Technical Note N-1343

STANDARDIZED HARDWARE FOR OIL SPILL
CONTAINMENT BOOMS

By

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June 1974

Naval Facilities Engineering Command
Alexandria, Virginia 22332

Approved for public release; distribution unlimited.

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TA
417
.N3
no. N 1343

Unclassified

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER TN-1343	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) STANDARDIZED HARDWARE FOR OIL SPILL CONTAINMENT BOOMS		5. TYPE OF REPORT & PERIOD COVERED Not final; Jan 1973-Dec 1973
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) F. J. Campbell		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS CIVIL ENGINEERING LABORATORY Naval Construction Battalion Center Port Hueneme, California 93043		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS 65-111
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Facilities Engineering Command Alexandria, Virginia 22332		12. REPORT DATE June 1974
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)		13. NUMBER OF PAGES 58
		15. SECURITY CLASS. (of this report) Unclassified
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE
16. DISTRIBUTION STATEMENT (of this Report) Approved for public release; distribution unlimited.		
17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)		
18. SUPPLEMENTARY NOTES		
19. KEY WORDS (Continue on reverse side if necessary and identify by block number) Hardware, containment booms, boom connector, towing assembly, boom-bulkhead attachment, oil spills.		
20. ABSTRACT (Continue on reverse side if necessary and identify by block number) This report describes the design, development, and testing of standardized hardware for use with existing and new Navy oil spill containment booms. This hardware which consists of boom connector, a towing assembly, and a boom-bulkhead attachment can be used to quickly interconnect and deploy oil booms of a wide variety of manufacturers.		

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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

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SECURITY CLASSIFICATION OF THIS PAGE(When Data Entered)

SECURITY CLASSIFICATION OF THIS PAGE(When Data Entered)

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INTRODUCTION

Under various Federal laws and Executive Orders the Navy is responsible for cleaning up its own spills, and assisting with men and equipment as directed to provide salvage capability for any private or government vessel that poses a pollution threat. The Navy is concerned with developing its own oil-spill-cleanup technology and capability because of the present lack of applied research and development of new procedures and hardware.

A 1969 study conducted by Battelle Memorial Institute, Pacific Northwest Laboratory, Richland, Washington, for the Civil Engineering Laboratory,* NCBC, Port Hueneme, California, disclosed that oil slicks were most effectively removed by the use of dispersants or by skimming the oil off. With dispersants banned except for use in hazardous spills, the Navy has pursued the development of better methods of physical removal. Rapid and effective containment of an oil slick by mechanical booms is of paramount importance to the successful removal of a spill. Therefore, one specific area of research is the design of more effective oil-containment booms.

The use of booms for passive containment, or as part of a recovery sweep system, requires well-designed and integrated boom components. Part of the Navy's problem has been the procurement of a wide variety of commercial oil-containment booms and, of course, the many new types of hardware associated with these booms.

Currently, each of the large number of oil-containment boom manufacturers has its own design for boom connectors and tow assemblies. Only a few have dock or bulkhead attachments available for use when the boom is deployed in areas with tidal fluctuations. A Naval field activity with more than one type of boom must fabricate special adaptors to inter-connect booms of different sizes or to attach a boom to a dock or bulkhead. Equipment adaptation is further complicated when different Naval commands join forces to combat a major spill. In addition, much of the hardware has exhibited serious design deficiencies, especially in the area of structural strength. The Navy is correcting this situation by updating its Military Specification for oil-containment booms.

The problem of equipment incompatibility can be solved by developing:

1. A universal connector, which would allow sections of different types and sizes of boom to be connected.

* Formerly the Naval Civil Engineering Laboratory.

2. A self-floating tow assembly, to provide easy attachment of tow cable during field use and to assure proper vertical loading of the boom end.

3. A bulkhead attachment, compatible with the universal boom connector and capable of maintaining a leak-proof seal at a dock or other vertical surface during vertical tidal variations.

CEL, sponsored by the Naval Facilities Command, undertook and completed the study of the designs, and then the development and testing of the above components.

DESIGN REQUIREMENTS FOR BOOM HARDWARE

Hydrodynamic forces experienced by a boom assembly during field use imposed restrictions on the design requirements of the boom connector and tow assembly. Appendix A discusses in detail the theoretical straight-line and catenary hydrodynamic forces for the boom which had to be considered for any successful design of the hardware.

All three components (boom connector, tow assembly, and bulkhead attachment) were to be compatible with each other. In addition, the following characteristics were identified as being essential to the design of the standardized boom hardware.

The boom connector was to have the following characteristics:

1. Insensitivity to decoupling when subjected to surge and heave forces in the field.

2. Easy attachment by field personnel from small craft.

3. Rotational flexibility to reduce bending loads in bidirectional tidal currents.

4. Fabrication materials resistant to marine corrosion, easily extrusible, and with high strength.

The tow assembly was also to be fabricated from high-strength, extrusible, corrosion-resistant materials, as was the boom connector. In addition, the design was to include the following characteristics:

1. Enhanced damping of roll motion.

2. Good low-speed control of the towed boom end.

3. Sufficient rigidity to resist the bending and buckling loads and yet be able to maintain vertical attitude.

4. Provision for proper towing load distribution over the vertical boom end.

Because boom-bulkhead attachments would be used primarily in confined harbor situations, the attachments were to have the following operational characteristics:

1. Easy adjustment to variations in water depth caused by tides.

2. Rotational flexibility about the track axis to minimize loads caused by bidirectional currents.

3. Bulkhead fitting to provide an oil-tight seal.

DESIGNS STUDIED

The structural requirements of the proposed designs, relating the size of components to the given materials, are analyzed in Appendix B. The design finally selected for development evolved from the information found in Appendix A and B and the findings from the study of the various hardware designs, as discussed below.

Boom Connector

Those designs considered in the CEL study for a universal boom connector to couple booms of different types and sizes* are shown in cross section in Figure 1. In each case, the boom end is clamped to the connector by means of a metal plate and bolts. All candidates have mounting holes for attachment of the primary tension member of a particular boom vendor. The boom end has a rod (flexible or rigid) running the full height of the boom to assist in keeping the connector attached to the boom under tensile loads. Each of the candidate designs shown have certain advantages and disadvantages which are tabulated in Table 1.

Towing Assembly

The two primary objectives of a separate towing assembly are the control of the towing load distribution over the boom height (depending upon the location of the primary tension member) and maintenance of the

* 12-inch (Type I, Class 1), 24-inch (Type I, Class 2), and 36-inch (Type II).

boom in a vertical attitude during sweeping to minimize loss of oil under the boom. A secondary objective is to provide a single point for towing during field use. The two towing assembly candidates considered are shown in Figure 2. The advantages and disadvantages are presented in Table 2.

Bulkhead Attachment

Bulkhead attachment assemblies are needed to provide a leak-tight seal with a dock or other vertical surface during tidal variations. The assemblies may also be used to provide a load-carrying seal between a skimmer craft and the connecting boom. The two candidates for this component are shown in Figures 3 and 4. Table 3 presents the advantages and disadvantages of the two designs.

DESCRIPTIONS OF SELECTED DESIGNS

All of the design candidates were determined to be structurally sound for the anticipated design loads. Therefore, the final selections of the recommended design from among the candidates was based upon the fulfillment of the design requirements for the boom hardware.

After review of the design candidates, the tubular assembly of the boom connector shown in Figure 1A was selected as the basic unit for the assemblies, primarily because of its advantage over the other candidates in its insensitivity to decoupling and its rotational flexibility. Figure 5 is a dimensional sketch of the recommended design.

Several items are of interest in the final design of the boom connector hardware.

Aluminum alloy 6061-T6 was selected for all assemblies because of its high strength and good extrusion properties. The 5000 series of aluminum alloys has a higher marine-corrosion resistance but are much lower in yield strength than the 6000 series. With the 6061-T6 alloy the connector is strong enough to withstand forces which would be placed on the connector during operation, yet it is light and requires no buoyancy tanks. The extrusions are cut to fit the overall height of each type of boom size, and all corners in contact with the boom end are rounded to minimize abrasion (see Figure 5).

The female connector tapers outward and upward to a wider diameter at the top to allow insertion of the male connector without the disadvantage of aligning the two connectors in a vertical or near-vertical position. This makes it relatively easy to connect the sections in an open-sea situation.

If the female and male connectors are pulled apart, sufficient elasticity remains in the jaws of the female connector to permit reuse of the connector though subsequent failure would occur at lower tensile loads.

A small retaining pin is used (see Figures 6 and 7) to prevent excessive relative vertical motion of the two sections of the connector.

A dacron line connected to the top of each connector makes them easier to lift out of the water when joining the boom connectors sections.

The self-floating towing assembly was recommended as shown in the dimensional sketch of Figure 8. The assembly presented in Figure 2B was not pursued because the use of a paravane-foot assembly to control the attitude of the towed boom end was considered impractical. The selected assembly is directly compatible with the male-connector section of the selected boom-connector assembly (Figure 5) and has a vertical plate near the bottom to assist in damping any roll motion (Figure 9). The towing assembly incorporates a female connector section welded to a single-tow-point float. It can easily be connected to or disconnected from deployed boom sections or a towing cable. Other advantages are its small size and its light weight.

The bulkhead-attachment design shown in Figure 10 fulfilled all of the requirements for the design. The same extrusion used for the boom connector was used for the sliding track. In this assembly, a female-connector section is welded to structural angle for attachment to a pier. Figure 11 shows the bulkhead attachment mounted on a piling in the harbor. A boom section can quickly be slid in or out of the top or bottom of the attachment.

It should be noted that the extrusion for the larger concentric tube (female connector) can be used for the bulkhead-attachment assembly on a towing vessel or oil skimmer so that a boom end supplied with the male extrusion can be connected directly to the towing assembly.

TEST PROGRAM

Prototypes of the boom connector, towing assembly, and bulkhead attachment were fabricated and subjected to an accelerated series of laboratory and field tests. These tests were designed to establish the adequacy of the prototype standardized boom hardware for its intended use and to verify the assumptions used in the design of the hardware. Laboratory tests determined the strength of the connectors and simulated conditions where leakage of oil could occur. Field tests simulated the force loadings and handling situations that would be encountered in actual field use. Thus, any necessary modifications to the assemblies were implemented before the equipment was put into field service.

Laboratory Tests

The laboratory tests, restricted to testing of the male and female boom connector, were conducted on a tensile test machine and in a small-wave tank, and quantitative and qualitative observations

were recorded. Three connector lengths (6, 12, and 18 inches) were subjected to tensile tests to determine whether the structural design requirements (see Appendix B) were met.

The strength required to separate the male and female connectors was measured. Failure occurred when the inner (or male) connector pulled out of the outer (or female) connector (see items 5 and 6 in Figures 6 and 7). The male connector bore against the outer ends of the female connector, thus springing the female connector open. After the tests, however, only a small permanent expansion was noted in this connector.

The connector sections were also tested for oil leakage at the area where they were joined (see Figure 12). In these tests for leakage, current speeds of 0.01 knot and 0.25 knot and a wave height of 2.7 feet were produced in the tank. Heavy and light weights of oil were used in the tests: 90-weight gear lube and 10 weight hydraulic oil, respectively.

Field Tests

All of the prototype hardware were field-tested to identify any operational faults in their design.

Harbor Tests. The boom-connector prototype was first tested for the ease with which two 3-foot connector sections, compatible with Type II boom, could be joined. The connector was installed on a 46-inch deep Kepner boom (Figure 13). A total length of 1,000 feet was available: two sections each 200 feet long and two sections each 300 feet long. The time required for two men in a small boat to join two sections of the boom together was measured (Figures 14 and 15): the connectors were easily joined together in 30 seconds.

The towing assembly was checked for ease of connection to a boom section. Then straight-line towing stability tests were conducted, using short lengths (200 and 400 feet) of boom (Figure 16) at tow speeds of 1, 2, 3, and 4 knots. The nylon towline length was 90 feet. The 400-foot oil-boom skirt tended to weave during the tow but did not turn on its side; however, the 200-foot length did turn on its side during the tests.

The 10-foot-long bulkhead-attachment assembly was fabricated and positioned on a pier piling in the harbor for the assembly to accommodate 6-foot tidal changes. During fabrication the assembly warped when the female connector was welded to the aluminum angle. A 200-foot section of Kepner oil boom was attached to the assembly to determine the ease in connecting the two pieces of male and female connectors together (Figure 17). The boom section was left connected to the bulkhead assembly for 6 hours and followed tidal variations with no difficulty.

Open Sea Tests. The Kepner boom sections were joined together to form a 1,000-foot length which was towed to sea in a straight line (Figures 18 and 19). At sea, a second boat attached a line to the free

end of the boom-towing assembly and formed a catenary with the first tow boat (see Figure 20). A 300-foot interval was maintained between the two boats and tow speeds of 0.2, 0.6, 0.8, 1.0 and 1.1 knots were established. Swells were 3 to 4 feet in height, and winds were 5 knots from the starboard quarter.

Measurements were taken of the tensile force developed in the center connector and in both towing cables. A calibrated tension unit (Figure 21) was mounted between the center boom connectors to measure the load at this location. The skirt tended to overturn and plane at a speed over 0.8 knot. At a speed of 1.1 knots the skirt tended to plane at the center of the catenary, and water spilled over the boom. At times the towing assembly tended to submerge during tow. A post-test inspection of the nuts and bolts used to hold the boom to the connectors showed they were galling and had siezed together.

RESULTS

Laboratory Tests

Figure 22 contains results of the load tests that were conducted on the 6-, 12-, and 18-inch lengths of boom connectors. The load required to separate the connectors is predicted to be 13,500 pounds for the Type I Class 1 connector, 29,000 pounds for the Type I Class 2 connector, and 43,000 pounds for the Type II connector.

In the oil-leakage tests, approximately 0.04 pint per hour leaked through the boom-connector joint when a 0.25-knot current applied tension on the joint. As the tension on the boom connector was reduced to 0.01 knot, the leakage through the joint increased to an estimated 0.5 pint per hour.

Field Tests

The measured straight-line towing loads for different lengths of booms and towing speeds and the theoretical towing loads predicted in Appendix A are shown in Figure 23. The largest load measured was 1,450 pounds which was recorded when 1,000 feet of boom was towed at 5 knots.

The average tensile loads in the one towing cable (recorded when 1,000 feet of boom was being towed in a catenary configuration at different speeds) is shown in Figure 24. The largest tow load, 1922 pounds, was measured when the boom was being towed at 1 knot. Beyond this speed the boom used in the tests turned on its side. Booms that do not turn on their sides at these and higher speeds would place a greater load on the towing cables and connectors. Loads recorded at the center of the

boom when it was being towed in a catenary configuration ranged from 500 pounds at 0.2 knot to 1,455 pounds at 1.1 knots (1,455 pounds is considered low as the boom had turned on its side at 1.1 knots).

The measured catenary tow loads and the loads predicted by the theoretical analysis in Appendix A are shown in Figure 24.

CONCLUSIONS

1. The boom connectors satisfactorily held the boom-skirt section together and were simple and easy to connect, even while two men worked from a small boat.
2. The towing assembly provided a satisfactory method of towing the oil boom in straight-line and catenary-tow configurations. This assembly was easily connected to and disconnected from the connectors on the oil boom.
3. The boom-bulkhead connector assembly was easily installed on a pier piling. An oil boom was quickly attached to the bulkhead connector, and the connector allowed the attached boom to follow tidal variations.
4. The boom towing loads measured during field tests confirmed the validity of the theoretical towing load predictions, Appendix A, that were used in the initial design of the hardware.
5. Oil did not leak through the hinge area of assembled connectors when a sufficient, and normally present, tensile force was applied to the connector. Oil leakage, which occurred when a sufficient tensile force was not present, was not significant; and a special gasket was not needed.
6. Galling and siezing of the bolts and nuts used to hold the oil boom to the boom connectors was due in part by the use of fine screw threads for the items. Coarse screw threads should be used instead.
7. The number 5-44NF round-head screws used in the connector assemblies stripped the threads tapped in the connectors because of the relative softness of the aluminum connector compared to the steel screw and the small size and therefore low strength of fine screw threads used. A larger screw with coarse threads would not strip the aluminum threads as easily.
8. The towing assembly submerged at times during the field tests because the towing load created a force couple in the assembly which could not be counteracted until the front end of the assembly submerged. This force couple could be counteracted by lowering the location of the tow hole on the assembly.

9. During field tests the towing cable chafed against the towing assembly because the shackle used to hold the cable to the assembly was too small to allow sufficient clearance. A larger shackle would prevent this problem.

10. Warping of the bulkhead assembly, which occurred during fabrication, was caused by the use of continuous fillet welds to join the female connector to the aluminum angle section. The use of staggered fillet welds would prevent warping.

RECOMMENDATIONS

1. The boom end connectors, towing assembly, and bulkhead-attachment assembly, modified as recommended below, should be procured and used as part of the Navy's oil-spill-containment program. An effort should be made to extend the use of this hardware to other Federal agencies and possible commercial users to reduce procurement costs and to prevent the occurrence of equipment incompatibilities in the future when these agencies join forces to combat an oil spill. A patent application which covers the end connectors, towing assembly, and bulkhead attachment has been filed and should be processed by the appropriate Navy agencies.

2. Alternative pier mounting configurations should be developed for the bulkhead-attachment assembly. The configuration used in the tests permitted the assembly to jut out from the piling where it could potentially cause damage to small craft and ships that came in contact with it.

3. Field activities that use these connectors should be instructed to take several sets of unmounted male and female connectors and bolt two male or female connectors together back to back. Situations will occur during field use where identical and therefore incompatible connectors must be joined together. (These extra sets of connectors (both male if the incompatible connectors are female, or vice versa) can be used to join the incompatible connectors together.)

4. The drawing modifications recommended in No. 2 through 7 above have been incorporated into the final fabrication drawings (Figures 5, 8, and 10) used for the procurement of the hardware as part of the Fiscal Year 1974 oil-boom procurement program.

- a. The number 3/8-24NF nuts and bolts used in the tests to attach the oil boom to the connectors should be changed to number 3/8-16NC nuts and bolts, and these parts should be coated with an antisieze compound to prevent galling and siezing.

- b. Number 6-32NC round-head bolts should be substituted for number 5-44NF bolts. The coarser threads of this bolt will decrease the probability of stripping aluminum screw threads.
- c. The towing hole of the towing assembly should be lowered 2 inches vertically to keep the front of the assembly from submerging when it is towed.
- d. Polyurethane foam should be placed inside the float of the towing assembly to prevent the assembly from sinking if damage should occur to the float section.
- e. The female connector of the bulkhead-attachment assembly should be attached to the angle section with 3-inch-length fillet welds spaced on 6-inch centers and staggered on opposite sides. This method will prevent the assembly from warping during welding.
- f. A 3/4-inch shackle should be substituted for the 5/8-inch shackle used in the tests. The larger shackle will give sufficient clearance between a towing cable and the towing assembly. The thickness of the bosses that are welded to the tow hole should be increased from 1/4 inch to 5/16 inch to maintain proper clearance between the shackle and the bosses.

ACKNOWLEDGMENTS

Mr. D. J. Graham for developing the overall conceptual designs and sizing the prototype bulkhead attachment and towing assembly.

Dr. J. J. Der for performing the analysis of boom towing forces (Appendix A).

Mr. E. P. Skillman for assisting in the structural analysis of connectors.

Mr. Kit L. Mack for developing the drawings of Figures 5, 8 and 10.

Mr. R. C. Towne and Mr. R. A. Bliss for the execution of the laboratory and field tests.

Mr. C. Quigley and G. S. Duffy for the instrumentation required in the laboratory and field tests.

Mrs. V. N. Spafford for compiling much of the material in this note and putting it in a readable form.

Figure 1. Boom-connector design candidates.

Figure 2. Towing-assembly design candidates.

Figure 3. Bulkhead-attachment design candidate No. 1.

Figure 4. Bulkhead-attachment design candidate No. 2.

Figure 5. Fabrication drawing of the selected boom connector.

Figure 6. Female (top) and male (bottom) boom connectors for joining sections of oil boom skirts.

Figure 7. Boom connectors locked together.

Figure 9. Towing assembly attached to boom.

BOOM BULKHEAD ASSEMBLY FIGURE 2

[illegible]

NOTES

1. ALL MATERIAL SHALL BE ALUMINUM 6061-T6 UNLESS NOTED.
2. ALL HOLES SHALL BE $\frac{1}{8}$ " NOMINAL DIAMETER UNLESS NOTED.
3. ALL DIMENSIONS ARE IN INCHES.
4. BREAK ALL SHARP CORNERS.
5. ALL ALUMINUM PARTS SHALL BE ANODIZED IN ACCORDANCE WITH MIL-A-8625, TYPE II CLASS 1.
6. TOLERANCES SHALL BE AS FOLLOWS, EXCEPT FOR WELDS:
1-RACE DECIMAL
±.030
7. ALL WELDS ARE NOMINAL.

Figure 10. Fabrication drawing of selected boom bulkhead-attachment assembly.

Figure 11. Boom-bulkhead connector mounted on piling in harbor.

Figure 12. Boom connectors installed on oil boom skirts for Laboratory tests.

Figure 13. Boom connectors joining oil boom sections.

Figure 14. Two men in small boat assembling two oil boom sections by coupling boom connectors.

Figure 15. Completing coupling of boom connectors.

Figure 16. Towing 200 feet of oil boom in harbor.

Figure 17. Oil boom skirt attached to bulkhead in harbor.

Figure 18. Oil boom skirt and boom towing connector assembled for sea tests.

Figure 19. Towing oil boom at sea in straight-line configuration.

Figure 20. Towing oil boom at sea in catenary configuration.

Figure 21. Calibrated tension unit installed at center of 1,000-foot boom.

Figure 22. Boom connector separation loads.

Figure 23. Straight-line towing forces* in oil boom.

* Theoretical forces calculated for C_F of .0049 and boom height of 3 feet.

Figure 24. Catenary towing forces* in oil boom.

* Theoretical towing force was calculated assuming significant wave height $H_{1/3}$ of 2 feet and catenary opening of 300 feet.

** Boom turned on its side.

Table 1. Comparison of Proposed Designs of Boom Connectors

Connector	Advantages	Disadvantages
Figure 1A	<ol style="list-style-type: none"> 1. Not susceptible to decoupling from horizontal surge forces acting along the boom. 2. A certain amount of swivel action possible. 	<ol style="list-style-type: none"> 1. Must slide full height of boom during assembly and disassembly.
Figure 1B	<ol style="list-style-type: none"> 1. Extruded shapes. 2. Easier fabrication than connector in Figure 1A. 3. Relatively insensitive to accidental decoupling. 	<ol style="list-style-type: none"> 1. Must slide full height of boom during assembly and disassembly. 2. No rotational freedom about axis perpendicular to the plane.
Figure 1C	<ol style="list-style-type: none"> 1. Two halves can be connected without sliding their lengths relative to each other. 2. Connector marketed commercially. 	<ol style="list-style-type: none"> 1. Vertical alignment for installing the small retaining pin through both connector halves may be difficult. 2. Only the retaining pin holds the sections together and could fail or be accidentally pulled out or lost. 3. Subject to decoupling from wide variety of surge forces always present in the field in event of pin failure.
Figure 1D	<ol style="list-style-type: none"> 1. Two halves can be attached without sliding their lengths relative to each other. 	<ol style="list-style-type: none"> 1. May prove difficult to align in field.

Table 2. Comparison of Proposed Designs of the Towing Assembly

Towing Assembly	Advantages	Disadvantages
Figure 2A	<ol style="list-style-type: none"> 1. Self-floating. 2. Paravane counteracts the vertical component of the towline force. 3. Roll damping plate. 	<ol style="list-style-type: none"> 1. None.
Figure 2B	<ol style="list-style-type: none"> 1. More compact than that of Figure 2A. 2. One end of boom connector used as tow plate without addition of paravane foot. 	<ol style="list-style-type: none"> 1. Not self-floating. 2. Flexibility of towing bridles (could make it difficult to attach under field conditions).

Table 3. Comparison of Proposed Designs of Bulkhead Attachment

Attachment	Advantages	Disadvantages
Figure 3	<ol style="list-style-type: none"> 1. Simple construction. 2. Same structural components as boom connector in Figure 1A. 3. Rotational flexibility about vertical axis, minimizing bending loads in tidal fluctuations. 	<ol style="list-style-type: none"> 1. None.
Figure 4	<ol style="list-style-type: none"> 1. Already marketed commercially. 	<ol style="list-style-type: none"> 1. Construction is more complicated than that of Figure 3. 2. Rotational flexibility about the roller track is quite limited.

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Appendix A

HYDRODYNAMIC FORCES

During field use, a boom assembly experiences forces resulting from the relative currents and wave action on the boom skirt and flotation and from wind action on the boom freeboard area.

In catenary tow, the hydrodynamic forces on the boom include the steady drag force caused by (1) the relative towing velocity and (2) the more complicated, unsteady surge and impact forces from the waves. Waves in an actual seaway are difficult to characterize in simple analytical form. Based on a statistical analysis, it is common to use the significant wave height of the one-third highest waves, $\bar{H}_{1/3}$,* as the significant wave height for the irregular ocean waves. An energy spectrum found to correlate the irregular waves of the ocean is the Pierson-Moskowitz spectrum [1]. This spectral description is used below to estimate the drag force on a boom under catenary tow.

For the straight-line tow, the drag force is estimated using the results for frictional drag, F_o , on a flat plate in turbulent flow.

CATENARY TOW

For a steady towing velocity the drag force on the skirt of the boom can be approximated by the relation for drag on a vertical flat plate, i.e.,

$$F_o = 1/2 C_D \rho v^2 hS \quad (A-1)$$

where C_D = drag coefficient

ρ = mass density of water (slug/ft³)

v = towing velocity (ft/sec)

h = skirt depth (ft)

S = catenary opening (ft)

For the total force including the effect of unsteady wave action, Miller, et al. [2] obtained an empirical relation for estimating the drag. The total drag force, F_D , on the boom skirt can be computed by the expression

* Significant wave height is determined by sampling the height of a population of waves over a given length of time, dividing the population into thirds by height, and then averaging the heights obtained for the highest one-third of the wave population.

The connector was modeled as a simple supported beam carrying a distributed load. This model was considered to be a valid representation of actual loading configuration as existing oil booms have tension members (cables, chains, etc.) built into the top and bottom of their structure that can be considered to act as simple structural supports. The connector loading model is shown in Figure A-2. For conservative design calculations, the distributed load was converted to a point load located 12 inches from the bottom of the 36-inch, Type II boom. This converted model is shown in Figure A-3.

The transverse hydrodynamic load on the connector is highest at the center of a catenary. This load is computed assuming the load is equal to that from the drag forces acting on a flat plate. The pressure distribution on the submerged portion of the boom is of the general shape shown in Figure A-1. The drag force per square foot of the connector due to this pressure distribution is calculated using the formula

$$\overline{F}_D = C_D \rho \frac{v^2}{2}$$

where F_D = drag pressure (lb/ft²)

C_D = drag coefficient

ρ = mass density of water (slugs/ft³)

v = tow speed (ft/sec)

The frontal area (A) of the connector assembly is 2.25 ft² for the largest Type II boom. The transverse hydrodynamic load or the drag force, on the connector assembly, is therefore

$$F_D A = 52 \text{ lb}$$

Figure A-1. Shape of the pressure distribution on the submerged portion of the boom.

Figure A-2. Loading diagram.

Figure A-3. Load model for sizing connectors.

$$F_D = F_o \left(1 + 0.59C \frac{H_{1/3} \omega_{\max}}{V} \right)^2 \quad (A-2)$$

where C = empirical constant

$\omega_{\max}$ = maximum frequency or maximum energy in wave spectrum (rad/sec) equaling $2.26/\sqrt{H_{1/3}}$ for Pierson-Moskowite spectra

V = towing velocity (KT)

$H_{1/3}$ = height of one-third highest waves

Assuming a Pierson-Moskowite spectra for the ocean and a value of $C = 0.23$ [2], Equation A-2 becomes

$$F_D = F_o \left(1 + 0.308 \sqrt{H_{1/3}/V} \right)^2 \quad (A-3)$$

Added to this is the wind force, F_W , which is

$$F_W = 1/2 C_D \rho_a v_a^2 f_s \quad (A-4)$$

where ρ_a = density (slug/ft³) of air

v_a = wind velocity (ft/sec)

f = freeboard height (ft)

Taking $C_D = 1.5$ and $\rho_a = 0.00269$ slug/ft³, Equation A-4 becomes

$$F_W = 0.00576 f_s v_a^2 \quad (A-5)$$

v_a is now wind velocity in knots.

The total drag force on a length of boom in catenary tow is given by the sum of Equations A-3 and A-5. For a 1,000-foot length of boom in a catenary with a 500-foot opening these equations were used to obtain the force values in Table A-1. Type I Class 1 (12 inch), Type I Class 2 (24 inch), and Type II (36 inch).

STRAIGHT-LINE TOW

For the straight-line tow boom towing configuration the force, F_D , acting on the boom, is given by the frictional drag on the skirt, i.e.,

$$F_D = C_F \rho v^2 h L \quad (A-6)$$

where C_F = friction coefficient

h = boom skirt depth (ft)

v = tow speed (ft/sec)

L = length of the boom (ft)

The value of C_F is dependent on the Reynolds number, $R = V L/\nu$, where ν is the kinematic viscosity (1.09×10^{-5} ft²/sec for seawater). It is noted that for a 1,000-foot length of boom the value of the Reynolds number is over 10^9 and the condition for straight tow is always turbulent.

In particular, at a tow speed of 10 knots, $R = 1.6 \times 10^9$, $C_F = 0.0015$, and Equation A-6 yields

$$F_D = 854 h \quad (A-7)$$

Equation A-7 was used to estimate the drag force on 1,000 feet of boom in a straight-line tow for each of the three boom types. The results are shown in Table A-2. It should be noted that the estimate given by Equation A-7 does not include the effects of unsteady forces caused by towing through the complex wave spectrum of an actual seaway. Limited data recorded in April 1972, during a field-acceptance test of boom of the same size as Type II indicated a towline force of 2,800 pounds when towing at 10 knots.

In any event, since the force loadings are much higher in catenary tow, they were used in the structural analyses described in Appendix B.

ANALYSIS OF TRANSVERSE HYDRODYNAMIC LOADS ON A BOOM CONNECTOR

For sound structural design the boom-end connectors must be sized to accommodate transverse hydrodynamic forces that are placed on the connectors when a boom is being towed in a catenary configuration (Figure A-1). A model that can be used in sizing connectors and an estimate of the magnitude of transverse hydrodynamic forces are determined below. Calculations are made for a connector located at the center of a boom catenary.

Table A-1. Forces Acting on 1,000 Feet of Boom Under Catenary Tow

Parameters	Boom Types		
	Type I, Class 1	Type I, Class 2	Type II
Tow Speed, knots	1	1.5	2
Wind Speed, knots	15	20	25
Wave Height, feet	0.416	0.75	1.0
Wind Force, pounds	269	864	1,800
Drag Force, pounds	2,231	9,823	25,390
Total Force, pounds	2,500	10,700	27,200
Towline Force, pounds	1,250	5,350	13,600

Table A-2. Forces Acting on 1,000 Feet of Boom in Straight Tow

Parameters	Boom Types		
	Type I, Class 1	Type I, Class 2	Type II
Tow Speed, knots	10	10	10
Skirt Depth, feet	0.667	0.333	2.0
Total Drag Force, pounds	570	1,140	1,700

Appendix B

STRUCTURAL DESIGN

Using the force loadings determined in Appendix A, the candidate designs for the boom connector, tow assembly, and bulkhead-attachment assembly were analyzed structurally to determine component size for 6061-T6 aluminum.

BOOM CONNECTOR

Structural calculations for the boom connectors included sizing the connector to withstand tensile loads along the boom, and transverse bending loads. The clamping area required to secure the boom end to the connector was also determined.

The structural analysis procedure applied to each of the four candidate connectors of Figure 1 is outlined below. For purposes of conservative design, the tensile force loads appearing in Table A-1 for the Type II boom were used to size each of the candidate connectors. Connector cross-sections sized for these loads can be used for the smaller booms and still provide adequate strength when subjected to smaller loads that occur when Type I, Class 1 and 2 booms are used. Extrusions of 6061-T6 aluminum with a modulus, E , of 10×10^6 psi and a yield strength of 35,000 psi were used for all calculations. The procedure used to determine connector size required to withstand tensile and transverse loads is outlined below.

1. Assume a nondimensional connector configuration from Figure 1.
2. Determine connector dimensions such that it will withstand the tensile load created by Type II boom as shown in Table A-1.
3. Determine the cross-sectional moment inertia required of the connector and its associated clamping plate to withstand transverse hydrodynamic loads.
4. Adjust the cross-sectional dimensions of connector and clamping plate to produce a moment of inertia equal to required moment of inertia developed in step 3 above.

The analysis outlined above is applied to the first candidate in the following discussion.

The maximum tensile load on the connector occurs during catenary tow. Using the load estimate for Type II boom from Table A-1, each tow-line exerts a tensile force of 13,600 pounds on the connector assemblies

close to the towed ends. The plate thickness, b , of the first candidate connector (shown in Figure B-1A) necessary to accept this load without failure is given by

$$b = \frac{F}{\sigma \ell} \quad (B-1)$$

where σ = tensile stress (psi)

F = tensile tow load (lb)

ℓ = boom height (in.)

Using a maximum allowable stress of 35,000 psi, a tow load of 13,600 pounds, and a boom height of 36 inches, Equation B-1 yields a required thickness of

$$b = 0.037 \text{ inch}$$

This calculation indicates that a very thin plate could be used if it were subjected to a pure static tensile load of 13,600 pounds. However, stress concentrations created by the presence of holes and attachment of primary tension members occur in practice which would require an increase in thickness over this value. From Reference 3 the stress-concentration factor for a hole one diameter away from the edge of a plate is about 3.6. A plate thickness of 0.25 inch, instead of 0.037 inch, is used in the remaining calculations, resulting in a safety factor of about 6.8 which should allow for the stress-concentration factors mentioned above.

The maximum stress on the large connector tube (see Figure B-1A) due to a tensile tow load occurs on the outside edge. The size of the tube wall is determined using the free-body diagram of Figure B-1B. From equilibrium considerations of this diagram, the wall thickness, t , is determined from the equation

$$\sigma = \frac{F}{\ell t} \left(3.5 + 3 \frac{R}{t} \right) \quad (B-2)$$

where the symbols are as defined for Equation B-1 or indicated in Figure B-1A.

Using the same values as used in Equation B-1 the wall thickness is given by Equation B-2 to be

$$t = 0.36 \text{ inch}$$

To determine what reinforcing was necessary to prevent transverse bending of the connector, some assumptions were made about the applied load. The connector was assumed to be a simply supported beam subjected to a point load of 52 pounds as shown in Figure B-2C (see Appendix A for details of this calculation).

The greatest stress occurs at the outside fiber of the beam at point A as indicated in Figure B-2B. The required moment of inertia about the neutral axis of the connector and any necessary reinforcing is given by

$$I = \frac{Mc}{\sigma} \quad (B-3)$$

where M = maximum bending moment in the connector

c = outer radius of the connector tube (in.)

σ = allowable stress (psi)

Inserting the values of these variables as determined in Appendix A the required moment of inertia, I , to prevent exceeding the allowable stress is

$$I = 0.46 \text{ in.}^4$$

The moment of inertia of the connector, using the dimensions of Figure B-1A, is

$$I = 1.12 \text{ in.}^4$$

and no additional reinforcing is necessary.

Similar computations were made for the other connector candidates. Dimensions for these are shown in Figure B-2.

Figure B-3 shows schematically the clamping and connector plates as they would be attached to a boom end. The connector plate has holes to accept the primary tension members (e.g., cable or chain) of a boom of a particular vendor. The boom end has a rod embedded in the fabric to assist in preventing the boom fabric from being torn loose by the tension force, T .

In order to conservatively size the width of the clamping plate, the friction force developed by clamping the fabric between two pieces of aluminum was assumed to be the only force available to keep the fabric from being pulled from between the two aluminum pieces. The required magnitude of this force can be calculated from Equation B-4 below which relates the width of the clamping plate to the force required per inch of height of the fabric.

$$T = 2\mu P_B W \quad (B-4)$$

where μ = coefficient of friction between fabric and metal

P_B = clamping pressure provided by the bolts (psi)

W = width of clamping channel (in.)

T = tensile load over boom height (lb/in.)

From Reference 4, the tensile load, F , in pounds that can be imposed on a bolt by hand tightening of a nut can be approximated by

$$F = 16,000 \times (\text{bolt diameter, inch})$$

Therefore, for 3/8-inch-diameter bolts the approximate load is 6,000 pounds. Using a bolt spacing of 3.5 inches to maintain a nearly uniform clamping pressure results in an "influence area" of 9.65 inches square and a clamping pressure, P_B , of about 625 psi for each bolt.

Typical values of μ range from 0.04 (Teflon on steel) to 0.61 (leather on oak). Assuming a value for μ of 0.10 and the value of P_B given above, Equation B-4 becomes, after solving the width, W ,

$$W = \frac{T}{125} \quad (B-5)$$

As shown in the hydrodynamic drag calculations of Appendix A, the maximum tensile load occurs for the Type II boom in a catenary tow. With the assumption that 50% of this load will be carried by the fabric and 50% by the primary tension members, the resulting vertical loading, T , is 190 lb/in. of fabric. Inserting this value of T in Equation B-5 gives a maximum required clamping channel width of

$$W = 1.5 \text{ inches}$$

Therefore, to assure that the fabric boom is not pulled out of the connector, a clamping plate at least 1.5 inches wide using 3/8-inch bolts on 3.5-inch centers is required.

TOWING ASSEMBLY

A serviceable towing assembly must be rigid enough to resist the bending and buckling loads that will occur in field service, must still be able to maintain the boom in a vertical attitude, and must provide a proper towing load distribution over the vertical boom end.

The tow assembly design of Figure 2A was analyzed for structural adequacy and for performance using the force loading diagram of Figure B-4. Figure B-4A is a diagram of forces on the entire assembly, and Figure B-4B is a free body diagram of the vertical member to which the boom is attached. This is the member most likely to fail under the loads encountered during field use.

Referring to Figure B-4A, expressions can be written for the vertical and horizontal force components and solved for the unknown paravane area, A, (an inclined flat plate) by eliminating the towline force, F_T . The resulting expression is

$$A = \frac{2(F_B + F_D)}{\rho v^2 C_D \left(\sin^2 \alpha \cos \alpha \frac{\cos \theta}{\sin \theta} - \sin^3 \alpha \right)} \quad (B-6)$$

where F_B = tension force exerted by boom (lb)

F_D = drag force of tow assembly alone (lb)

ρ = mass density of water (slug/ft³)

v = tow speed (ft/sec)

C_D = drag coefficient for flat plate

θ = towline angle

α = paravane angle

The highest straight line tensile tow force occurs when Type II boom is towed at 10 knots. In this case, the tensile tow load, F_B , is estimated from Appendix A to be 2,800 pounds. The estimated drag force of the assembly itself was calculated to be 254 pounds. Using these values and a towline angle of 5 degrees, a paravane angle of 55 degrees, and a flat-plate drag coefficient of 1.28 [5], the paravane area, A, necessary to counteract the vertical component of the towline force is

$$A = 2.2 \text{ ft}^2$$

The required areas for Type I, Class 1 and Type I, Class 2 would, of course, be smaller. The paravane angle of 55 degrees was chosen to maximize the terms involving α in Equation B-6. Changing the towline angle 5 to 10 degrees increases the vertical towline force component and increases the required paravane area to 5.1 ft². Thus, for the paravane to be effective, a small towline angle (long scope on towline) is essential.

For a catenary tow at 2 knots, the largest forces occur with the Type II boom. Using the tensile load of 13,600 pounds, from Table A-1 and the same data for angles and drag coefficient as above, the required area, A, becomes

$$A = 244 \text{ ft}^2 \quad (\text{B-8})$$

Thus a realistic paravane is clearly inadequate to counteract the vertical component of the towline force for Type II boom at the low speed of 2 knots. For the smallest Type I Class 1 boom in a catenary tow the required area is 2.7 ft^2 . Keeping the towed boom end from rising up under the action of the vertical force component and then lying over is more important in the catenary sweep mode than in the straight-line tow since only in the catenary sweep mode is oil being contained. Since the paravane areas determined for this operating mode are quite large, even for the small Type I Class 1 boom, it was decided that no paravanes were to be used and this mode of failure avoided by maintaining as long a scope as is necessary to prevent the boom end from being pulled out of the water to an unstable roll condition.

The structural analysis of the tow assembly is based on the free-body diagram of Figure B-4A. The vertical strut of Figure B-4B is considered as a beam rigidly supported at its ends with the uniform load, F_B , along its length. The equation for deflection at the center of such a beam is

$$Y_{\max} = \frac{1}{384} \left(\frac{F_B \ell^3}{EI} \right) \quad (\text{B-9})$$

where $Y_{\max}$ = maximum deflection at center of beam span (in.)

F_B = load imposed by boom (lb)

ℓ = length of strut (in.)

E = modulus of elasticity (psi)

I = area moment of inertia of beam cross section (in.^4)

In addition, the maximum fiber stress on the outer surface of the strut due to the applied bending load, F_B , is given by

$$\sigma_{\max} = \frac{F_B \ell C}{24 I} \quad (\text{B-10})$$

where $\sigma_{\max}$ = maximum fiber stress in strut

C = distance from neutral axis to outer surface of strut

Eliminating F_B between Equation B-9 and Equation B-10 results in an expression for the maximum deflection, $Y_{\max}$, associated with the given maximum stress, $\sigma_{\max}$, for a given material.

$$Y_{\max} = \frac{24}{384} \left(\frac{\sigma_{\max} \ell}{CE} \right) \quad (B-11)$$

The maximum allowable design stress for 6061-T6 aluminum is $\sigma_{\max} = 35,000$ psi. Assuming, as before, a design safety factor of 3.5, the maximum stress used in Equation B-11 is 10,000 psi. Setting the value of $\ell = 36$ inches, $E = 10 \times 10^6$ psi, and $C = 2.5$ inches, the maximum deflection is

$$Y = 0.009 \text{ inches}$$

which is quite acceptable for the functional aspects of the tow assembly design. Inserting $Y_{\max}$ into Equation B-9 or using Equation B-10 directly, the required strut cross-section moment of inertia, I , can be found to be $I = 5.1 \text{ inch}^4$. This moment of inertia value is achieved if the dimensions of Figure B-4B are: $r_o = 1.1$ inch, $r_i = 0.74$ inch, $t = 0.5$ inch, and $\ell = 2.5$ inches. To check for the possibility of buckling of the strut when under tow, the maximum allowable compressive end load for the particular strut is calculated. From Reference 3, the relation for the maximum end load for column buckling is

$$P_{\max} = 18,000 - 120 \frac{\ell}{K}$$

where $P_{\max}$ = maximum end load (psi)

ℓ = length of column (in.)

k = radius of gyration (in.)

Using the dimensions of the column strut as determined above the resulting end load for buckling is

$$P_{\max} = 14,580 \text{ psi}$$

The cross-sectional strut area is 2.98 in.^2 , so the compressive load required to buckle the tow assembly strut is over 43,000 pounds. Since the total tow load as estimated in Table A-1 never exceeds 14,000 pounds, failure of the strut by buckling is highly unlikely.

BULKHEAD ATTACHMENT

The bulkhead attachment candidates were analyzed structurally for their ability to withstand the tensile and bending loads indicated in Figures 3 and 4.

Referring to Figure 3, if the dimensions for the split tube sliding track are the same as for the first boom connector candidate discussed in this Appendix, the assembly can safely carry the full Type II boom tensile tow load appearing in Table A-1. Using a free-body diagram similar to that employed for the boom connector calculations, it can be determined that the point of maximum stress in the assembly of Figure 3 when subjected to bending loads occurs at the outer diameter of the sliding track near the support angle. To maintain the design safety factor of 3.5, this maximum stress must be held to 10^4 psi. The maximum allowable bending load, directed perpendicular to the tensile load shown in Figure 3, is calculated to be 1,740 pounds, which should be sufficient because the bulkhead attachment will be used in harbor situations where the currents and boom lengths are relatively low. It should be noted that from Table A-1 for 1,000 feet of Type I Class 1 boom (which is the type most likely to be used with the bulkhead attachment) the towline force is 1,250 pounds, or almost 500 pounds less than necessary for a design safety factor of 3.5.

The design candidate of Figure 4 was studied in the same manner as described above. It was found that under a pure tensile load of 13,600 pounds (see Table A-1) a plate thickness of 0.019 inch was required for the roller assembly. As in the design candidate of Figure 3, the more strict structural requirements were derived from the bending loads the assembly would experience with fluctuating tidal currents. If the plate thickness of the roller assembly is 0.25 inch, the maximum allowable bending load is calculated to be 1,500 pounds, similar to the allowable design load for the candidate of Figure 3. The roller track is quite rigid, and will support the full 13,600-pound bending load with a flange 0.25 inch thick by 2 inches wide.

Figure B-1. Boom connector loading.

Figure B-2. Structural sizing of candidates.

Figure B-3. Boom-end clamping detail.

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