configurations is shown in Table IX. Note that the laminate is adequately designed to carry all loads. The design allowables used in the stress analysis were originally presented in Tables V and VI. The critical speed and critical buckling torque for both designs are also computed and presented in Table IX. The critical speed equation was previously presented. The critical buckling torque is calculated by the following equation:

$$T_{CR} = 1.688 / \pi L \ E_1^{.375} E_2^{.625} t^{2.25} D^{1.25}$$

Where,

E_1 = longitudinal modulus, psi
E_2 = transverse modulus, psi
t = thickness, in.
D = diameter, in.
T_{CR} = critical buckling torque, in.-lbs.

Table IX

LAMINATE STRESSES AND DESIGN FACTORS

Baseline Combined Shaft 160 inches Long (Design Option 73)
34 msi Modulus Graphite at 45°, 50 msi Modulus Graphite at 12°

Material	34 msi Graphite	50 msi Graphite	S-2 Fiberglass
Wind Angle	45°	12°	90°
Thickness Composite Wall Center	.060	.060	.030
Long. Tension/Allowable	75/230	51/195	0/210
Long. Compression/Allowable	75/100	51/85	0/75
Trans. Tension/Allowable	3.7/5	1.6/5	0/5
Trans. Compression/Allowable	3.7/23	1.6/20	0/30
Shear Allowable	0/8.5	5/7	7/10

Critical Torque = 3650 ft.-lb. Critical Speed = 1110 rpm
Total Thickness = .150 inches

Baseline Dual Shafts 80 inches Long (Design Option 15)
All S-2 Fiberglass

Material	S-2 Fiberglass	S-2 Fiberglass
Wind Angle	45°	90°
Thickness Composite Wall Center	.216	.054
Long. Tension/Allowable	18/210	0/210
Long. Compression/Allowable	18/75	0/75
Trans. Tension/Allowable	3/5	0/5
Trans. Compression/Allowable	3/30	0/30
Shear Allowable	10/10	11/10*

Critical Torque = 138,000 ft.-lb. Critical Speed = 3360 rpm
Total Thickness = .270 inches

*The allowable is exceeded, but this is a noncritical layer. The stress can be reduced below the allowable by adding .010 inch of 90° glass material.

5.0 JOINT DESIGN FOR COMPOSITE SHAFT

This section addresses the interface between the composite tube and the existing drive train. Three approaches were identified and analyzed for an optimum joint attachment. A riveted joint was selected and fully analyzed. The rivet selection made was a high strength 1/4 inch diameter, 5/8 inch long aerospace grade rivet. The composite construction was analyzed and a reinforced laminate design defined for the grip area. The metal shaft coupling was analyzed for strength and fatigue.

5.1 Brunswick's Past Experience

In the past few years, Brunswick has been awarded contracts from Boeing Vertol, Naval Ships Research and Development Center (NSRDC), and Litton-Ingalls Shipbuilding to develop composite shaft technology.

The most critical component in the shaft is the joint design. Brunswick has conducted several R & D projects to evaluate various joint concepts and achieved certain success. The detailed progress of these development programs is presented in Appendix A.

5.2 Joint Concepts Under Study

There were three types of joint concepts studied under this program. Each concept has good potential for future development. The following sections present the design concepts and brief stress analyses of these three concepts.

5.2.1 Riveted Joint

This joint was designed having a metal shaft press fit inside a locally reinforced composite shaft. There are two rows of 10 rivets installed on the joint at 36° apart. The analysis of this joint design was divided into two major areas. The first one is the stress concentration area due to the fillet radius on the metal shaft coupling. The second area is the rivet and composite bearing stress analysis under the maximum required loading condition. The riveted joint has been used in several applications and is a proven concept.

5.2.2 Interference Fit Joint

Another potential means of transferring torque between a metal coupling and composite tube is through friction at the tube/coupling interface. This should be particularly effective for tubes with relatively low torque requirements, such as the drive shaft for the amphibious armored personnel carrier.

The torque level which must be carried at this interface may be calculated as follows:

$$N_\tau = \frac{2T_{max}}{\pi D^2} = \frac{2(40,560)}{\pi (2.78)^2}$$

$$N_\tau = 3,341 \text{ lbs./in.}$$

where,

N_τ = Shear force per circumferential inch of interface

T_{max} = Maximum torque requirement = 3,380 ft.-lb.
= 40,560 in.-lb.

D = Diameter of interface

The average shear stress, τ_{ave}, at the interface then becomes:

$$\tau_{ave} = \frac{N_\tau}{\ell} = \frac{3,341 \text{ lb./in.}}{\ell}$$

where,

ℓ = Effective axial length of joint interface

Therefore, the average shear stress as a function of length is as shown below. These shear stress levels are within the realm of frictional load transfer.

EFFECT OF JOINT LENGTH ON SHEAR STRESS

ℓ, in.	τ_{ave}, psi
1	3,341
2	1,670
3	1,114
4	835
5	668

The basis of the concept is that the coupling should be capable of being pressurized after it is installed in its desired location at the end of the tube. This concept is depicted in Figure 19. During pressurization, the wall of the coupling would be stressed beyond its yield point and deformed into contact with the tube wall. Additional forming pressure would then further deform the coupling wall plastically while the tube wall dilated elastically. After the pressure was removed, a substantial residual interface pressure would exist between the coupling and the tube.

The primary disadvantage of this concept is the potential for relaxation of the interface pressure after prolonged periods of time due to loss of interface pressure or relaxation due to creep.

5.2.3 Hex Joint

This joint design concept consists of a hexagonal shape that acts to transmit the torque to and from the composite shaft in the form of compressive (FWD END) or tensile (AFT END) load in the hoop direction. A sketch of the shaft cross section is shown in Figure 20.

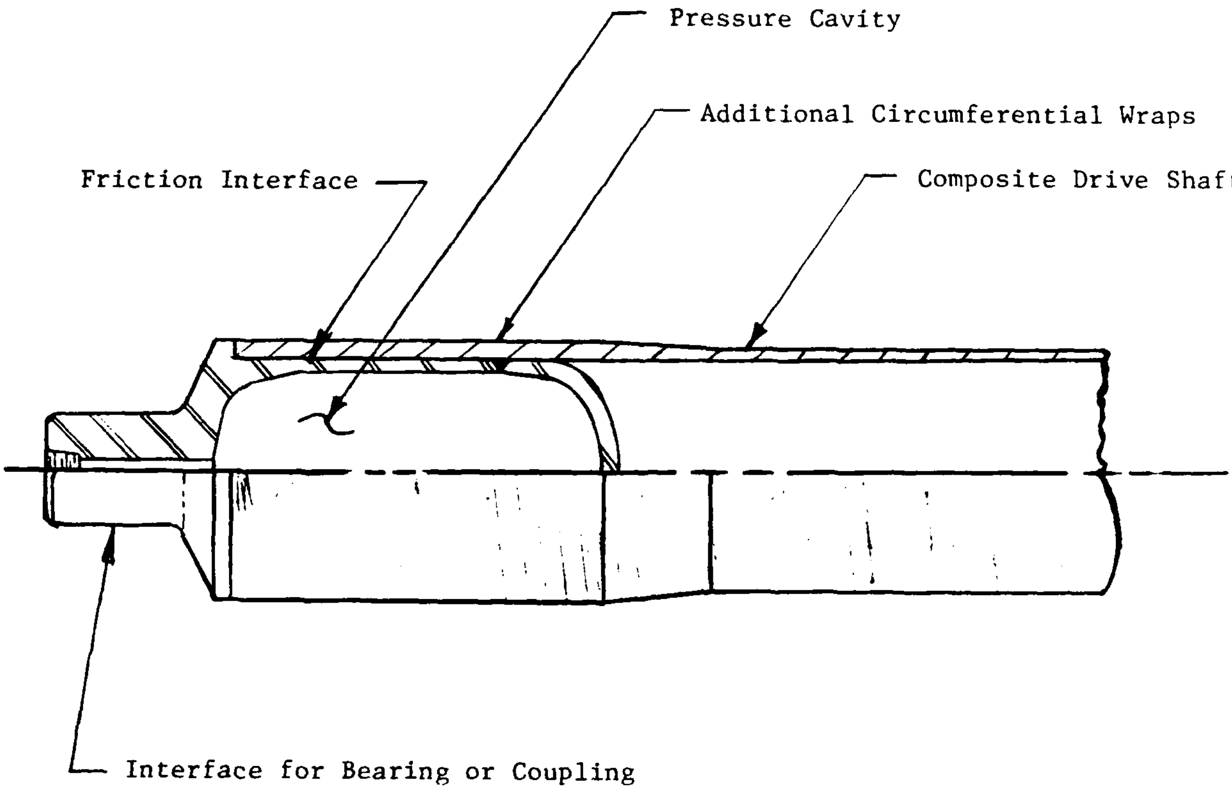


Figure 19. Interference Fit Coupling

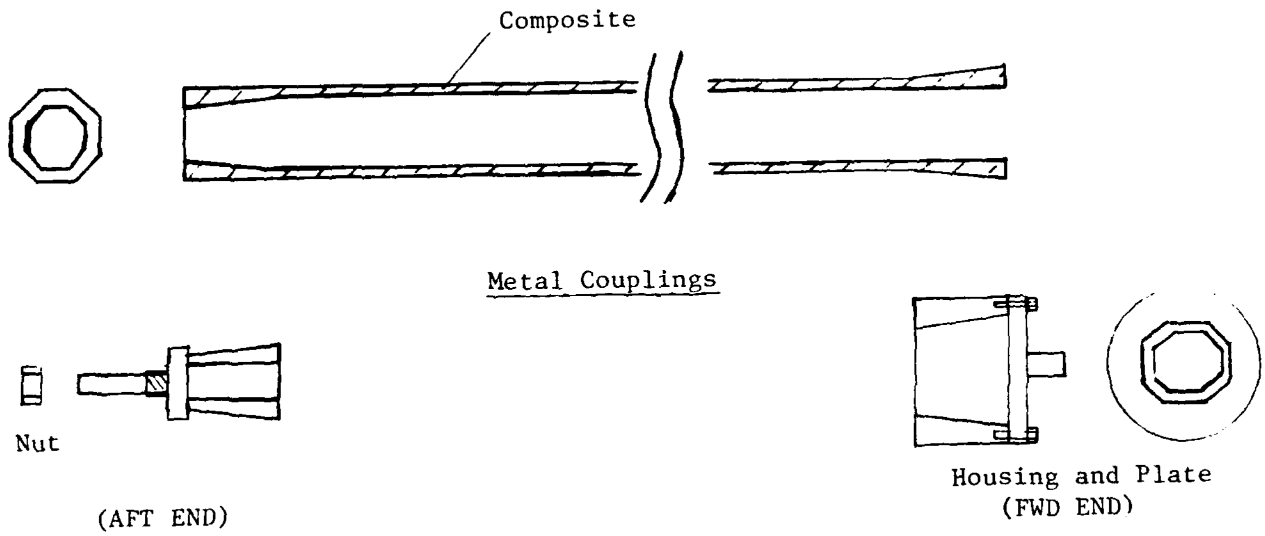


Figure 20. Hex Joint

5.2.4 Joint Selected

A riveted joint design is recommended and is illustrated in Figure 21, since it can be reliably shown to satisfy all the requirements and is manufacturable using current technology.

The interference and hex joint concepts also demonstrate potential to be efficient joint designs. However, both concepts are more expensive to produce and provide no significant improvement over the riveted joint.

5.3 Analysis of Riveted Joint

The following paragraphs describe the detailed riveted joint design. The proper selection of the rivet requires that the bearing strength of the rivet and joined materials be sufficient to support the loading.

5.3.1 Metal Shaft Coupling Stress Concentration Area

The following dimensions were obtained from Figure 21:

$$r = .090'', \qquad D = 3.08'', \qquad d = 2.78'', \qquad d_1 = 2.38$$

$$r/d = .090/2.78 = 0.03237$$

$$D/d = 3.00/2.78 = 1.079$$

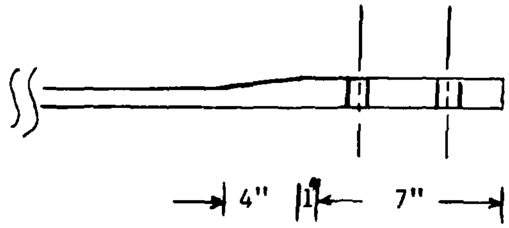
From Figure 22, the stress concentration factor is $k_{tso} = 1.6$ for a solid shaft. The modification factor to a hollow shaft is shown in Figure 23.

$$d_i/d = 2.38/2.78 = 0.856$$

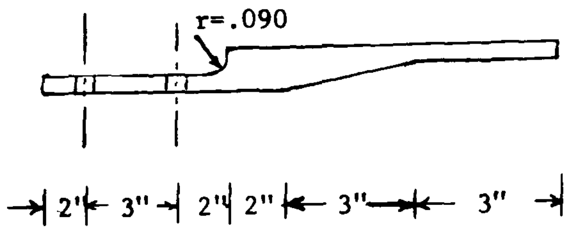
$$(k_{ts} - 1)/(k_{tso} - 1) = 0.55, \quad k_{ts} = 1 + 0.55 \times (k_{tso} - 1)$$

$$= 1 + 0.55 \times (1.6 - 1) = 1.33$$

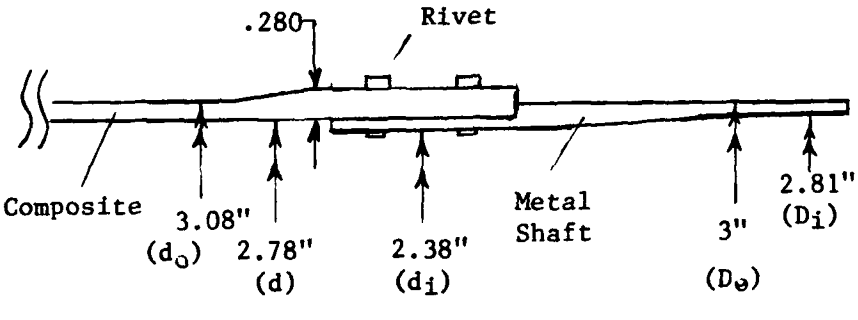
Therefore, the stress concentration factor for a hollow shaft with a fillet radius is $k_{ts} = 1.33$.



composite shaft joint end



metal shaft joint end



assembled joint

Figure 21. Joint Design

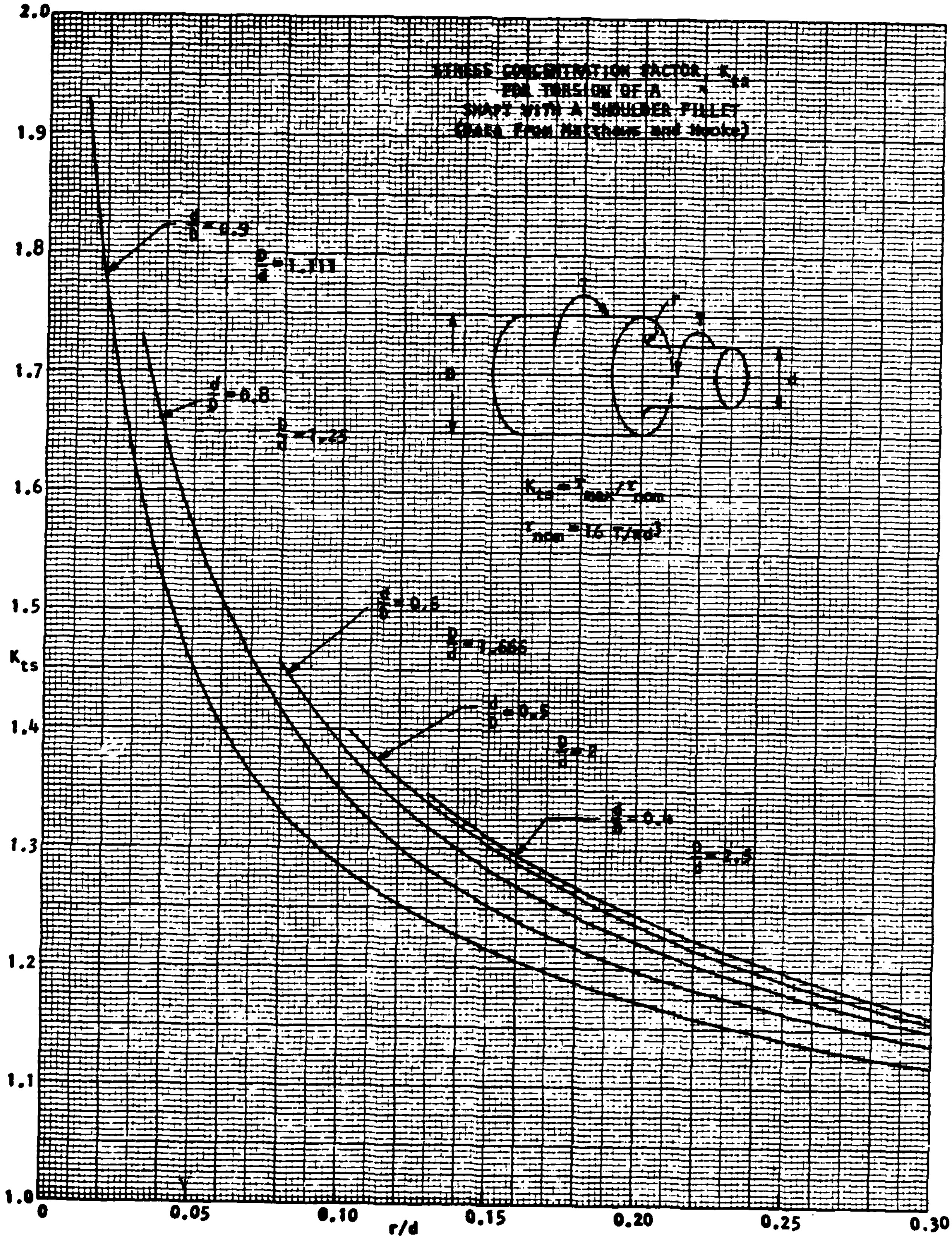


Figure 22. Stress Concentration Factor - Solid Shaft

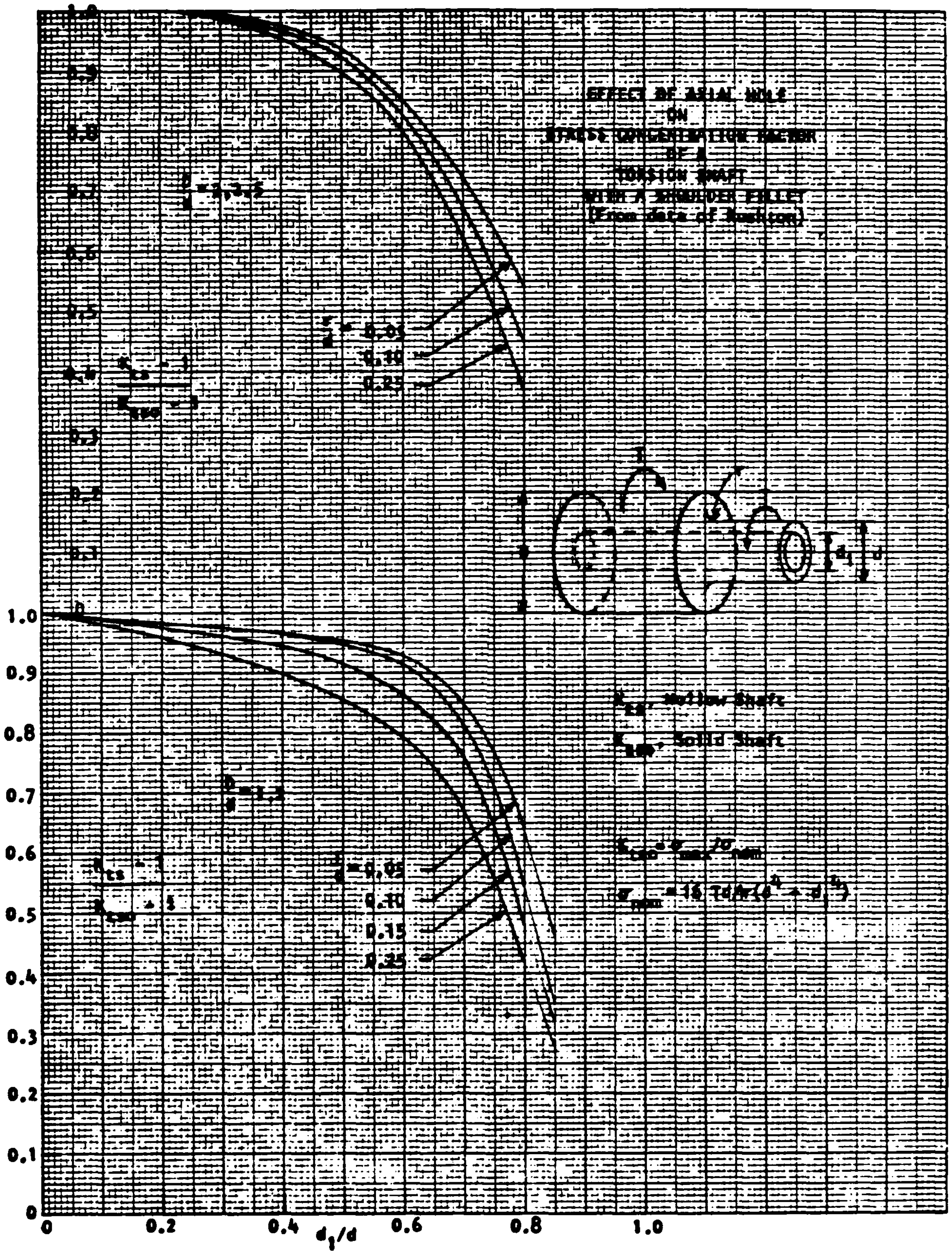


Figure 23. Stress Concentration Factor - Hollow Shaft

The effective torsional rigidity at the stress concentration area is G Jo, while the original shaft is assumed to be G J.

$$Jo = 1/32\ \pi\ (d^4 - d_i^4)/k_{ts} = 1/32\ \pi\ (2.78^4 - 2.38^4)/1.33 = 2.04$$

$$J = 1/32\ \pi\ (D_o^4 - D_i^4) = 1/32\ \pi\ (3^4 - 2.81^4) = 1.8311$$

Therefore, the torsional safety factor at the stress concentration area over the original shaft is SF = Jo/J = 2.04/1.83 = 1.115.

5.3.2 Composite Bearing Stress Analysis

The following paragraphs contain the analysis for the composite bearing stress. Listed below are the terms used in the analysis.

t_c = thickness of composite
T = total torque
T_A = torque transferred through fastener A
T_B = torque transferred through fastener B
r_s = torsional deformation of steel shaft between A and B
r_c = torsional deformation of composite shaft between A and B
G_c = shear modulus of composite shaft at the joint
G_s = shear modulus of steel shaft
J_c = polar moment of inertia of the composite shaft at the joint
J_s = polar moment of inertia of the steel shaft at the joint
N = number of fastener(s)
F_A = force at fastener(s) A
F_B = force at fastener(s) B
R = mean radius
D_b = fastener(s) diameter
σ_{CA} = composite bearing stress at A
σ_{CB} = composite bearing stress at B
σ_{SA} = steel bearing stress at A
σ_{SB} = steel bearing stress at B
ℓ = shaft length

$$r_s = \int_0^{\ell} \frac{T_A\, dx}{G_s\, J_s} \quad (1) \qquad\qquad r_c = \frac{T_B \ell}{G_c\, J_c} \quad (2)$$

Physical Constraint: $r_s = r_c$ (3)

Summation of Reactant Forces from Two Rows of Rivets: $T_A + T_B = T$ (4)

Rearranging (1) and (2): $T_A/T_B = [\ell/(G_c\, J_c)]\{1/[\int_0^{\ell} 1/(G_s\, J_s)\, dx]\}$ (5)

Combining (4) and (5) Yields (6) and (7): $T_A = T/\{1 + (G_c\, J_c/\ell)[\int_0^{\ell} 1/(G_s\, J_s)\, dx]\}$ (6)

$$T_B = T \frac{\{[(G_c\, J_c)/\ell]\}\, \{[\int_0^{\ell} 1/(G_s\, J_s)\, dx]\}}{1 + \{[(G_c\, J_c)/\ell]\}\, \{[\int_0^{\ell} 1/(G_s\, J_s)\, dx]\}} \quad (7)$$

Calculation of individual forces on fasteners:

$$F_A = T_A\, R/N \quad (8)$$

$$F_B = T_B\, R/N \quad (9)$$

Resultant bearing stresses:

$$\sigma_{CA} = F_A\, D_b/t_c \quad (10)$$

$$\sigma_{CB} = F_B\, D_b/t_c \quad (11)$$

$$\sigma_{SA} = \frac{F_A}{t_s\, D_b}, \quad \sigma_{SB} = \frac{F_B}{t_s\, D_b} \quad (12)$$

The optimum joint design is for all the rivets carrying the same load. That is $T_A = T_B = T/2$ and $G_c J_c = G_s J_s$ (assuming the steel shaft is not tapered).

Since the composite modulus is relatively low in comparison to steel, reinforcements for the composite shaft at the joint are necessary. The detailed design of the joint was presented in Figure 21. The wind sequence and general laminate information is shown in Figure 24. With all the design information gathered, the following calculation demonstrates that $G_c J_c$ is roughly equal to $G_s J_s$.

$$G_c = 4.994\ 10^6 \text{ psi},\ J_c = 1/32\ \pi\ (3.34^4 - 2.78^4) = 6.354$$

$$G_s = 11.538\ 10^6 \text{ psi},\ J_s = 1/32\ \pi\ (2.78^4 - 2.38^4) = 2.714$$

$$G_c J_c = 4.984\ 10^6 \times 6.354 = 32.667 \times 10^6$$

$$G_s J_s = 11.538\ 10^6 \times 2.714 = 31.314 \times 10^6$$

For simplicity, assuming the load is evenly distributed among the rivets, the composite bearing stress σ_c can be calculated as shown below:

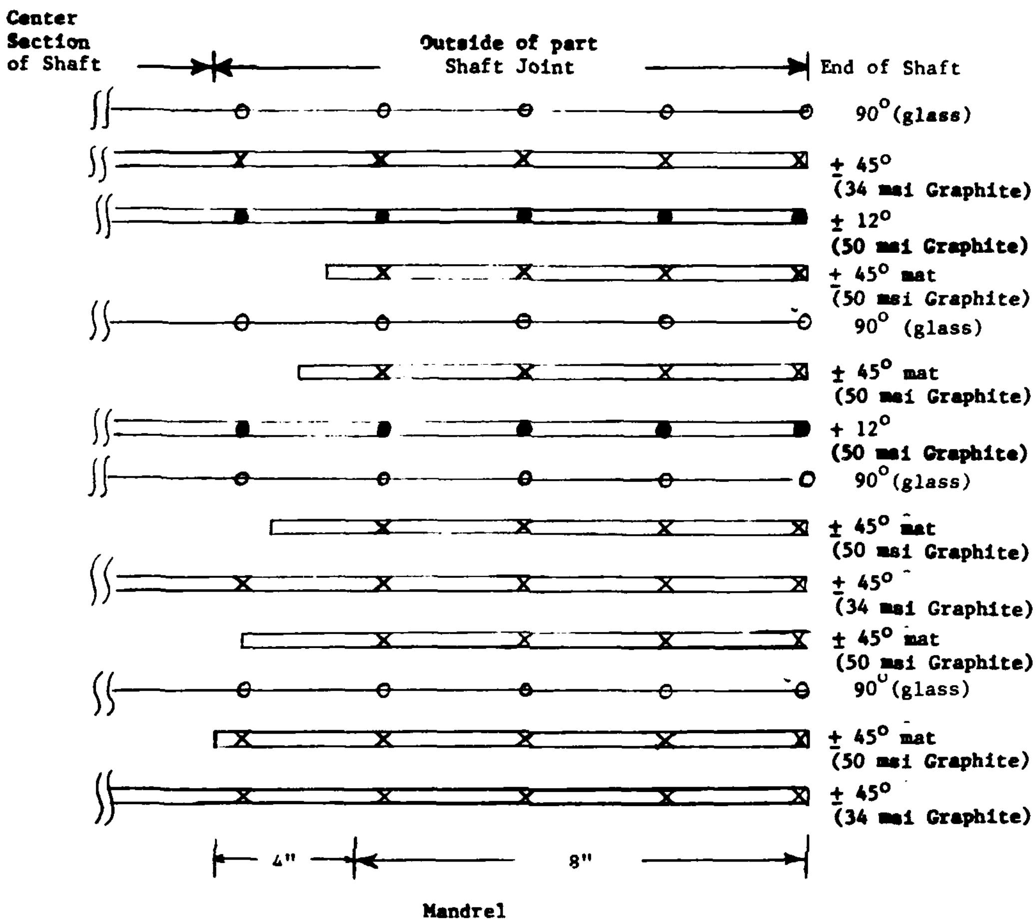
$$\sigma_c = (T/RN)/(D_b \times t_c) = (3380 \times 12/3.0/20)/(.25 \times .280)$$

$$= 9656 \text{ psi (average bearing allowable is 40 ksi)}$$

5.3.3 Rivet Stress Analysis

The selected rivet was Cherrylock Rivet NAS 1398, Universal Head, with 1/4 Diameter Dash No. 8-7. The detailed dimension is shown in Figure 25. For the A286 CRES rivet, the ultimate shear strength (S_{ult}) is 95,000 psi, as shown in Figure 26. The total rivet cross section area is:

$$A_s = \pi\ (.178)^2/4 \times 20 = .49769 \text{ in.}^2$$



NOTES: 1. Band density for 90° S-2 glass is 233.1 ends/in/ply; for ± 45° 34 msi graphite is 9.077 ends/in/ply; for ± 12° 50 msi graphite is 9.441 ends/in/ply.

2. Band density for ± 45° 50 msi graphite mat is 9.441 ends/in/ply.

3. Composite thickness.

	Center Section of Shaft	Shaft Joint
S-2 Glass + 90°	.030	.030
34 msi Graphite ± 45°	.075	.075
50 msi Graphite ± 12°	.045	.045
50 msi Graphite ± 45°		.130
Total Thickness	.150	.280

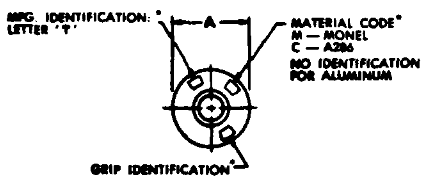
Figure 24. Wind Sequence for Combined Shaft

CHERRYLOCK® RIVETS

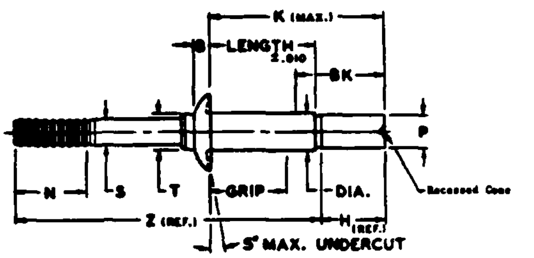
U.S. PAT. 2,931,532

NAS 1398 UNIVERSAL HEAD

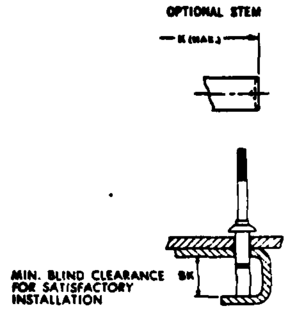
PROCUREMENT SPECIFICATION NAS 1400 IS APPLICABLE TO NAS 1398 RIVETS.



*NO HEAD MARKING ON -3 DIAMETER RIVETS



DIA. DASH NO.		-3(2)	-4	-5	-6	-8
DIA. +.003/-.001		.094	.125	.156	.187	.250
A		.187±.009	.250±.012	.312±.016	.375±.019	.500±.025
B +.010/-.000		.040	.054	.067	.080	.107
N (MIN.)		.375	.375	.375	.375	.375
S ±.003		.069	.090	.112	.132	.178
T (REF.)		.089	.119	.149	.174	.232
P ±.007		.080±.004	.111	.139	.164	.219
Z (REF.)	A GROUP	1.78	1.79	1.81	1.85	1.97
	B GROUP			2.06	2.10	2.22
	C GROUP					2.47



Rivet head may be above sheet before installation

GRIP LIMITS 1/16" RANGE MIN.	MAX.	RIVET GROUP	3/32 DIAMETER(2) DASH No.	LGTH	H	K	BK	1/8 DIAMETER DASH No.	LGTH	H	K	BK
(1)	.062	A	3-1	.156	.131	.33	.22	4-1	.188	.147	.39	.30
.063	.125	A	3-2	.219	.172	.43	.26	4-2	.250	.203	.51	.35
.126	.187	A	3-3	.281	.211	.53	.30	4-3	.313	.259	.63	.41
.188	.250	A	3-4	.344	.254	.63	.34	4-4	.375	.315	.75	.46
.251	.312	A						4-5	.438	.371	.87	.52
.313	.375	A						4-6	.500	.427	.99	.57
.376	.437	A						4-7(3)	.563	.483	1.11	.63
.438	.500	A						4-8(3)	.626	.549	1.23	.68
.501	.562	B										
.563	.625	B										
.626	.687	B										
.688	.750	B										
.751	.812	C										
.813	.875	C										

GRIP LIMITS 1/16" RANGE MIN.	MAX.	5/32 DIAMETER DASH No.	LGTH	H	K	BK	3/16 DIAMETER DASH No.	LGTH	H	K	BK	1/4 DIAMETER DASH No.	LGTH	H	K	BK	
(1)	.062	5-1	.188	.127	.36	.28	6-1	.188	.107	.37	.27						
.063	.125	5-2	.250	.183	.48	.34	6-2	.250	.163	.48	.32	8-2	.313	.179	.57	.37	
.126	.187	5-3	.313	.239	.61	.39	6-3	.313	.219	.60	.37	8-3	.375	.235	.69	.43	
.188	.250	5-4	.375	.295	.73	.45	6-4	.375	.275	.72	.43	8-4	.438	.291	.81	.48	
.251	.312	5-5	.438	.351	.85	.50	6-5	.438	.331	.84	.48	8-5	.500	.347	.93	.54	
.313	.375	5-6	.500	.407	.97	.56	6-6	.500	.387	.96	.54	8-6	.563	.403	1.04	.60	
.376	.437	5-7	.563	.463	1.09	.62	6-7	.563	.443	1.08	.60	8-7	.625	.459	1.16	.65	←Rivet Chosen
.438	.500	5-8	.625	.519	1.20	.67	6-8	.625	.499	1.20	.65	8-8	.688	.515	1.28	.71	
.501	.562	5-9(4)	.688	.628	1.37	.78	6-9	.688	.610	1.37	.76	8-9	.750	.630	1.46	.82	
.563	.625	5-10(4)	.750	.686	1.54	.84	6-10	.750	.670	1.49	.82	8-10	.813	.690	1.58	.88	
.626	.687						6-11	.813	.730	1.61	.88	8-11	.875	.750	1.70	.94	
.688	.750						6-12	.875	.790	1.74	.94	8-12	.938	.810	1.83	1.00	
.751	.812											8-13	1.000	.870	1.95	1.06	
.813	.875											8-14	1.063	.930	2.07	1.12	

(1)

RIVET DIA.	MIN. GRIP
3/32	.020
1/8	.025
5/32	.031
3/16	.037

NOTES:

(2) 3/32" diameter available in A-286 only. CR 2663 not covered by NAS 1398
(3) Not covered by NAS1398
(4) Only covered by NAS1398 for A-286 and Monel

RIVET GROUP REFERS TO SHIFT-POINT SETTING OF RIVETER.

Figure 25. Cherrylock Rivets

MATERIALS

Cherrylock rivets are manufactured in a wide variety of materials in order to give the user the widest possible choice for optimum design. Each was developed for a particular functional use as shown below:

CHERRYLOCK

RIVET SLEEVE MATERIAL	RIVET ULTIMATE SHEAR STRENGTH (Room Temp.)	TYPICAL MAXIMUM TEMPERATURE USE, °F.	
5056	30,000 psi	250	
2017	38,000 psi	250	
Monel	55,000 psi	900	
A286 CRES (NAS1398/99 style)	75,000 psi	1200	
A286 CRES *	95,000 psi	1200	← Rivet Chosen

The technical facilities of Cherry Fasteners are available to investigate the use of materials in addition to those shown.

*95KSI fastener for use in high bearing strength material, steel, CRES, Ti, etc.

STRENGTH

MINIMUM RIVET SHEAR & TENSILE STRENGTH (lbs.) IN STEEL COUPON

WIREDRAW CHERRYLOCK RIVETS			SINGLE SHEAR PER NAS1400										TENSILE PER MIL-STD-1312 TEST 8			
			ALUMINUM				MONEL		CRES				ALUMINUM	MONEL	CRES	
Rivet Diam. & Grip		Sheet Thickness	2162 2164	2163	2262	2263	2562 2564	2563	2642 *	2643 *	2652 2662 2664	2653 2663	2162 2163 2164 2262 2263	2562 2563 2564	2642 2643 2652 2653 2662 2663 2664	
−3	−2	2 x .062									453(4)	543(4)			330(4)	
	−3	2 x .093									543(4)	543(4)				
	−1	2 x .031		302		238		435		810		590				
−4	−2	2 x .062	312	494	245	388	450	710	700	1215	610	970	230	340	640	
	−3	2 x .093	450	494	355	388	640	710	1230	1230	870	970				
	−4	2 x .125	494	494	388	388	710	710	1230	1230	970	970				
	−1	2 x .031		410		325		595		1115		810				
	−2	2 x .062		645		510		930		1620		1270				
−5	−3	2 x .093	620	755	490	596	880	1090	1450	1885	1230	1490	375	550	1000	
	−4	2 x .125	710	755	550	596	1000	1090	1885	1885	1350	1490				
	−5	2 x .156	755	755	596	596	1090	1090	1885	1885	1490	1490				
	−2	2 x .062		810		640		1180		2085		1800				
	−3	2 x .093	710	1090	560	862	1030	1580	1550	2720	1400	2150				
−6	−4	2 x .125	930	1090	730	862	1310	1580	2385	2720	1800	2150	540	780	1500	
	−5	2 x .156	1020	1090	820	862	1490	1580	2720	2720	2000	2150				
	−6	2 x .187	1090	1090	862	862	1580	1580	2720	2720	2150	2150				
	−2	2 x .062		1180		930		1700		3200		2330				
	−3	2 x .093		1580		1240		2280		4000		3120				
	−4	2 x .125	1250	1970	980	1550	1800	2840	2740	4830	2450	3890				
−8	−5	2 x .156	1610	1970	1280	1550	2300	2840	3870	4910	3200	3890	1000	1450	2700	
	−6	2 x .187	1780	1970	1400	1550	2550	2840	4910	4910	3450	3890				
	−7	2 x .219	1920	1970	1520	1550	2800	2840	4910	4910	3700	3890				← Rivet Chosen
	−8	2 x .250	1970	1970	1550	1550	2840	2840	4910	4910	3890	3890				

Note:
1. Values shown are fastener capabilities only.
2. Consult Mil-Hdbk-5 for joint design allowables.
3. For rivet grips greater than listed, use highest value shown for the basic part number and diameter.
4. 3/32" diameter fasteners available in 2662 & 2663 only.

* 95KSI fastener for use in high bearing strength material, steel, CRES, Ti, etc.

Figure 26. Rivet Strength

Since the maximum torque requirement is 3,380 ft.-lb., the average shear stress, S_{avg} in the rivet can be obtained as shown below:

$$S_{avg} \times A_s \times (d1)/2 = 3{,}380 \text{ ft.-lb.} = 40{,}560 \text{ in.-lb.}$$

$$S_{avg} = 40{,}560/.49769/(2.78/2) = 58{,}630 \text{ psi}$$

The safety factor for the rivets is SF = 95,000/58,630 = 1.620.

6.0 MANUFACTURING TRADE STUDY

This section discusses the manufacturing processes available to produce the lowest cost part. The manufacturing study was divided into three parts: tube fabrication, machining, and assembly. The tube fabrication involved the method of applying the resin to the roving, the filament winding, the curing process, and the quality of parts to be processed. These steps are described in Section 6.2 and Section 6.3. The combination of a wet wound part on a numerically controlled (NC) winder using an internal stream cure was selected for fabrication. For the purposes of this analysis, the production volume has been fixed at 2,000 shafts. The quality requirements emphasize the strength related factors and minimize appearance and surface condition factors. The production plan would employ automation where an immediate return can be realized within the 2,000 shaft production run.

6.1 Selecting an Optimum Process of Fabrication

The first basic decision in the manufacturing of a composite part is to identify the processes to be used. The quantity of parts is the first consideration since capitalization costs for automation and permanent tooling will greatly effect the unit cost. The complexity of the part is the second consideration. The strength requirements of the part require uniform orientation of fiber in a specified direction for each layer. There are four production processes that could satisfy this: a hand lay-up of unidirectional mat, filament winding, a continuous lamination process, and automatic tape lay-up. Based on previous experience, Brunswick has selected filament winding which provides the lowest cost alternative for low volume production.

6.2 Optimization of Filament Winding

There are three basic trade-off decisions to be made in optimizing the filament winding of this part. These are the methods of applying the resin to the roving, the type of winder to be used, and the number of parts to be wound at the same time. The first decision is to select the method of applying the resin to the roving. Two methods exist for applying resin to the roving. The roving could be preimpregnated prior to winding or the resin could be applied directly to the roving using a wet bath as the part is being wound. The latter method is called wet winding.

The prepreg could either be purchased or produced internally by Brunswick. Preimpregnated roving provides for a uniform distribution of the resin on the roving independent of wind angle and winder speed due to the continuous smooth operation of the process. Preimpregnation increases the storage costs and adds an additional processing step, thus, the labor required is significantly increased. This additional cost must be offset by either an increase in the winding production rate over the wet winding method or justified by a critical resin content requirement.

Wet winding requires a resin bath where the low viscosity resin is applied directly to the roving. There are speed limitations with this process that vary with the wind angle, due to high accelerations that occur during turnaround when the winder carriage quickly reverses direction. Depending on the design of the roving delivery system and the wet bath, these limitations could be minimized.

The next trade-off is to determine the type of winder to be selected. It can be either a mechanically programmed type or a numerically controlled (NC) type. The shafts can be produced using either. Employing an NC winder would result in a reduction in set-up time and operating time; therefore, the NC winder was selected. Figure 27 shows the filament winding of a composite shaft for a helicopter at Brunswick in a NC winder.

The third trade-off decision was to determine the number of parts to be produced at a time. Due to the 160 inch length of the single shaft device, it is not feasible to produce multiple parts on a single mandrel. However, it is possible to wind multiple parts if the winder is modified to have a second winding spindle and additional winding eyes. This vertical stacking of wind mandrels is the most economical approach and is proposed.

6.3 Curing and Processing of Filament Wound Tube

There are two methods used for curing tubular products. In the first method, the part is rotated and heated while excess resin is removed. This continues until the resin is advanced to a gel condition such that it has cross-linked (B-staged) sufficiently so it will not flow when heated to higher temperatures during cure. The part is then oven cured without rotation. The second curing method is to overwrap the part with a bleeder fabric that will absorb the excess resin that migrates outwa. during cure. Pressurized steam is injected into the steel mandrel to provide a quick cure. After the steam cure, the bleeder material is removed. Since the part was cured initially from the inside, thermal expansion will aid in easy mandrel removal.

The selected cure for this application is a steam cure that requires only 40 minutes, versus an oven cure that requires 6 to 8 hours. The shorter cycle time with the steam cure provides a decided cost advantage in that less tooling is required. Only 12 mandrels would be required for the 2,000 unit process; however, 24 mandrels would be required for the oven cure.

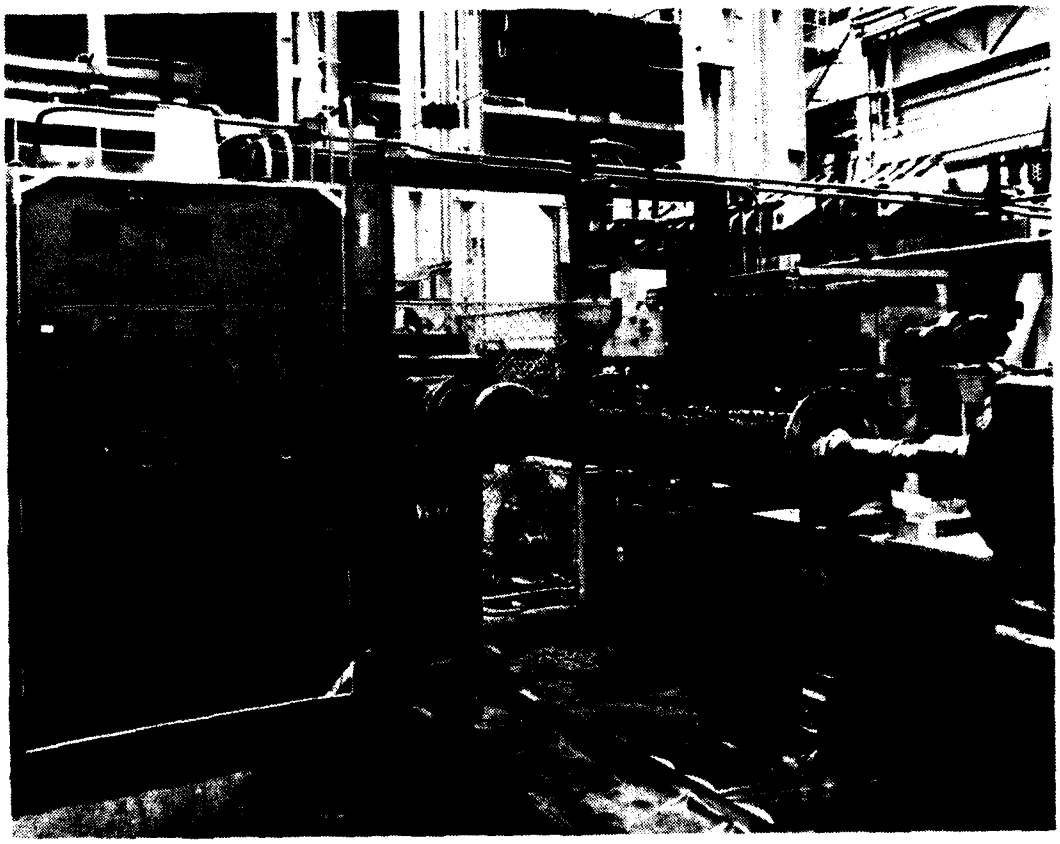


Figure 27. Winding of Helicopter Drive Shaft

6.4 Machining Methods

The manufacture of this part would require two machining processes. First, the part would be trimmed to the required length and then ground to constant diameter at both ends. These lathe operations could be performed using either manual, tracer, or numerical controls. The initial cost and the recurring cost of each method is significantly different. The key factors in selecting the control method were: part complexity, number of units, and required repeatability. This part would be relatively simple to machine and the volume of production parts relatively low. Therefore, the lowest cost option would be the use of a manual lathe with indicating stops for trimming to length.

The second machining process must provide a series of very accurate holes for attaching end fittings. Methods considered were: drilling, water jet cutting, or laser cutting, all of which can provide the required accuracy.

The key factors in drilling are: minimize heat-up of the composite and prevention of delaminations. This could be accomplished using a special spade drill, slow feed speed, and light drill pressure. When the drilling set-up takes these key factors into account, good repeatable results can be obtained.

An alternate method of providing holes in a composite structure is using water jet cutting. Since water jet cutting can cause delaminations in thick graphite laminates, this process was not considered.

A third method involves laser machining, see Figure 28. This would be an attractive method of machining the rivet holes in high volume programs except for the possibility of charring the laminate. Also, the equipment required is very expensive and not practical for small volume production.

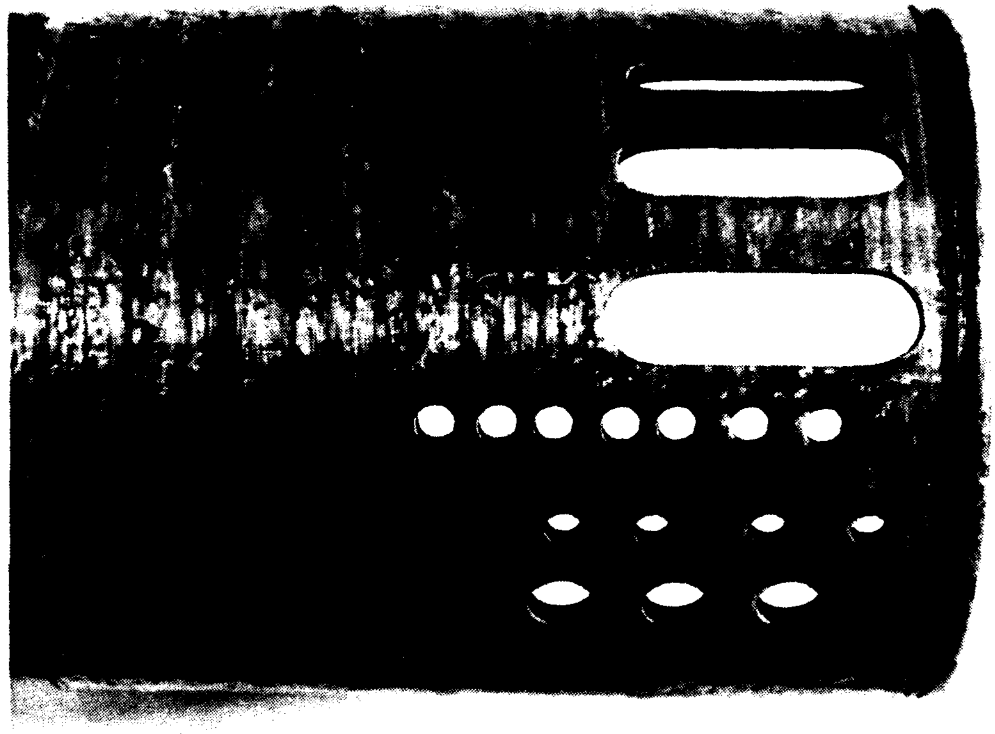


Figure 28. Laser Trimmed Holes

6.5 Assembly of Joints

The assembly of the steel end fitting to the composite tube has been basically fixed by the design selection of the blind rivet as a fastener. These blind rivets are set in place by a pulling force on the inner stem, which engages the tapered stem fully in the outer housing such that the outer housing is expanded radially. These rivets and their associated fastening tools are routinely used in the aerospace industry. The only production improvement for consideration would be to employ multiple riveting heads. However, the volume would not justify more than two rivets being installed simultaneously. Both rivets and the part would be indexed to all ten positions by the fixture. Reversing the shaft to assemble the opposite end would be manually accomplished with a hoist assist.

7.0 PROTOTYPE AND QUALIFICATION UNITS

As part of Phase I study, Brunswick quoted and made recommendations for a tentative demonstration program. However, this demonstration has not been funded to date and the actual work on the LVTP(7) vehicle shaft program ended with completion of the feasibility study. The prototype program proposed includes the production of three test shafts and two prototype shafts for actual installation in a vehicle. A tentative program schedule and recommended testing program are shown in Figure 29 and Table X. The prototype shafts would be produced using processes similar to those proposed for the production shafts (see Section 8.0).

The prototype tooling would consist of one steam-cured mandrel and some minor modifications to an existing stripping fixture. An existing winder would be used in its present configuration, and parts would be wound one at a time. The part would be steam cured, trimmed, and stripped.

All the turning and grinding would be performed on a standard lathe. The holes would be manually drilled. The machined parts would have the inside foamed filled. The foam is used to prevent water entry into the vehicle should the laminate wall be penetrated. Also, water in the tube can drastically affect critical speed. Then the joints would be fastened with air-powered rivet guns. The shafts would be final balanced by grinding off excess surface material.

The testing of the shafts would consist of six individual tests that would be used to verify the design. These are listed separately in Table X. The listed tests have been chosen to verify that the shaft would perform as designed and fail in a safe manner.

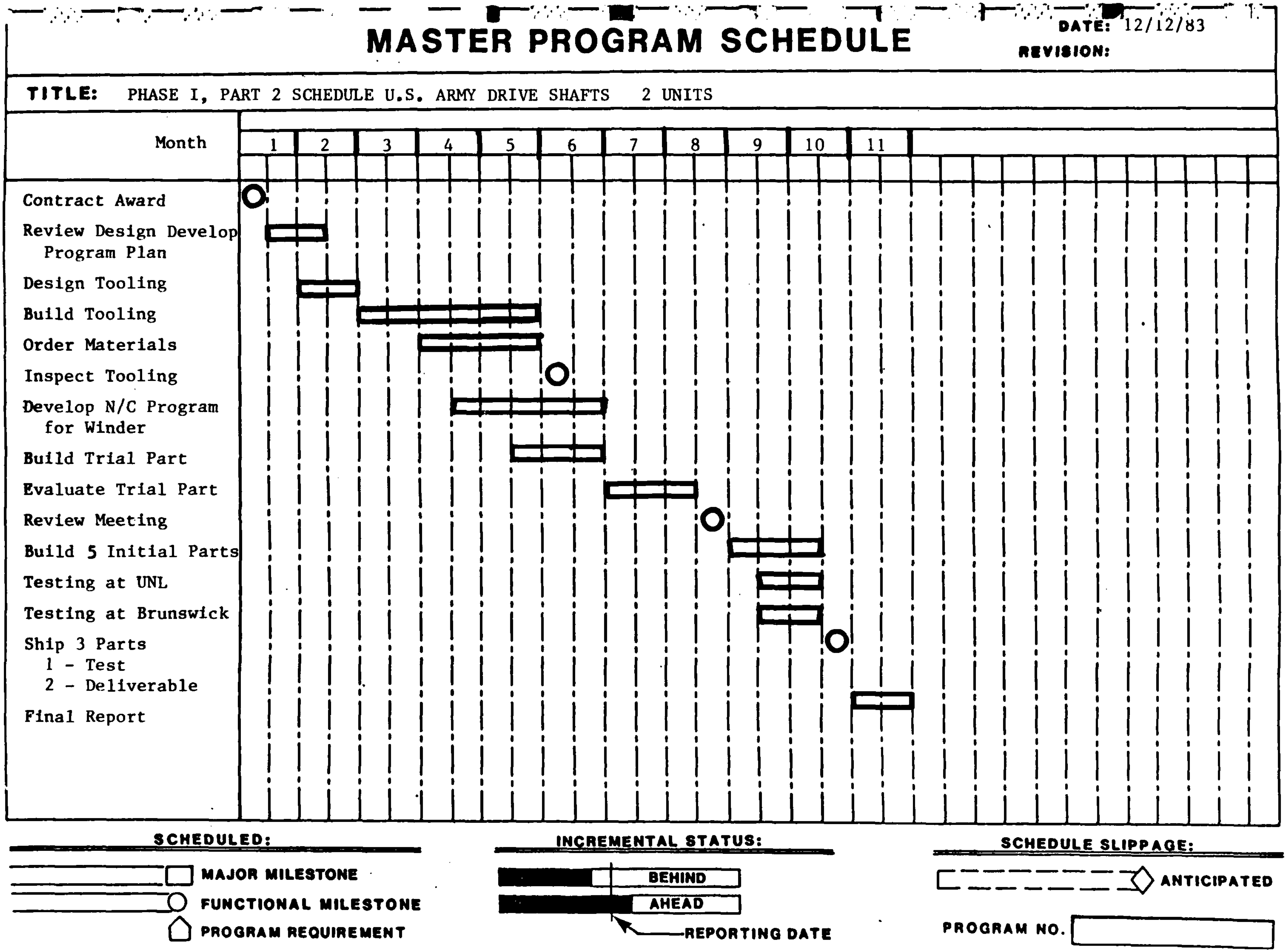


Figure 29. Schedule Phase I

Table X

PROTOTYPE/QUALIFICATION PROPOSED TESTING PROGRAM

UNIT	TEST	LOCATION	REASON
1	Critical Frequency**	University of Nebraska-Lincoln	Confirm critical frequency of part outside of operating envelope
2	Bending/Torsional Similar to ASTM A618-74	University of Nebraska-Lincoln	Establish if shaft installed incorrectly will not fail catastrophically
2	Torsional*	Brunswick	Verify design calculations
2	Joint Destruction**	Brunswick	Determine mode of failure
2	Impact Test*	Brunswick	Determine effect of large impact
3	Verification of* Design Life	AMMRC	Cycles to failure

*Similar testing to ASTM Spec. D2310-80 which covers machine made reinforced thermosetting resin pipe but does not define the specific tests specified.

**Design verification tests to be defined for this special application by Brunswick.

8.0 PRODUCTION PLAN FOR 2,000 SHAFTS

The winding, machining, and assembly of the quantity of 2,000 composite drive shafts (1,000 vehicle shaft sets) would be in accordance with the basic processes selected in Section 6.0. A flow diagram is shown in Figure 30. The production plan is illustrated in Figures 31 through 33 and the proposed production schedule in Figure 34.

The proposed winder would be similar to that shown in Figure 35 except that it would be modified to wind two parts simultaneously. The machine would be numerically controlled with at least two axes of movement. The winding program would be stored on a standard floppy disk and read only memory. The program would provide steps through the winding operation for program verification after each layer is completed. The actual production steps and time estimation are covered in the following sections.

8.1 Winding and Curing of the Composite Tube

Figure 31 shows a production plan for winding and curing of the composite tube and the approximate time required to complete these steps on a production lot of six parts. The production times are estimated for the 200th unit and projected to the entire production run of 2,000 units using a progress trend curve. The composite tube would be fabricated using the raw materials listed in Table XI.

The mandrel would be a steel cylinder 164 inches long and 3 inches in diameter that has hemispherical domes with a winding axis attached. The mandrel would be cleaned and then coated with a mold release. The first composite layer would be wet wound at ±45° using 34 msi graphite roving and a resin bath. The winding pattern would be established by numerical controls which set the mandrel rotation and the carriage positions. At the completion

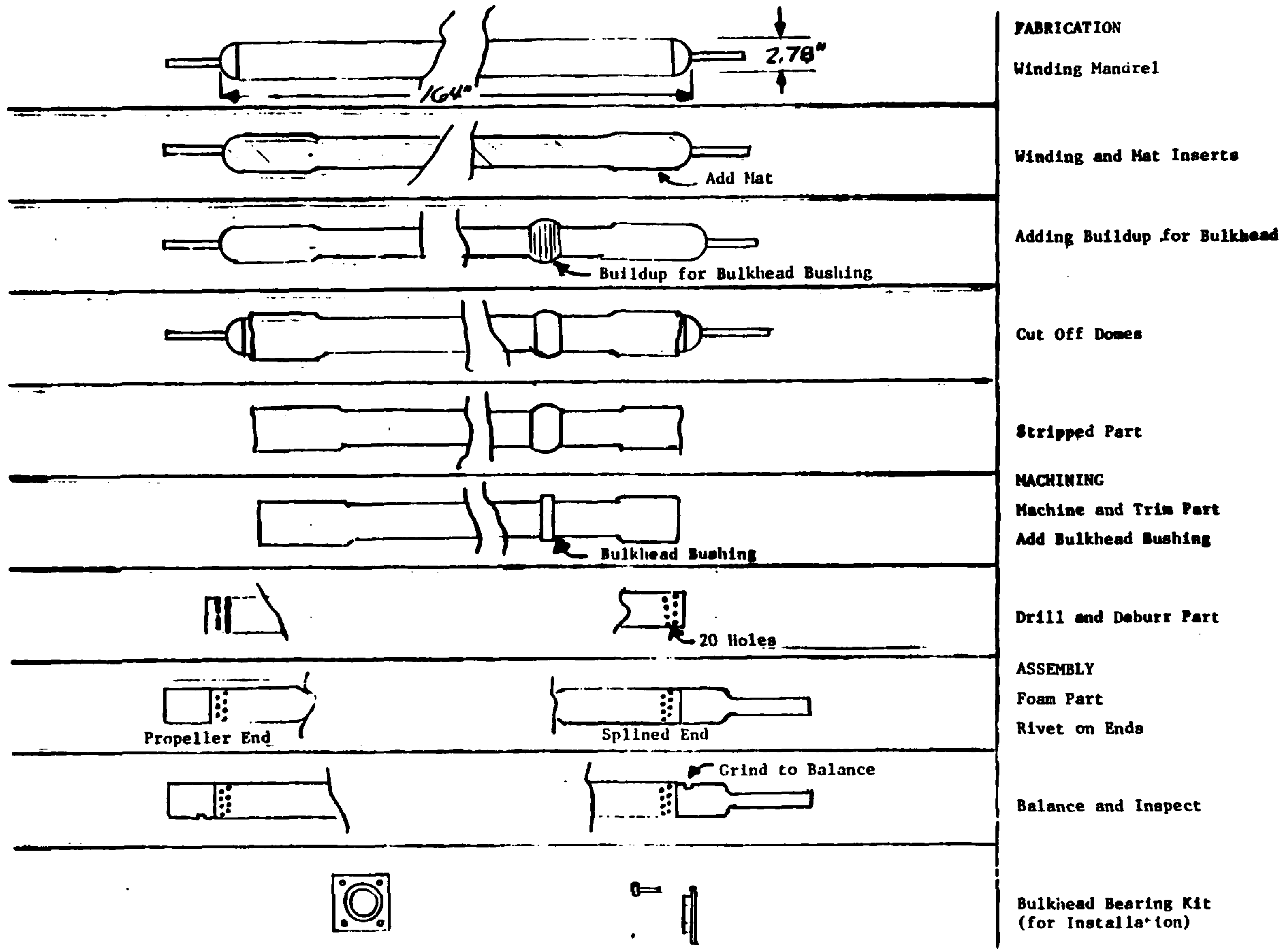
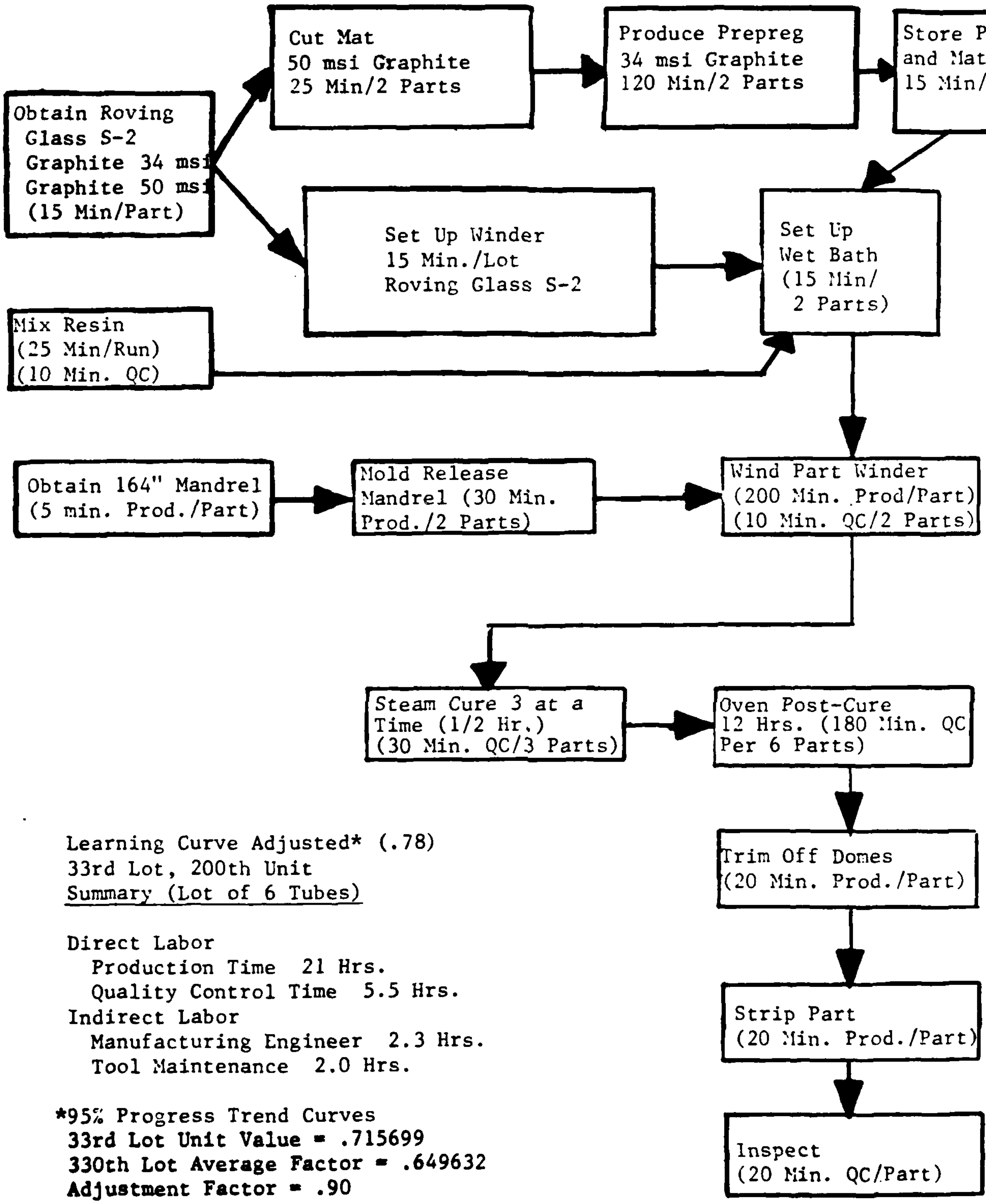


Figure 30. Flow Diagram



Learning Curve Adjusted* (.78)
33rd Lot, 200th Unit
<u>Summary (Lot of 6 Tubes)</u>

Direct Labor
  Production Time 21 Hrs.
  Quality Control Time 5.5 Hrs.
Indirect Labor
  Manufacturing Engineer 2.3 Hrs.
  Tool Maintenance 2.0 Hrs.

*95% Progress Trend Curves
33rd Lot Unit Value = .715699
330th Lot Average Factor = .649632
Adjustment Factor = .90

Figure 31. Winding and Curing of Composite Tube

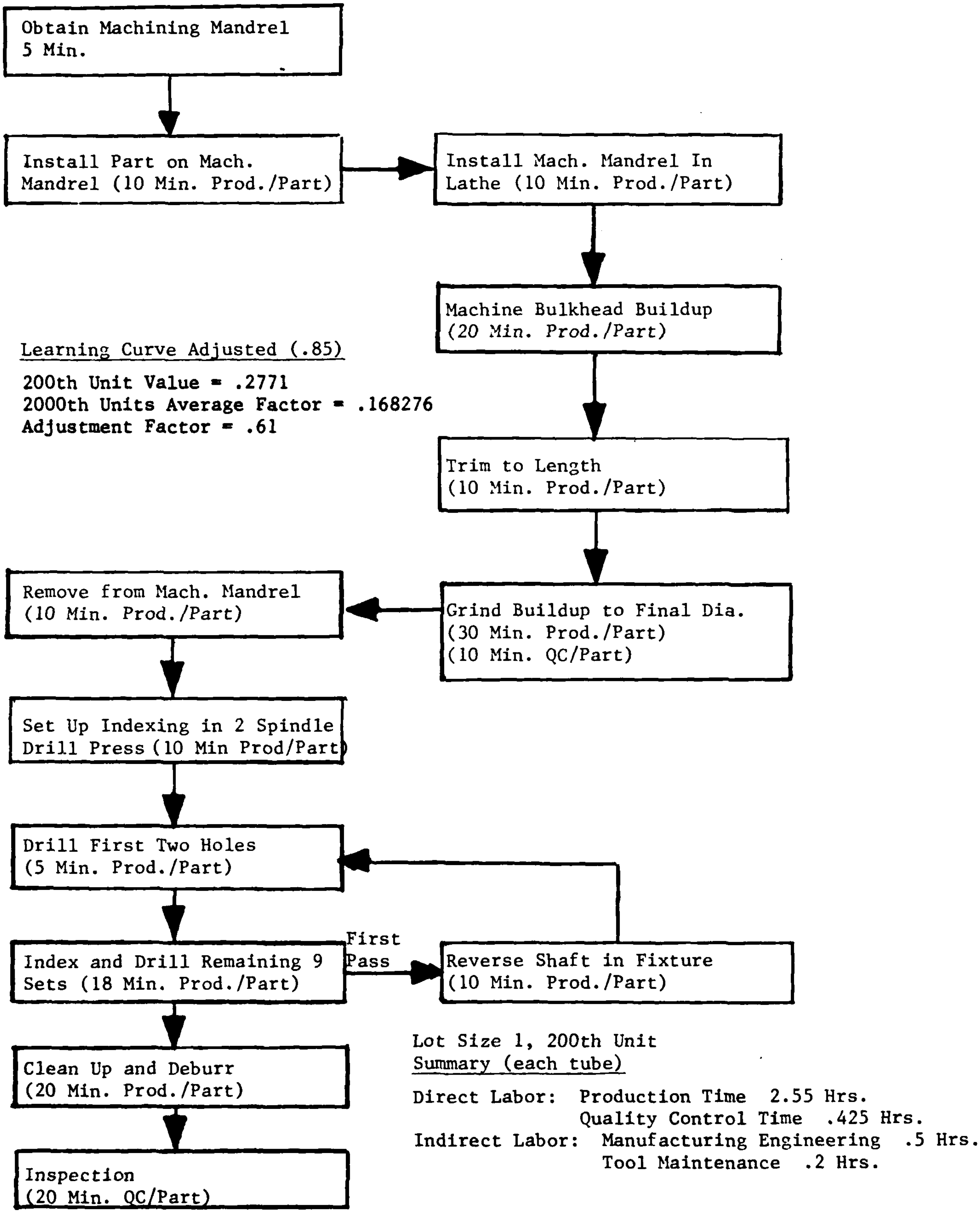
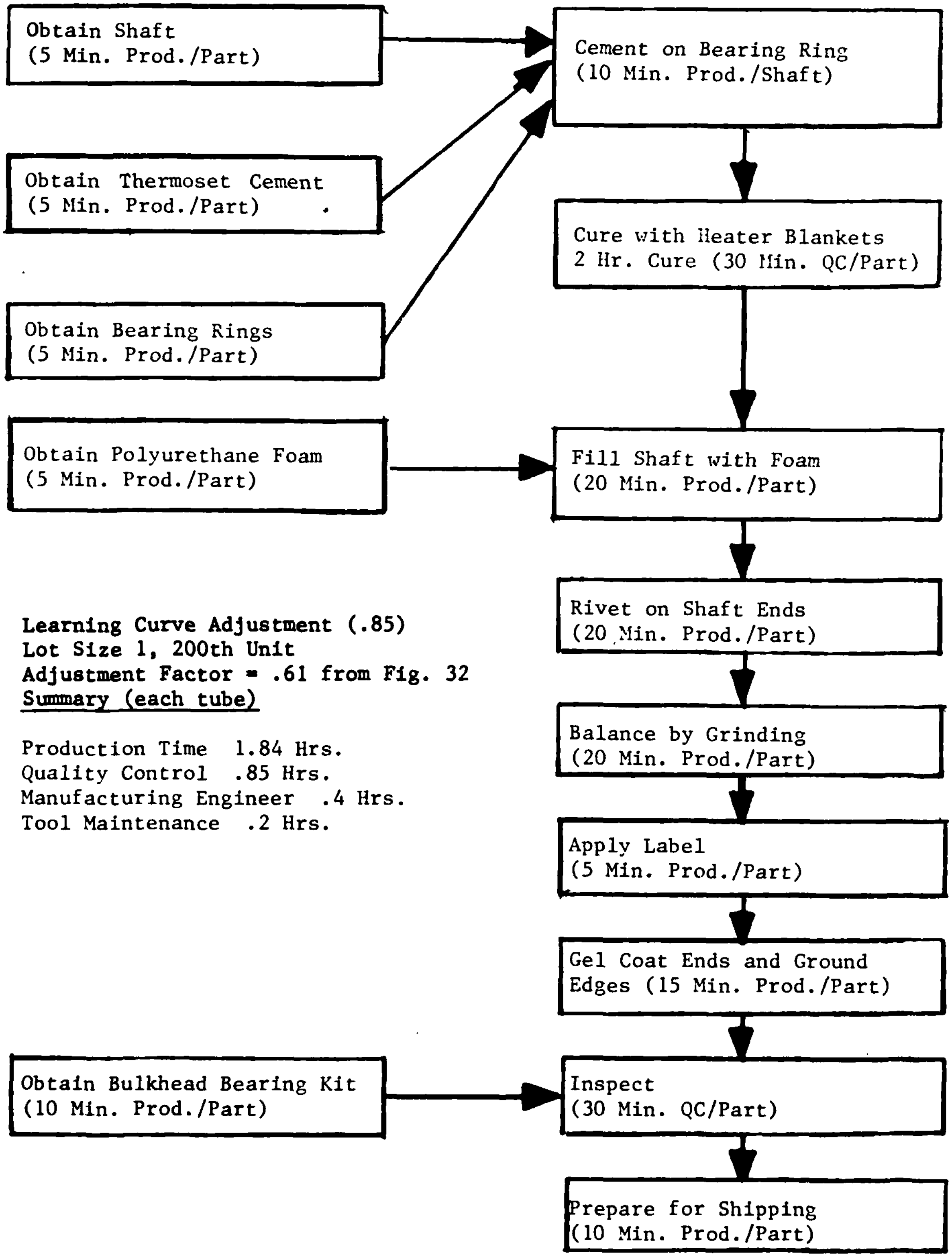


Figure 32. Machining of Composite Tube



Learning Curve Adjustment (.85)
Lot Size 1, 200th Unit
Adjustment Factor = .61 from Fig. 32
Summary (each tube)

Production Time 1.84 Hrs.
Quality Control .85 Hrs.
Manufacturing Engineer .4 Hrs.
Tool Maintenance .2 Hrs.

Figure 33. Assembly of Shaft

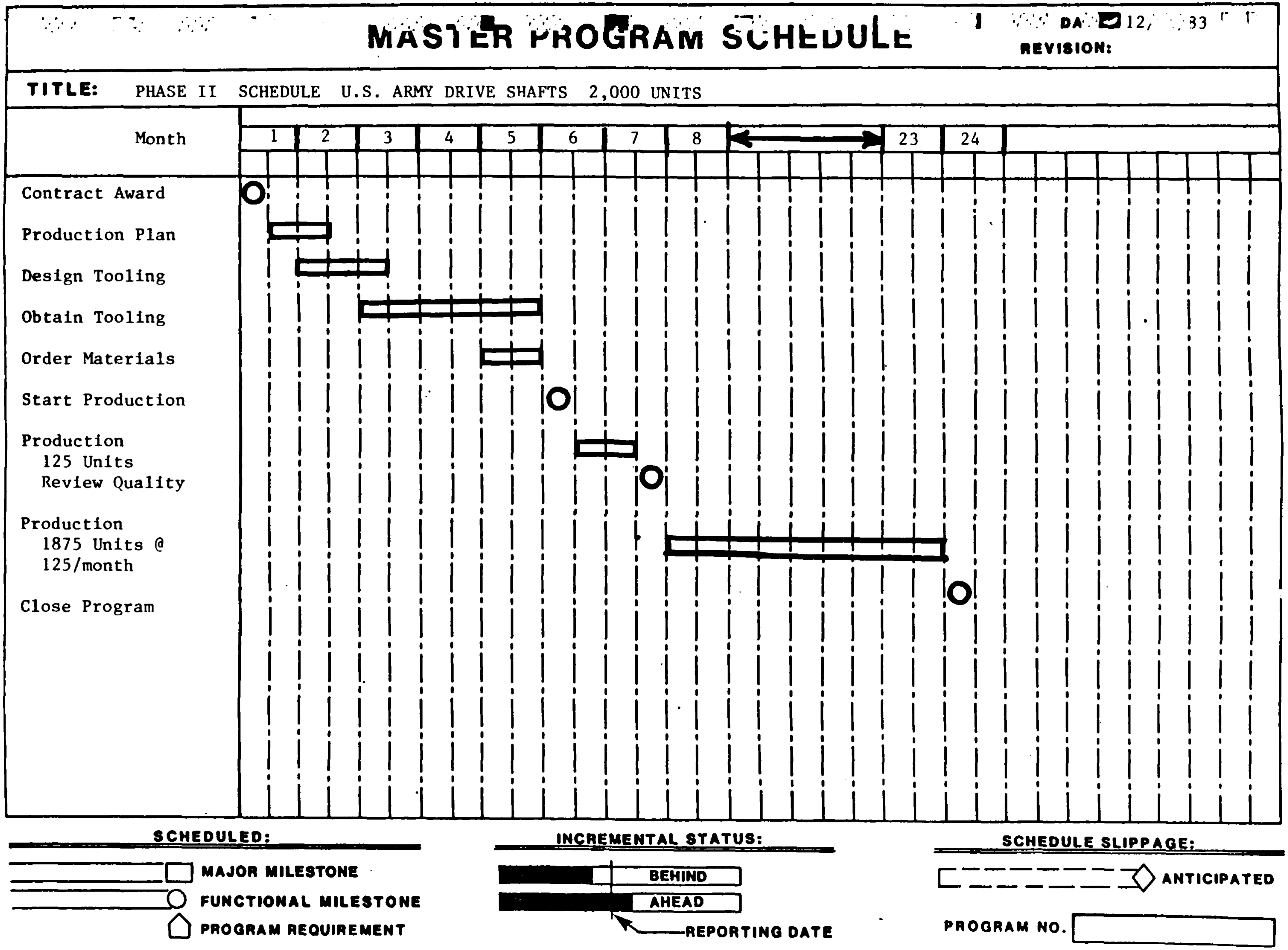
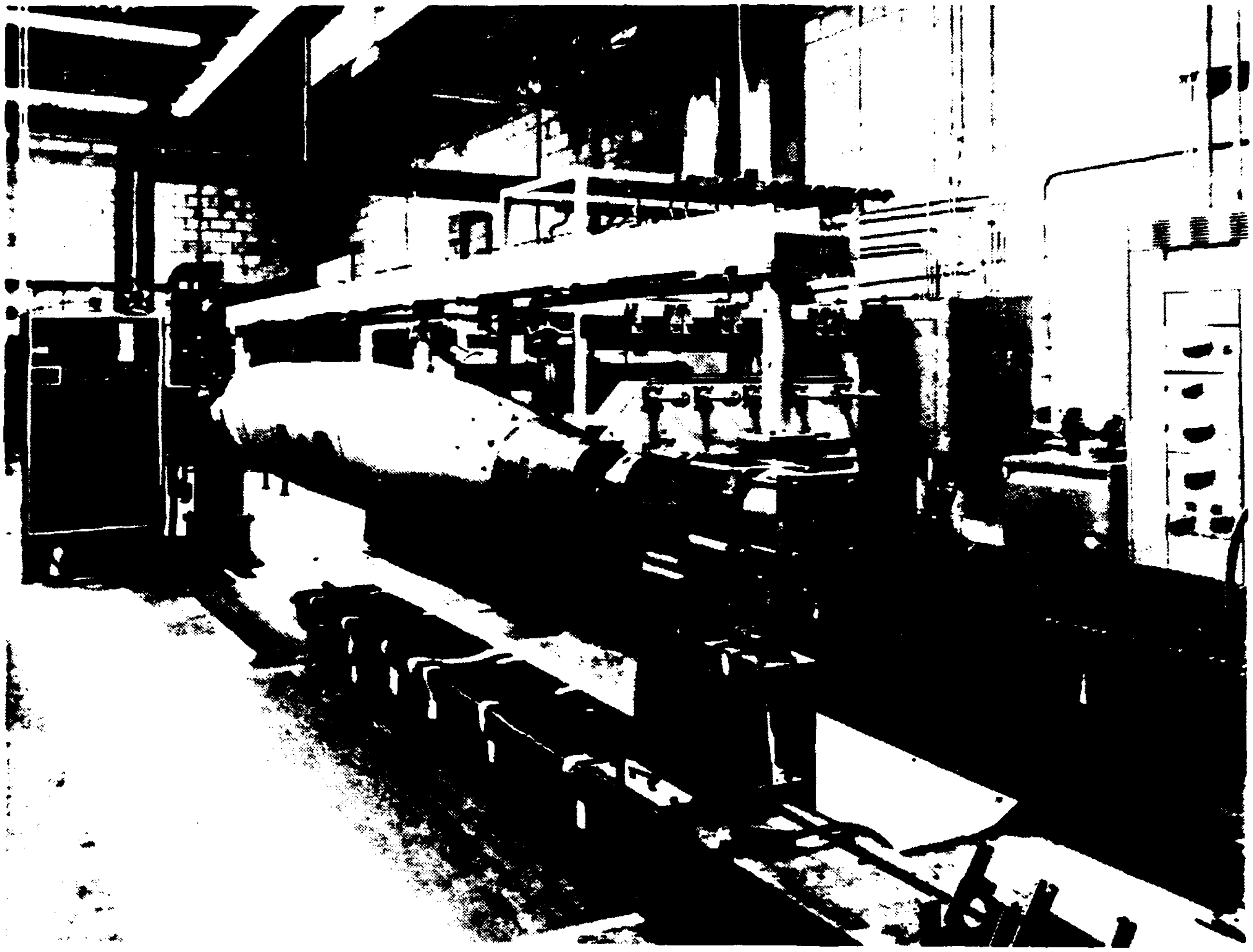


Figure 34. Schedule Phase II



SPECIFICATIONS

A. Control System: Computer Numerical Control
   1. Four Each Independent Motion Axis

      I. Axis #1 - Mandrel (Radial)
      II. Axis #2 - Carriage (Linear)
      III. Axis #3 - Crossfeed (Linear)
      IV. Axis #4 - Eye (Radial)

   2. Capabilities

      Maximum Carriage Travel Exceeds 20 ft.

Figure 35. MAW I - NC Winder

Table XI

ESTIMATE COST OF RAW MATERIAL (2,000 UNITS) SINGLE COMPOSITE SHAFT DESIGN
Composite Weight 15 Pounds (10% Loss in Process/10% Trim)

Material Item	% by Weight of Composite	Cost per Unit	Quantity	Cost per Item
S-Glass Roving	12	$ 5.12	1.8 lbs.	$ 9.22
Graphite Roving 34 msi (Raw Material)	21	21.00	3.15 lbs.	66.15
Graphite Roving 50 msi (Raw Material)	27	55.00	4.05 lbs.	222.75
Epoxy Resin	40	1.60	6.0 lbs.	9.60
End Fittings		42.00	2.00 each	84.00
Rivets		2.50	40.00 each	100.00
Flexible Seal		30.00	1.00 each	30.00
Polyurethane Foam		10.00	1.00 each	10.00
Label		.10	1.00 each	.10
				$ 531.82

*Estimated cost based on lot requirements and 1985 delivery.

of the first layer, the operator would stop the winder and apply precut 45° mat of 50 msi graphite. Next, a layer of circs (90°) would be wound the length of the mandrel using wet-wound glass roving. The process would be continued as shown in Figure 33. Finally, during the winding, the bulkhead bearing surface buildups would be wound. Upon completion, excessive resin would be wiped off and an overwrap of release fabric applied.

The laminate would be cured by passing live steam through the I.D. of the mandrel. The part would be removed from the mandrel by cutting off the domes and using a hydraulic stripper to separate mandrel and part. The release fabric would be removed and the part inspected for defects.

8.2 Machining of Composite Shafts

Figure 32 shows a production plan for machining and the approximate time required to complete these steps again on a production lot of six shafts. After the cured shaft was inspected, it would be placed on a machining mandrel which would allow the joint end diameters and bulkhead bearing surface to be rough machined using a single point carbide tipped tool. Then the shaft would be cut to length using two mechanical stops to position the cutting tool. The last lathe operation would be a grinding operation that would grind the final outside diameter of the part at the bulkhead bearing surfaces and the joint ends to a 125 finish and within ±.003 of the specified diameter.

The composite tube would be removed from the machining mandrel and placed in an indexing multiple spindle drill press. The multiple spindle drill press would be set up to drill two holes in line with the center line of the shaft. An indexing fixture would rotate the shaft 36° per index position so as to provide a pattern of ten sets of two holes each. The shaft would be removed from the drilling fixture and the other end machined in an identical manner.

The final operation would be a deburring operation using a flapper wheel to remove any loose fiber ends or rough edges.

8.3 Assembly of Composite Shaft

Figure 33 shows a production plan for assembling the composite shaft and the approximate time required to complete these steps on a production lot of six tubes. The metal shaft coupling would be purchased with interfaces compatible with existing shaft components in the vehicle. A bearing ring and seal would be purchased and supplied to provide for the bulkhead interface.

A brass ring with a protective coating would be cemented on the shaft at the bulkhead bearing surface areas to provide a support for the seal. An epoxy adhesive would be used, and it would be cured using two small strip heater blankets that will provide localized heating to cure the cement.

The shaft would then be filled with a room temperature cure polyurethane foam to provide a watertight seal. Next, the metal coupling would be blind riveted onto the shaft using automatic tools. The shafts would then be balanced by grinding away excess material on the metal couplings. A label would be applied and clear seal coat applied to all ground surfaces. The completed shaft would be inspected and packaged with a bulkhead bearing kit. It would be ready for installation when received.

8.4 Schedule, Tooling, and Labor Estimate

The schedule in Figure 34 details the required steps to develop a production capability of 125 shafts per month and to produce 2,000 shafts over an 18-month production run.

The first step in planning the production run would be to order any of the following tooling not currently available at Brunswick's facilities.

New Tooling

Quantity	Description
12	Mandrels 164 Inches Long (Steam Cure)
2	Mandrel Carts
1	Balancing Machine or Subcontract
1	Set Lathe Stops
1	Adapter to Wind Two Parts at Once on Winder
1	Air-Powered Indexing Fixture
1	Two Spindle Drill Press with Exhaust Fan
2	Cleaning Swabs 14 ft. long
2	Testing Fixtures for QC
1	Air-Powered Rivet Gun

The following equipment is currently available at Brunswick.

Quantity	Description
1	NC Winder
1	Lathe 180 Inch Bed
1	Production Oven
1	Steam-Curing Station
1	50 ft. x 50 ft. Work Area
1	Foam Machine

The second step would be to order the raw material shown in Table XI. The third step would be to implement the production plan for fabrication, machining, and assembly. The projected recurring labor hours are obtained by adding the subtotals from Figures 31, 32, and 33.

The estimated cost of producing five prototype units (two for delivery) and 2,000 production units is included in Appendix B (which has been distributed on a need to know basis).

APPENDIX A

DESIGN AND DEVELOPMENT OF JOINT CONCEPTS

Table of Contents

List of Figures

1.0 INTRODUCTION

The need for large thickwall torque shafts has been evidenced by both recent contracts and request for quotes. Brunswick, in the past year has built and delivered various configurations of helicopter rotor shafts which varied in wall thickness from 1.0 to 2.0 inches and lengths from 27 inches to 96 inches. Brunswick has also built various configurations of propeller test shafts that have been tested by the Naval Ships Research and Development Center (NSRDC). The major concern in the design of shafts has been the method of attachment of the shaft to the adjacent members of the drive train. The helicopter rotor shafts utilized a pressed pin connection concept designed by the helicopter manufacturer. The shafts built for test by NSRDC utilized various Brunswick designed connection concepts. The various concepts utilized to date have had various levels of success. This report presents the following design concepts with corresponding test results:

1. Pinned flange torque shaft built under this task.
2. Wound around pin concept tested under this task.
3. Propeller shafts built under other cost centers* and tested by NSRDC.

2.0 PINNED FLANGE TEST SPECIMEN

2.1 Specimen Description

One test shaft was fabricated and tested under this task. The shaft consisted of approximately 50 percent S-2 glass and 50 percent T-300 graphite by volume in an LRF 092 matrix. The winding sequence is shown in Figure 1 along with the shaft physical dimensions. This shaft had tapered metal

*Cost Centers 12406 and 12439.

Material	Wind Angle, degrees		Thickness, inches*
Graphite	±45		.375
S-2 Glass	±45		.195
S-2 Glass	90		.180
		Total	.75

The specimen was 48 inches long and had an inside diameter of 1.0 inch and an outside diameter of 2.5 inches.

*Three equal thickness layers of each material and wind angle were interspersed through the cross section.

Figure 1. Winding Sequence of Pinned Flange Shaft

flanges pinned in place on each end using eight .375 inch diameter dowel pins. The flange configuration is shown in Figure 2. The flange taper was calculated so each row of pins would carry an equal load under simple torque.

2.2 Test Method

The testing was conducted by NSRDC, Annapolis, Maryland. The test consisted of a cyclic torque test only. The specimen was vertically mounted with a hydraulic actuator used to apply the torque. The test setup is shown in Figure 3.

2.3 Strength Prediction

The estimated failure stress for this specimen was 71,900 in.-lbs. torque for 1×10^6 cycles of full reversal load. The method of prediction is shown in Figure 4. The stress concentration factor used is an estimate.

2.4 Test Results

The test results are presented in Figure 5. The failure was evidenced by the specimen catastrophically failing around the inboard pin holes. No evidence of hole elongation was noted during the test, but the pin movement in the composite does indicate a less than optimum condition which certainly did cause some fretting wear, and decreased the strength below that which was estimated. This accounted for the discrepancy between theoretical and actual failure levels. This test has indicated the importance of using body-bound fasteners (fasteners that are not allowed to move in relation to the adjacent material).

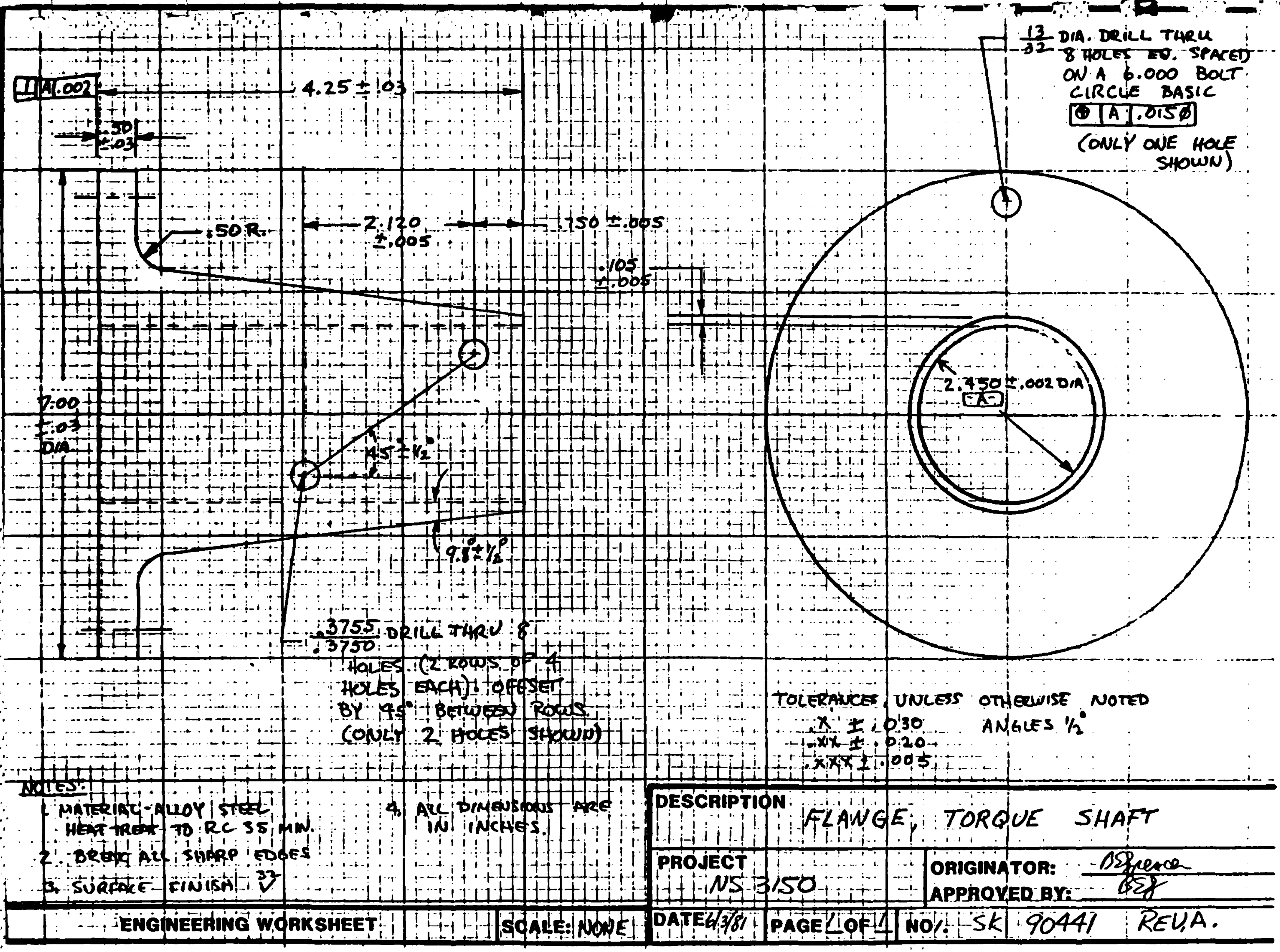


Figure 2. Flange, Torque Shaft

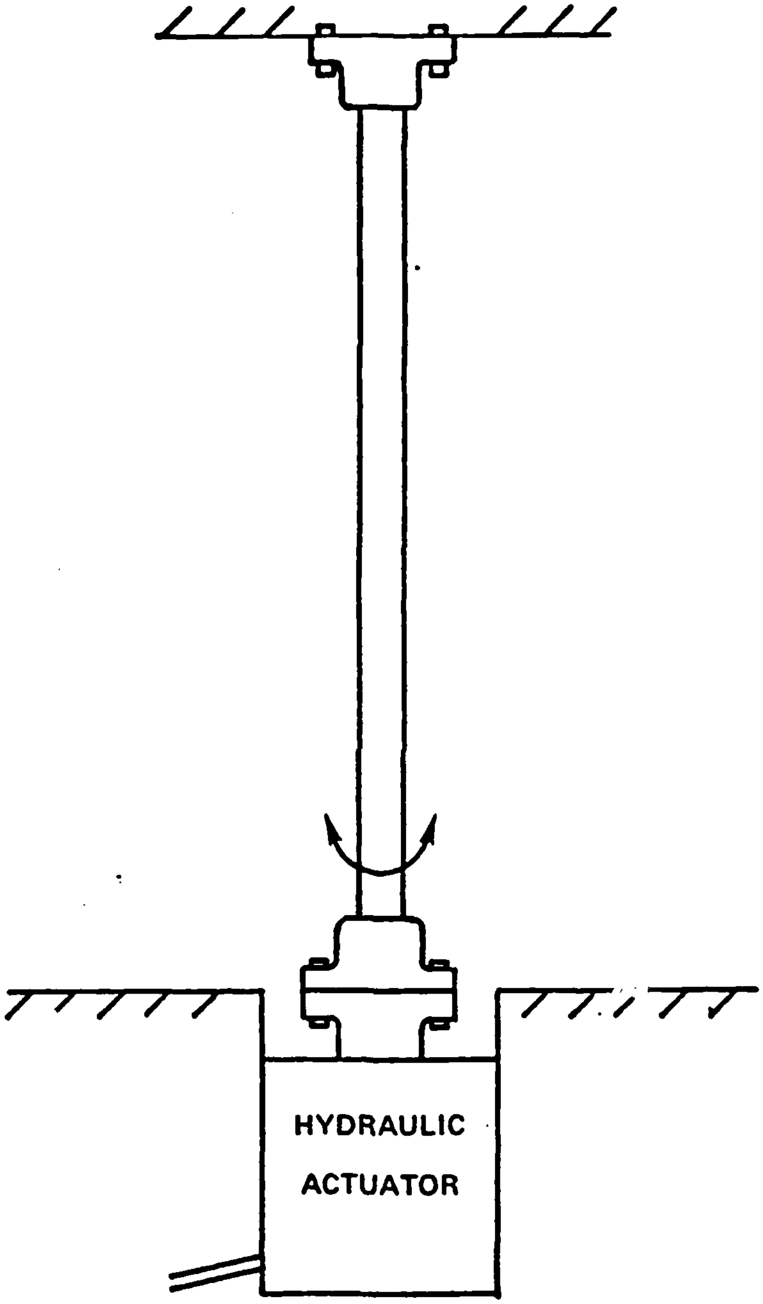


Figure 3. Torque Testing Apparatus

Item*	Design Allowable Ultimate	Fatigue	Layer Stress Under 47,829 in.-lb. Torque (16 ksi Torsional Stress)
45 GL_T	200 ksi	140 ksi	7.56 ksi
45 GL_C	150 ksi	105 ksi	7.56 ksi
45 GR_T	170 ksi	120 ksi	18.06 ksi
45 GR_C	80 ksi	56 ksi	18.06 ksi
90 GL_T	215 ksi	150 ksi	0
90 GL_C	160 ksi	112 ksi	0

Failure is predicted in compression in the 45° graphite fiber. The predicted fatigue failure torque level after 1×10^6 cycles is estimated as follows:

$$\text{Failure Torque} = (SL) \frac{(T)}{SLT} (AR) \frac{1}{SCF} = 56 \frac{(47.828)}{18.06} (.727) \frac{1}{1.5} = 71.9 \text{ ksi}$$

SL = stress at layer with minimum factor of safety
T = torque of 47,829 in.-lb.
SLT = layer stress at 47,829 in.-lb.
AR = area reduction for fastener holes
SCF = estimated stress concentration factor

Failure Torque = 71,900 in.-lbs. (24 ksi Torsional Stress)

NOTE: The current Navy design manual allows a maximum torsional stress of 16 ksi and maximum bending stress of 6 ksi in a propeller shaft.

*AA 11_2

AA = Wind angle
11 = Fiber, GL = Glass, GR = Graphite
2 = Load direction, T = Tension, C = Comparison

Figure 4. Pinned Flange Specimen Failure Prediction

Test Sequence	Comments
1. Torque Proof Test at 19.3 ksi torsional stress	57,600 in.-lb. full reversal, one cycle. No damage noted.
2. Fatigue Test at 13 ksi Torsional Stress	After 2,500 cycles of full reversal, the test was stopped and the specimen examined. Pins used to couple the flanges to the shaft had moved radially inward. The pins were backed out and welded in place.
3. Ultimate Torque Test	The specimen failed at 50,400 in.-lbs. The failure was at the pin holes.

NOTE: 13 ksi Torsional Stress = 38,862 in.-lbs.
19.3 ksi Torsional Stress = 57,600 in.-lbs.

Figure 5. Test Results Pinned Flange Test Specimen

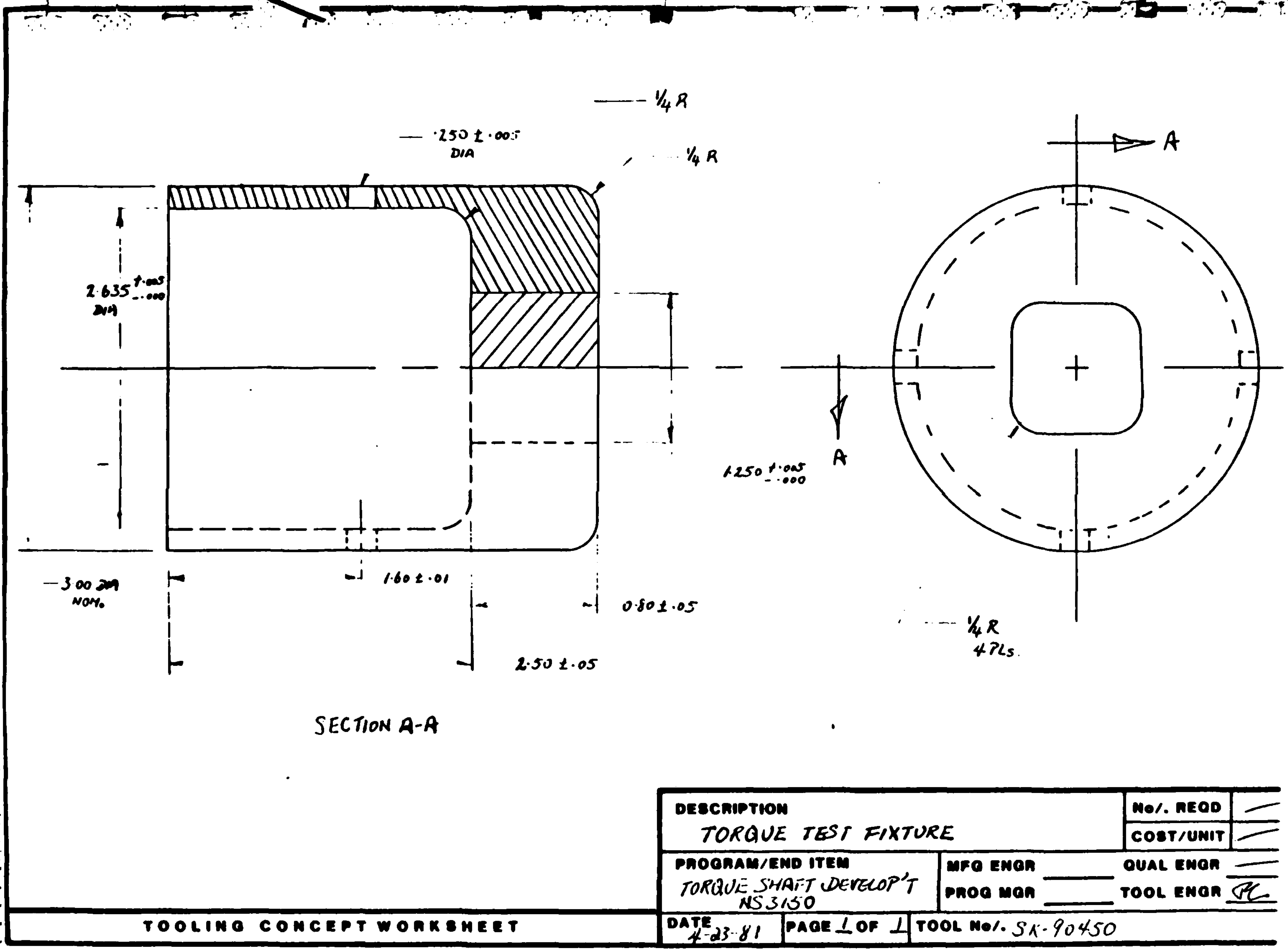


Figure 6. Torque Test Fixture

3.0 WOUND AROUND PIN TEST SPECIMENS

3.1 Specimen Description

To test the torque carrying ability of the wound around pin concept, four VIPER motor cases (S-2 fiberglass and LRF-268 epoxy) were modified for test on the University of Nebraska, Lincoln (UNL) torsion test machine. The modification included first cutting the nozzle off the motor case to form a cylindrical test specimen. A plug was manufactured which was then glued in place in the aft end. For the wound around pins, a fitting was made that fit inside the case and supported the pins. The test fittings are shown in Figures 6 and 7 and the test assembly is shown in Figure 8.

3.2 Test Method

The test was conducted on the UNL torsion test machine. Only an ultimate strength test under tension was conducted.

3.3 Strength Prediction

The ultimate failure prediction of 25,750 in.-lbs. torque is based upon the cross-sectional description shown in Figure 9 and an allowable shear stress of 6 ksi, which is based upon historical data.

3.4 Test Results

The test results are presented in Figure 10. The results are encouraging since two of the specimens failure torque approached the theoretical level of 25,750 in.-lbs. of torque. The failure mode was a shear failure as expected. The joint tested was designed for axial tension in the motor case. Modifying the joint for torsion should certainly increase its load carrying capacity.

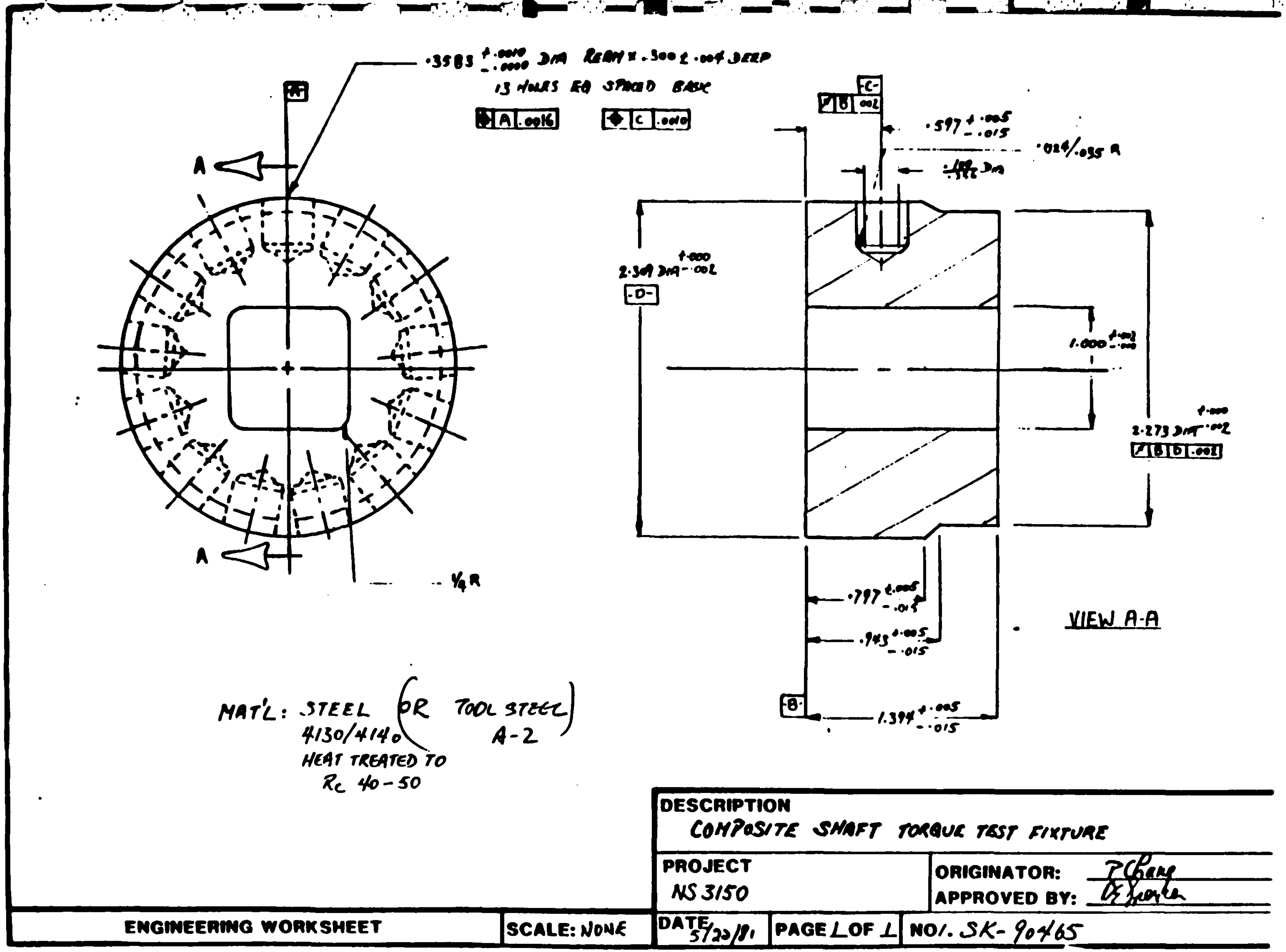


Figure 7. Composite Shaft Torque Test Fixture

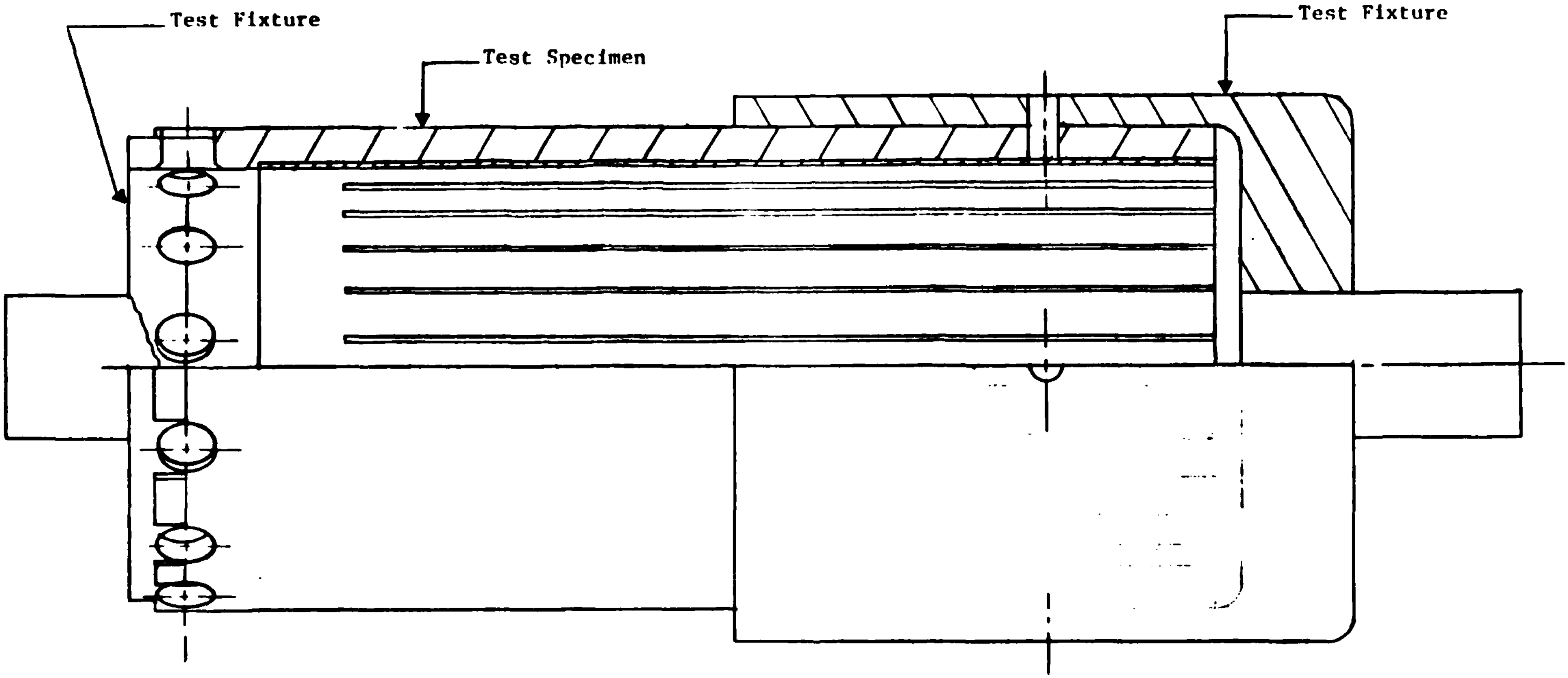


Figure 8. Wound Around Test Specimen

Item*	Ultimate Unidirectional Design Allowables	Layer Stress Under 1,000 in-lb Torque
27 GL_T	200 ksi	1.13 ksi
27 GL_C	150 ksi	1.13 ksi
27 GL_S	6 ksi	.233 ksi
33.7 GL_T	200 ksi	1.30 ksi
33.7 GL_C	150 ksi	1.30 ksi
33.7 GL_S	6 ksi	.152 ksi
40.3 GL_T	200 ksi	1.39 ksi
40.3 GL_C	150 ksi	1.39 ksi
40.3 GL_S	6 ksi	.06 ksi
47 GL_T	200 ksi	1.40 ksi
47 GL_C	150 ksi	1.40 ksi
47 GL_S	6 ksi	.028 ksi

The prediction indicates the specimen will fail in shear between the 27° glass layers. The ultimate load is predicted as follows:

$$\text{Ultimate Torque} = \frac{6 \text{ ksi}}{.233 \text{ ksi}} (1{,}000 \text{ in.-lbs.}) = \frac{25{,}750 \text{ in.-lbs.}}{(23.2 \text{ ksi Torsional Stress})}$$

*AA 11_2

AA = Wind Angle
11 = Fiber, GL = Glass, GR = Graphite
2 = Loading, T = Tension, C = Compression, S = Shear

Figure 9. Wound Around Pin Concept Failure Prediction

Specimen No.	Comments
001	Yielding started at 19,400 to 20,250 in.-lbs. with failure at 22,500 in.-lbs. Holes sheared out.
002	Yielding started at 24,000 in.-lbs. with failure at 22,840 in.-lbs. Holes sheared out.
003	Yielding started at 23,120 in.-lbs. with failure at 25,000 in.-lbs. Metal forward fitting failed, holes did not.
004	Yielding started at 16,800 in.-lbs. with no final failure. Adhesive joint between composite and test fixture failed at 20,100 in.-lbs.

NOTE: Average value for start of yielding is 20,800 in.-lb. which is 18.7 ksi torsional stress.

Figure 10. Test Results Wound Around Pin Concept

4.0 NAVY PROPELLER SHAFTS (DOUBLE FLANGE)

4.1 Specimen Description

As a comparison, the results from various propeller shafts built for Ingalls Shipbuilding by Brunswick and tested at NSRDC are presented. The specimens were in two configurations, torsion specimens, and bending specimens. The specimens were designed to test various attachment techniques. The specimen descriptions are presented in Figure 11 with an assembly sketch shown in Figure 12.

4.2 Test Methods

The testing was conducted by NSRDC, Annapolis, Maryland. The tests consisted of cyclic torsion and bending. The test setups are shown in Figures 3 (torsion) and 15 (bending).

4.3 Test Results

The test results are presented in Figure 16. Specimen A, which used shoulder bolts through an inner and outer metal flange, demonstrated the adequacy of the design for torque applications. Failure occurred at the inboard row of fastener holes. Specimen B, which had an inner and outer composite coupling, did not perform satisfactorily. The shoulder screws failed during proof load. It is felt that if body-bound pins are used the design should carry a torque equivalent to the inner and outer metal flanges used in Specimen A.

Although Specimen C passed the required cycle load bending test, improvements could be made to the flange design to improve bending

Specimen A, Torsion Specimen

Physical Dimensions: 48 inches long, 2.78 inch inside diameter, 4.0 inch outside diameter with 4.5 inch diameter buildups on ends for flange attachment. The laminate construction is shown in Figure 13.

Flange Attachment: Metal iiner and outer flange each end. Flange design shown in Figure 14. Steel bushings in composite wall (.938 inch OD). 5/8 inch shoulder bolts two rows of six bolts each on a flange.

Specimen B, Torsion Specimen

Physical Dimensions: Same as A except 4.5 inch diameter full length.

Flange Attachment: Same as A except teflon bushings (.938 OD) used in place of the steel busting.

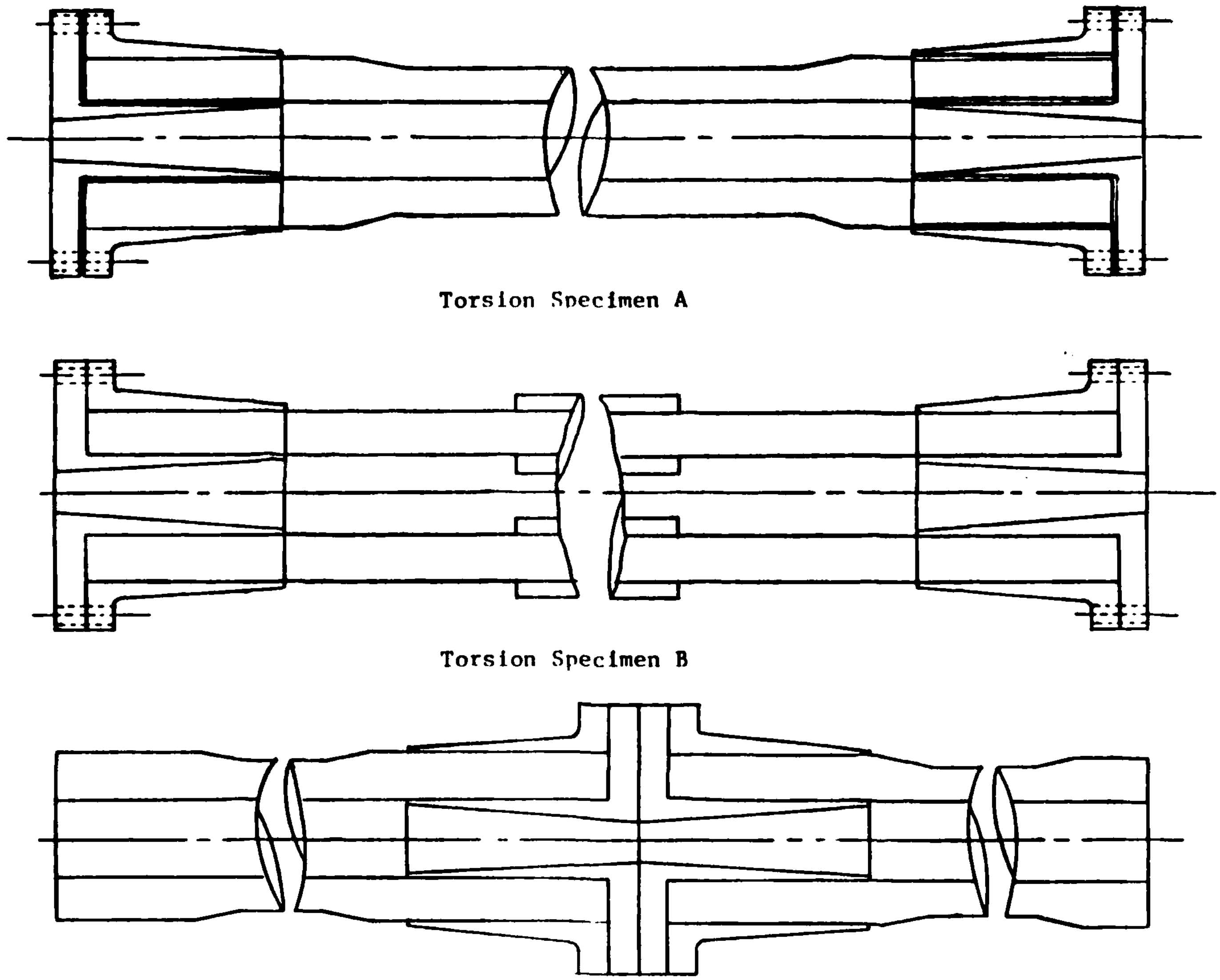
Center Coupling: Shaft cut in two in center and joined using inner and outer composite coupling with .44 inch wall thickness. Threaded insert in inner sleeve to accept shoulder screws.

Specimen C, Bending Specimen

Physical Dimensions: Identical to two specimen A's, back to back, 96 inches long.

Flange Attachment: Same as A with flanges in center of unit connecting 48 inch lengths together.

Figure 11. Navy Propeller Shaft Specimen Description


Torsion Specimen A

Torsion Specimen B

Bending Specimen C

Figure 12. Test Assemblies

Material	Wind Angle, Deg.	Center Section No. Layers Thickness, In.	Built-Up Ends No. Layers Thickness, In.
E-glass	90	1 - .022	1 - .022
Graphite*	15	2 - .044	2 - .044
E-glass	90	----	2 - .044
E-glass	45	6 - .132	6 - .132
E-glass	90	----	2 - .044
Graphite	15	2 - .044	2 - .044
E-glass	90	----	2 - .044
E-glass	45	6 - .132	6 - .132
E-glass	90	----	2 - .044
Graphite	15	2 - .044	2 - .044
E-glass	90	----	2 - .044
E-glass	45	6 - .132	6 - .132
E-glass	90	----	2 - .044
Graphite	15	2 - .044	2 - .044
E-glass	90	1 - .022	1 - .022
	TOTAL	28 - .616"	40 - .880"

The inside diameter of the shaft is 2.78 inches.

The graphite used has a modulus of 34×10^6 psi.

Figure 13. Laminate Construction for NSRDC Shafts

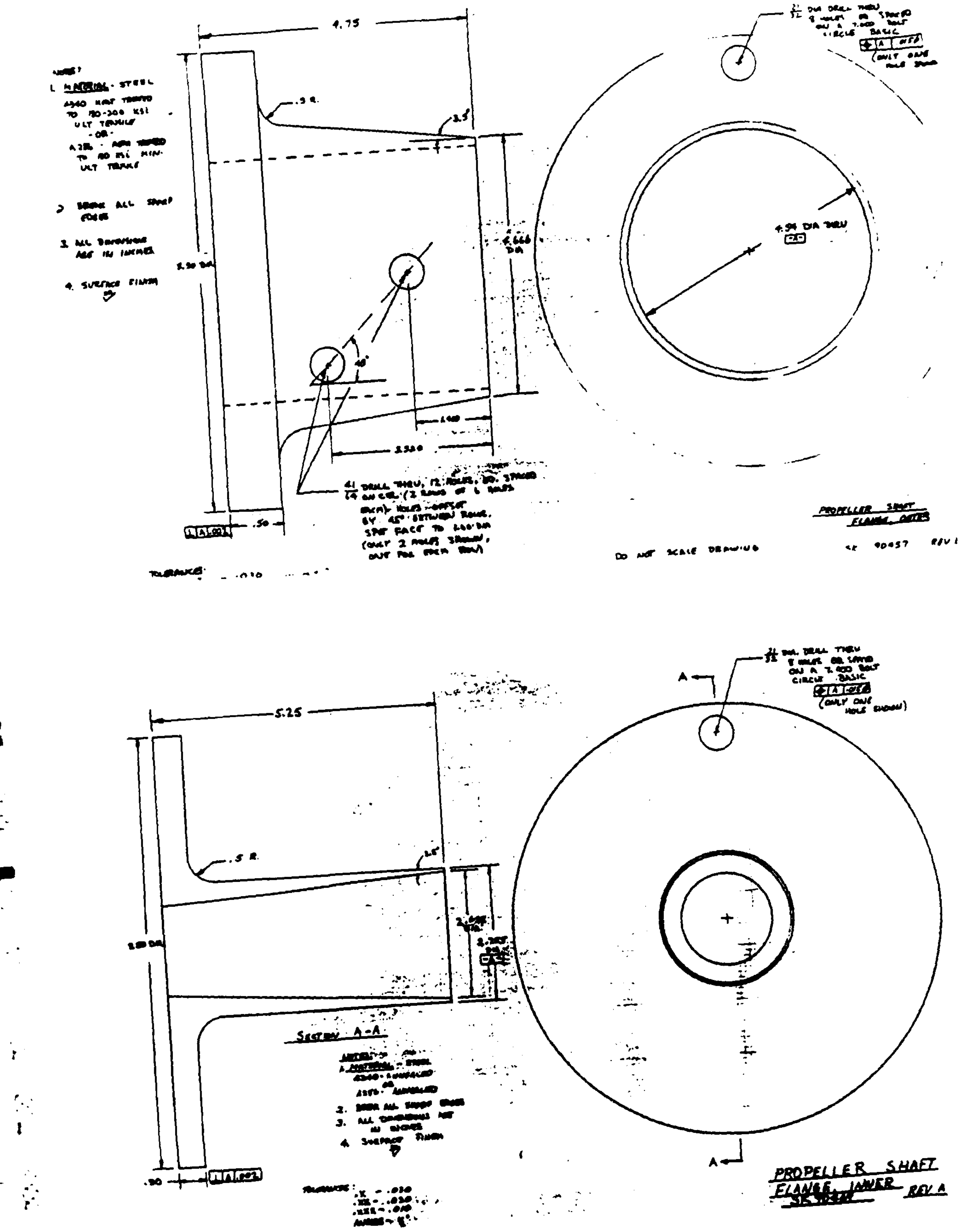


Figure 14. Flange Design

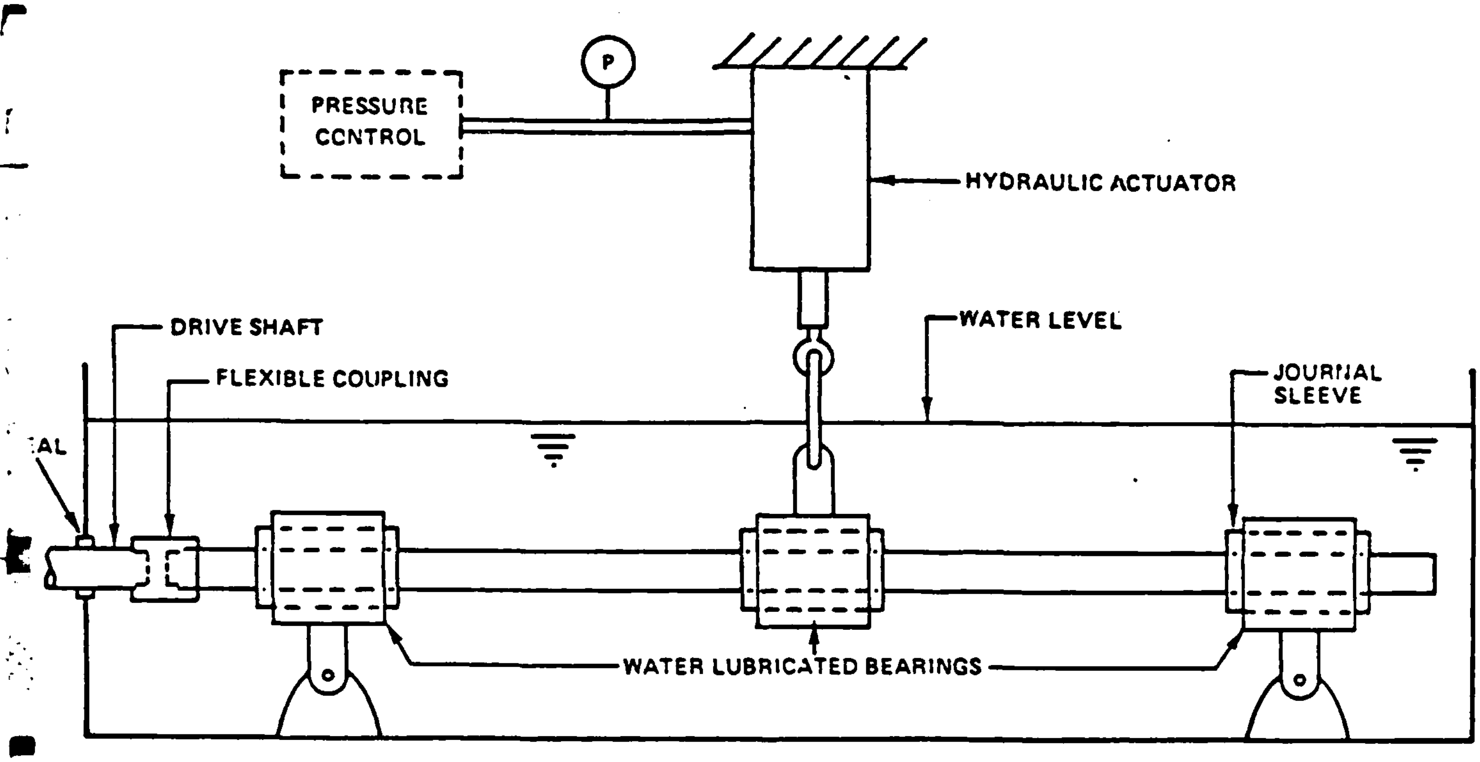


Figure 15. Bending Test Apparatus

Specimen A, Torsion Test

Test Sequence	Comments
1. Torque proof test	172,800 in.-lbs. (19 ksi torsional stress) full reversal, one cycle, no damage.
2. Fatigue test at 8 ksi torsional stress	Cycled at ± 88,000 in.-lbs. 14 sec./cycle for 1,000 cycles.
3. Torque proof test	172,800 in.-lbs. full reversal, one cycle, no damage.
4. Fatigue test at 13 ksi torsional stress	Cycled at 126,000 in.-lbs. one direction only for 6,600 cycles. Three random shoulder screws failed.
5. Ultimate torque test	Shoulder screws replaced shaft torqued to failure 290,000 in.-lbs. (30 ksi torsional stress).

Specimen B, Torsion Test

Test Sequence	Comments
Torque proof test	The outer rows of shoulder screws in the composite coupling failed at 130,000 in.-lbs. (14 ksi torsional stress) test stopped).

Specimen C, Bending Test

Test Sequence	Comments
Cyclic bending at 6 ksi bending stress	All radial bolts started falling out of shaft after a short time. Under reverse bending, the threads backed out of flange. The threaded section of shoulder bolts failed. Clamp rings were used to secure bolts and test resumed. At 1×10^6 cycles test stopped due to cracks from hole to hole in metal flanges.

Figure 16. Test Results Navy Propeller Shafts

characteristics. Specifically, the tapered metal flange should be adequately thick at the inboard end to carry the imposed bending load. This design change could be made without adversel_ affecting the torque carrying ability of the shaft.

5.0 SUMMARY

The current Navy design maximum allowable torsional and bending stress in a propeller shaft is 16 ksi and 6 ksi, respectively. The results presented indicate that the torsional requirement can be met with the bolted double flange arrangement. The pinned flange has merit if the pins are truly body bound in the composite to prevent movement. The tapered flange concept as used on the pinned flange and double flange tests provides equal load sharing between rows of fasteners when subjected to only torsional loads. Bending loads produce detrimental stresses in the thin section of the metal flange as presently designed which can exceed the material allowables.

The torque tests of the VIPER motor cases indicate that the wound around pin concept has the potential to carry required torque loads. Modification to wind angles could produce a usable concept.

The failure predictions are presented as a comparison for the experimental results. The predictions are higher in all cases than the experimental results but certainly within reason. The discrepancies between analytical and experimental results are most likely due to the uncertainties of fatigue allowables, stress concentration factors, and inefficiencies of fasteners that are not body-bound.

6.0 RECOMMENDATIONS/CONCLUSIONS

The results of this test program are certainly not conclusive but indicate areas where more information is needed. This work points out the need for a standardized test specimen with which to optimize coupling concepts. As indicated by these tests, metal couplings can certainly be designed to transmit the required loads in bending and torque. This is evidenced also by the success Boeing Vertol has had with their coupling design used on Brunswick built rotor shafts. The design consists of an internal metal sleeve with a pressed pin through the composite and sleeve wall. These units surpassed design requirements in both bending and torsion.

For certain load applications, it may be possible to use composite sleeves which will reduce the coupling weight. This concept, along with the wound around pin concept, needs to be evaluated in fatigue. A composite coupling has the obvious advantage of reducing coupling weight.

Specific recommendations for future programs are listed below:

1. Design and test metal flange to transfer increased bending and torque loads.
2. Design and test composite couplings for torque only applications.
3. Design and test wound around pin concept specimens in both bending and torque.

The contents of Appendix B will be included in the final report on a limited distribution.

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Jim McGee, Brian Spencer and David Shy
Brunswick Corporation
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Key Words
Filament Wound Con struction, Tracked Vehicle Shafts, Graphite, Fiberglass and Epoxy Resin.

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