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FIAT S.p.A., Sezione Energia Nucleare — Torino
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COLLISION TESTS WITH SHIP MODELS

TOPICAL REPORT

1971

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European Atomic Energy Community — EURATOM
Fiat S.p.A. — Sezione Energia Nucleare — Turin
Società Ansaldo S.p.A. — Genoa
with the participation of:
Comitato Nazionale per l'Energia Nucleare — CNEN, Rome

TOPICAL REPORT ON COLLISION TESTS WITH SHIP MODELS

1971

Prepared for publication by the Centro per gli Studi di Tecnica Navale (CETENA), Genoa, for the Directorate-General for Industrial, Technological and Scientific Affairs of the Commission of the European Communities

ABSTRACT

The Euratom/Fiat/Ansaldo contract of association for the development of a nuclear-powered tanker included the execution of a series of collision tests with models for the purpose of evaluating the effectiveness of various types of ship-side structures designed to protect the reactor space in the event of collision.

During the initial stage of this part of the programme it became clear that a number of technical problems relating to the carrying out of such tests had to be resolved before any conclusive information could be derived from them. A series of 14 collision tests were performed between May 1963 and July 1970; the conditions for each test were specified only after analysis of the results obtained with the previous one, only one particular parameter being changed at a time. A set of twe steel models on a scale of 1 : 15, consisting of a ship's bow-section and a section representing the collision protection structures in the side of the reactor compartment of a nuclear ship, were constructed for each experiment.

The report contains an account of the general criteria adopted for these experiments and the particular considerations underlying the parameter variations. A detailed description of the results is given and a tentative correlation with the full-scale example elaborated.

KEYWORDS

NUCLEAR MERCHANT SHIPS
REGULATIONS
MOCKUP
TESTING
COLLISIONS
MECHANICAL STRUCTURES
DEFORMATION
SAFETY
MEASURED VALUES

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Notice to Readers

A set of seven white prints of drawings on 1:10 scale, giving detailed information on the extent of the damage caused to the corresponding models during collision tests No. 8-14 and indicating the plate-thicknesses and dimensions of their structural parts, can be obtained separately from the Commission's Centre for Information and Documentation. These are obtainable together with full-size copies of the complete acceleration and speed diagrams as recorded for each of the 14 tests described in the present report, in which a reduced scale example of such a diagram is given in Fig. 5.

These drawings and diagrams can be obtained from the CID at the address below at a price of B.Fr.500.- When ordering specify clearly : "Supplementary information to topical report on collision tests with ship models - EUR 4560e".

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1. INTRODUCTION

1.1 General

The Euratom/Fiat/Ansaldo research contract concerning studies on a nuclear-powered ship and her reactor, in which the Comitato Nazionale per l'Energia Nucleare (CNEN) also participated, was signed in Brussels on 15 December 1961.

A tanker of about 53,450 tdw was selected by common agreement. After slight modifications during the design stage, this ship had the following characteristics:

Length between perpendiculars	230.00 m
Breadth	31.00 m
Depth to the main deck	17.30 m
Draught (summer freeboard)	12.00 m
Block coefficient	0.785
Full load displacement	69,200 tonnes
Displacement in ballast	50,000 tonnes

A two-volume report was published in June 1965 on the intermediate design of the tanker, the pressurized water reactor and the conventional-type main propulsion unit, which has a normal output of 23,600 metric hp and a maximum output of 29,000 metric hp. This report covered all the studies performed between January 1961 and June 1965.

In June 1968 a supplementary report was published concerning the studies carried out by Fiat and Ansaldo between July 1965 and the end of 1967.

In this supplementary report emphasis was placed on a particular aspect of the studies carried out by Ansaldo on hull structures, namely, collision tests with models in order to verify, by means of experiments, the strength of ship side structures intended for the nuclear-powered tanker designed by Ansaldo when struck by a ship of similar dimensions as the struck tanker but at various displacements and speeds and with two types of striking bow: a normal bow with a waterline entrance angle of 64° and an ice-strengthened bow with a waterline entrance angle of 38°.

These collision tests, which were described in the above-mentioned supplementary report and are referred to again in this report, were numbered from 1 to 5 and from 7 to 9.

Collision test No. 6 was carried out with the purpose of reproducing on models an actual collision between a striking ship displacing 11,000 tonnes and a struck tanker in ballast (8,500 tonnes displacement).

This also will be returned to again later, it merely being noted here that the two models for test No. 6 were reduced to one-tenth of the ships' actual sizes, while all the models for the other collision tests were on a 1/15 scale.

The idea of using a striking bow of a type similar to that of the struck ship was abandoned when tests No. 10, 12, 13 and 14 were planned. In these tests two other types of bow were employed which were considered to be more dangerous in the event of collision, namely, the bow of a transatlantic liner with a waterline entrance angle of about 20° and the bow of a large tanker, with a large bulb and an entrance angle of about 86° at the full-load waterline.

The results of all the collision tests carried out by Ansaldo were described in the report published in 1971 on the studies performed by Fiat and Ansaldo hitherto.

1.2 Collision tests on models performed by other researchers prior to or in the same period as, the Ansaldo tests.

1.2.1 Collision tests performed in Japan at the Yokohama Shipyard and Engine Works

(see "Research on the collision-resisting construction of ship sides" by K. Kagami, T. Hamada et al., Symposium on Nuclear Ship Propulsion, Taormina, Italy, November 1960).

Tests were carried out with models on a 1/20 scale. The shipside in way of the reactor space and the bow represented by the models were hull structures of 45,000 tdw tanker, both ships being assumed to be fully laden.

Eight shipside models were tested, each having differently designed hull structures between the outer shell and the inner longitudinal bulkhead bounding the reactor space. The model of the striking bow was always of the same type.

The experimental apparatus consisted of a high tower from which the striking bow was suspended by means of swinging arms. The shipside model to be struck was fixed to a rigid mounting and the bow model struck it at the end of its run (see Fig. 1). The angle of impact between the colliding bow and the struck ship side was always 90^{o}.

The speed of the striking ship at the moment of impact was about 5 knots, since it was considered that most collisions occur near the coast or in port, and furthermore the speed of the striking ship is presumably reduced as a result of action taken as soon as an accident is seen to be likely.

This opinion seems to be rather optimistic, but the tests carried out are very interesting and the results can be extremely useful for comparative purposes.

The tests showed that better results (smallest penetration into the shipside and largest bow deformation) were obtained when the outer side plating was thicker (32 instead of 21 mm i.e., Rule thickness for the ship's actual size) and strong side girders were fitted. It was considered that the outer shell plating acted as a membrane. The poorest result was recorded on a model having an outer side plating 21 mm thick (Rule thickness) and several light side girders.

As could be easily anticipated, the heaviest damage to the inner longitudinal bulkhead demarcating the reactor space occurred when the shipside structures were fairly rigidly connected to the above bulkhead.

The similarity law adopted when planning these collision experiments was the "explosion theory", according to which, if $\lambda = 20$ is the model scale, the weight of the striking model is assumed to be equal to $\frac{1}{\lambda^3}$ of the actual ship weight and its speed to be equal to the actual speed of the ship (5 knots = 2.57 m/sec, arbitrarily reduced to 2.15 m/sec in order to account for the fact that the struck ship has not really a rigid structure but an elastic one and can move sidewards when struck, thus entraining a mass of water which it is almost impossible to estimate).

1.2.2 <u>Collision tests performed by the Istituto di Construzioni Navali (Shipbuilding Institute) of Naples University</u>

See the papers by Prof. Franco Spinelli "Défense des réacteurs nucléaires de navire contre les abordages" (Protection of marine nuclear reactors against collision) published in the "Bulletin de l'Association Technique Maritime et Aéronautique, No. 62, Year 1962 Session" and "Protection du compartiment du réacteur nucléaire contre les abordages. Résultats d'essai sur modèles" (Protection of nuclear reactor compartment against collision - Results of model tests) published in the "Bulletin, No. 64, Year 1964 Session".

These tests were carried out by means of an experimental apparatus which was different from the one employed in Japan for the experiments previously described.

As can be seen in Fig. 2, the ship side to be struck was mounted on a frame on a carriage which was free to move after the impact and was fitted with three drift fins immersed in water tanks on both sides.

By means of such a system the lateral motion of the ship side model when struck entrains a mass of water (which is a variable quantity according to the level of water in the tanks), as occurs in the event of an actual collision at sea.

The fins must have an area corresponding to the drift area of the struck ship reduced to the model scale.

The bow model was placed on a carriage which runs down a slope of variable height. The weight of the bow model, suitably ballasted, has to be varied according to the displacement adopted for the colliding ship.

All five sets of models described in Prof. Spinelli's paper published in 1964 were built to 1/15 scale.

Both the struck ship side and the bow of the colliding ship were considered as belonging to a fully loaded 45,000 tdw tanker (60,750 tons displacement). The waterline entrance angle was 65^{o}.

The structural elements were joined together by means of a 60% tin and 40% lead solder, which tests to determine tensile stress indicated to be a suitable choice.

Several speeds, ranging from 2.42 to 5.80 m/sec (4.7 and 11.275 knots), were adopted for the striking ship.

To permit valid comparisons, the type of ship side structure chosen was the same as that tested in Japan (T 3 model, see Fig. 3).

In the fifth test the structure was changed by placing the longitudinal bulkhead demarcating the reactor space at a shorter distance from the outer plating (about 6 m instead of 7.50 m).

The similarity law followed was the explosion theory, as for the tests carried out in Japan. According to this theory the ship's speed is the same as the model speed.

In each test the following data were recorded:

- values of speed and acceleration, for both the striking and the struck models,
- stresses recorded by means of strain gauges.

The virtual mass of the struck model (ship + entrained water) was also calculated from the recorded speed and acceleration data; in this way the percentage of the water entrained by the struck ship relative to the mass of the ship was estimated (see Section 4.1.1).

This percentage, which was less than 10% in the lower speed range, was estimated to be about 50% for the highest speed used.

Results obtained from collisions of models of the same type at the same speed in the tests carried out both in Japan and in Naples are very similar and fully comparable, notwithstanding the different systems employed to perform them.

Even at the highest speed (more than 11 knots), the damage caused to the lower part of the ship side model did not affect the inner longitudinal bulkhead.

1.2.3 Collision tests performed by the "Deutsche Werft" at Hamburg on behalf of the GKSS (Gesellschaft für Kernenergieverwertung in Schiffbau und Schiffahrt, Hamburg). (See article "Kollisionsversuche mit Schiffsteilmodellen" by G. Woisin, published in "Kerntechnik, Isotopentechnik und Chemie", No. 8, 1967).

Tests were carried out with models on a scale of 1/7.5. This scale was adopted in order to make electric welding easier owing to the larger dimensions of the models.

The model of the ship side, as can be seen in a figure printed in the above article, reproduced the side structure of the German nuclear ship "Otto Hahn", while the bow was considered as being that of a transatlantic liner (Bremen). The displacement of the striking ship was estimated at 30,000 tonnes and the displacement of the struck ship 25,000 tonnes. The speed of the striking model was 5.95 m/sec (11.6 knots).

The experimental apparatus employed for performing these tests was of the same type as that built by the University of Naples (see Fig. 2). The ship side model was fixed to a rigid mounting.

Fig. 4 taken from the above-mentioned article, shows the models after the collision. In the diagram the structures of the models are shown schematically.

The author of the article states that about 300 strength tests had previously been carried out on separate structural elements of the collision protection.

Some similar collision tests have been performed, but their results have not been officially reported.

As is stated in Section 4.1.9 below, a bow model identical to that used in the above test performed by Deutsche Werft was built for test No. 10 carried out by Ansaldo, making use of the experimental apparatus available at Naples University.

2. SPECIAL REQUIREMENTS FOR THE CONSTRUCTION OF NUCLEAR SHIPS ISSUED BY CLASSIFICATION SOCIETIES

Before describing the 14 collision tests performed under the Euratom/Fiat/Ansaldo contract and before deducing proper technical considerations from these tests, it was felt desirable to outline here the requirements published by various Classification Societies as a guidance to the design of nuclear-powered ships in order to attain safety when a collision occurs in way of the reactor compartment.

2.1.1 American Bureau of Shipping

First of all it has to be established that the collision protection is so effective that the longitudinal bulkhead bounding the reactor space is not damaged as a result of the collision.

To ensure this, the protection is to be longitudinally extended so that it can also be efficient if the colliding ship has an angle of impact of between 30 and 150° with the struck ship. The protection must extend vertically so as to preclude any damage both to the high and to the low portions of the reactor vessel.

The longitudinal bulkhead limiting the reactor space must be mounted not less than 20% of the midship breadth from the ship side plating. The reactor space should preferably be located in the after part of the ship; the height of the double bottom must be not less than that laid down by the Rules.

Particular requirements are imposed concerning the absence of vibrations and to assure longitudinal strength.

The American Bureau of Shipping only requires that actual collisions regarding struck ships having dimensions similar to those of the designed vessel are to be examined. As to the colliding ships, a T2-type tanker travelling at a speed of 15 knots is mentioned. The speed of the struck ship is to be taken as the design speed.

However, the American Bureau of Shipping will always examine results and conclusions derived from actual collisions and calculations, if any, submitted by the designer.

2.1.2 Bureau Veritas

Bureau Veritas requires that the ship side structures should not be directly connected to the longitudinal bulkhead which bounds the reactor space and which must be fitted at not less than 20% of the ship's breadth from the side plating. Furthermore, the Bureau Veritas will review the strength of all special structures, drawings of which are to be submitted. Particular attention is to be paid to the longitudinal bending moment. The double bottom in way of the reactor space should be as high as possible. The plating of the outer shell is to be of special steel.

The collision protection must extend longitudinally 3% of the ship's length beyond the boundaries of the reactor compartment.

Furthermore, Bureau Veritas will carefully examine hull vibrations on the basis of data recorded during sea trials.

2.1.3 Det Norske Veritas

Det Norske Veritas requires that the collision protection near the reactor compartment is to be extended longitudinally 10% of the ship's breadth beyond the boundaries of the compartment and vertically from the highest deck forming part of the protection to a height from the bottom equal to 40% of the height of the above-mentioned deck.

The very last part of this requirement does not seem to be in accordance with the results of the test performed with ship models; as will be seen elsewhere in this report, these tests have shown that the heaviest collision damage occurs in the lower part of the ship side, especially when the striking ship is a fully loaded tanker and more particularly if the tanker has a bulbous bow, which is normal practice nowadays.

Furthermore, Det Norske Veritas requires that a longitudinal bulkhead be fitted as a shield for the reactor space. To this end, side platform decks and large side stringers are to be provided at a distance apart of not more than 3 m; stringers must be of a strength calculated by means of a special formula taking into consideration the ship's displacement.

2.1.4 Germanischer Lloyd

This Classification Society requires that the reactor compartment be located not less than 20% of the ship's length from the stern and not less than 30% of the ship's length from the bow.

In accordance with the Rules published by other Classification Societies, longitudinal bulkheads bounding the reactor space are required to be fitted not less than 20% of the ship's moulded breadth from the outer shell.

A structural protection is to be provided for the ship side which should not be directly connected with the above-mentioned longitudinal bulkheads and is located not less than 1 m away from them, or 5% of the ship's moulded breadth. The compartments adjacent to the longitudinal bulkheads bounding the reactor space must always be empty, i.e., they must not be used for liquids.

The longitudinal strength of the ship must be carefully examined and approved, and hull vibrations must be reduced to a minimum.

The hull should preferably be longitudinally framed. Electric welding must be checked radiographically (not less than 25% of all joints).

The reactor compartment must be limited fore and aft by means of cofferdams not less than 1.50 m long. Furthermore, the protection structure must extend longitudinally for 20% of the maximum breadth fore and aft of the reactor vessel and vertically it must protect the height of the same vessel.

The reactor compartment must be gastight and every hole or passage cut through the structures bounding it must be specially approved.

2.1.5 Lloyd's Register of Shipping

Lloyd's Register of Shipping requires that the longitudinal strength be carefully examined.

The midship section must have a minimum section modulus in accordance with the Rules, but in general the arrangements and loading should be so adjusted that the maximum stress in still water does not exceed 90% of the normal maximum value according to the Rules.

Furthermore, Lloyd's Register requires that special quality steel be used for the protecting structure in way of the reactor compartment and that all structures bounding the reactor compartment be entirely welded.

Lloyd's Rules also require longitudinal bulkheads bounding the reactor space and connected fore and aft to suitable longitudinal structures. No opening is to be provided for access to the reactor space in these bulkheads, which are regarded as forming an integral part of the protection structure of the reactor space. There must be a distance of 20% of the breadth of the ship, or at least 3 m, between the outer shell and the longitudinal bulkhead bounding the reactor space or, in other words, a minimum of 1.50 m between the outer shell and each intermediate longitudinal bulkhead.

The reactor must be enclosed by a gastight vessel, which if possible should not be connected to the surrounding hull structures, in order to avoid a deformation in the same structures causing a deformation in the reactor vessel.

However, the collision-protection structures must have scantlings which are based on particular strength and continuity criteria; they must also be specially examined and approved.

Compartments between the ship's side and the reactor space are not, as a rule, to be used for liquids.

2.1.6 Nippon Kaiji Kyokai

Particular care must be taken in designing the ship's longitudinal strength and the continuity of the longitudinal hull structures in way of the reactor space, which is not to be contiguous to fuel tanks or compartments intended for carrying explosive substances.

Cofferdams must be provided with flooding facilities; however, careful consideration must be given to the effects that a flooded cofferdam could have on a collision.

Longitudinal bulkheads limiting the reactor space are required.

Protecting structures must extend for 20% of the ship's breadth fore and aft of the reactor space and must not be directly connected with the inner longitudinal bulkheads in order to avoid deformations being transmitted from the ship side protecting structures when damaged in the event of collision.

2.1.7 Registro Italiano Navale

In 1969 the Registro Italiano Navale published some guidelines for the design of nuclear-powered ships; these guidelines were confirmed as mandatory in 1970.

In the main they are similar to the rules issued by other Classification Societies.

The following requirements are specified:

a) The hull structures in way of the protected zone are to be built with special steel.

b) At least 25% of all the electric weldments made in the protecting structures must be checked radiographically.

c) Careful consideration must be given to all stresses due to longitudinal bending of the ship.

d) Longitudinal bulkheads must be fitted on each side of the reactor space; they must contain no openings and must be located not less than 20% of the ship breadth from the outer shell.

e) Tanks which are contiguous to the longitudinal bulkheads must normally be kept empty, but the possibility of carrying liquids in them can be examined.

f) The reactor compartment must be located as far aft as possible, generally between 20% of the ship's length from the stern and 35% from the bow.

g) In way of the reactor compartment special protective structures must be fitted which are not directly joined to the longitudinal bulkhead bounding the reactor space so that deformations due to collision can not be transmitted by damaged ship side structures to the above-mentioned longitudinal bulkheads. The collision protection as a rule must extend at least 20% of the ship's breadth, but not less than 2.50 m, fore and aft of the reactor compartment.

With regard to the collision protection mentioned under g) above, it should be specially noted that Registro Italiano Navale requires that the builder of a nuclear-powered ship must previously determine, with its approval, the following characteristics of the type of collision which is to be taken into consideration:

- value of the striking masses
- impact velocity
- characteristics of the colliding bow
- geometric characteristic of the impact

Furthermore the following requirements are specified:

"The estimate of the actual efficiency of protecting structures against collision must be based on experimental tests with models; these tests are to be prepared and performed with the cooperation of Registro Italiano Navale, unless the shipbuilder can refer to data which are esteemed sufficiently valid by this body.

"The criteria adopted for the testing procedure or for the studies from which the above-mentioned data have been derived, must be previously made clear and agreed upon by Registro Italiano Navale.

"In particular the similarity criteria must have been submitted which are to be adopted to guarantee a correct transfer of results from tests with models to full-size conditions, both for the strength of the structure and for the dynamic characteristics of the collision.

"The hypotheses must also be specified which have led to the choice of the constraining conditions simulating the reaction of the medium (water) and the structural continuity in the case of incomplete or partial models'

According to the above requirements, Registro Italiano Navale acknowledges the importance of tests on models and the validity of the results obtained from the above tests if based on technical criteria having an established validity.

3. GENERAL CRITERIA ADOPTED FOR COLLISION TESTS WITH SHIP MODELS CARRIED OUT UNDER THE EURATOM/FIAT/ANSALDO CONTRACT

3.1 Purpose of the Tests and Experimental Apparatus

As indicated in Section 2 above, all Classification Societies require a special structure against collision in order to provide a proper margin of safety against damage to the reactor vessel.

The criteria used to define the type of structure are theoretically the same for all the Classification Societies mentioned.

However, reliable data are practically non-existent and there seems to be no basis for a method of calculating the scantlings of the structures providing collision protection.

As indicated above, the American Bureau of Shipping recommends the study of the consequences of actual collisions involving ships having characteristics similar to those of the ship being designed.

On the basis of collisions which occurred in the period 1953-56 and were recorded by the US Salvage Association and the Liverpool Underwriters' Association from US Coast Guard data, Gibbs & Cox Inc., New York, carried out a study in 1960 (revised 1961) published as "Report PB 173602.1", to determine in each particular case the collision energy absorbed by the deformed structures (separately for the striking bow and for the struck ship side).

A fairly satisfactory correlation was established between the volume of destroyed or deformed structures and the collision energy (see Gibbs & Cox Inc. Report PB 173602.1 - Criteria for guidance in the design of nuclear powered merchant ships - Section 4 - Collision barrier).

Previously (October 1959) an article had been published in the Journal of Ship Research by V.U. Minorsky (Naval Architect at George G. Sharp Inc., New York) proposing a theory based on similar criteria.

This theory attempts an empirical correlation between resistance to penetration (proportional to the summation of all the volumes of the various partial elements of the structure affected by collision damage, both for the striking and for the struck ship) and the energy absorbed by the collision.

Depending on the speed and the masses of the two ships, the collision energy is expressed by V.U. Minorsky as:

$$\left[\frac{\Delta_A\ \Delta_B}{1.43\Delta_B + 2\Delta_A}\right] (V_B \sin \theta)^2$$

which is an approximate value.

In the above expression Δ_A is the displacement of the collided ship plus 40% for entrained water, Δ_B is the displacement of the colliding ship, V_B is the speed in knots of the colliding ship and θ is the angle of impact. It can be deduced that, for a specified structure and a particular collision case, the extent of damage could be predicted, or a particular type of structure designed to limit the extent of the collision damage to a predetermined value.

It should be pointed out that, according to the Minorsky theory, the outer shell and similar plated structures perpendicular to the direction of impact are assumed to contribute very little, if at all, towards absorbing the collision energy. This assumption seems to be in contrast with the results of the experiments carried out in Japan (see Section 1.2.1), from which it appears that an increased side shell plating thickness is fairly efficient in reducing the damage suffered by the collided ship (at the same time, of course, the energy absorbed by the striking bow increases, as does its deformation).

In 1963 a decision was taken jointly by Euratom, CNEN and Ansaldo to perform some collision tests with models, employing the apparatus already in use at the Shipbuilding Institute of Naples University (see Section 1.2.2). These tests were planned for the following purpose:

(a) to examine the efficiency of various types of structures for various conditions of speed, displacement and bow type of the colliding ship;

(b) to compare the results of collisions involving the side of a normal merchant ship with those of a side structure specially designed to withstand collision;

(c) to compare the reliability of calculation theories on the absorption of collision energy with experimental results obtained from models;

(d) to ascertain whether the use of thicker shell plating is really an efficient way of reducing the penetration of the colliding bow, and if so, to what extent.

The experimental apparatus at the Shipbuilding Institute of Naples, which was built with the aid of CNEN assistance to Naples University, is shown in Fig. 2 and has already been described in Section 1.2.2.

The procedures adopted in order to reproduce the actual conditions as much as possible in the various tests (method of fixing the models, tin soldered joints changed to partially welded and then totally welded joints in the construction of the models, etc.) will be described further below, when the actual tests are described.

3.2 Similarity Theory and Symbols

All the collision experiments already described in Section 1.2 were based on the so-called "explosion theory" according to which, λ being the ratio between the ship and the model dimensions, the similarity relations are as follows:

for speeds	1
for forces	λ^2
for energies	λ^3
for time	λ
for accelerations	λ^{-1}
for masses	λ^3
for pressures	1

This similarity theory was also assumed for the collision tests described below.

The ratio $1/\lambda = 1/15$ was employed because it was the most suitable one for use with the available apparatus and had already been employed with models of the same scale.

In only one case (test No. 6) was a 1/10 scale adopted (see Section 1.1), but this was a special test which was not included in the series initially planned and was carried out for a purpose to be explained later.

The expressions employed to determine the virtual mass of water entrained by the ship side model and the energy absorbed by the collision are as follows:

$$f = m_1 \, a_1 = (m_2 + dm) \, a_2$$

whence:

$$m_2 + dm = m_2' = \frac{m_1 a_1}{a_2}$$

and then:

$$dm = \frac{m_1 a_1 - m_2 a_2}{a_2}$$

$$\Delta E = \frac{1}{2} m_1 V_1^2 - \frac{1}{2} (m_1 + m_2') V^2$$

In the above expressions:

f = the mutual force of impact (tonnes)

m_1 = mass of the colliding ship (tonnes)

a_1 = deceleration of the colliding ship (average of maximum values)(m/sec^2)

m_2 = mass of the struck ship (tonnes)

a_2 = acceleration of the struck ship (average of maximum values)(m/sec^2)

dm = virtual mass of water entrained by ship side model (tonnes)

m_2' = m_2 + dm (tonnes)

ΔE = energy absorbed during collision

V_1 = speed of the bow (m/sec)

V = combined speed of models after impact, recorded experimentally (m/sec)

In those cases where the combined speed of the models after impact can not be recorded owing to the ship side model being rigidly mounted, the combined speed V defined above is replaced by the following theoretical value:

$$U = m_1 V_1 / (m_1 + m_2')$$

The speed of the struck ship was always taken to be equal to zero, except for test No. 8, in which, as will be explained later, a low speed of the struck ship was taken into consideration (about 5 knots) by means of a particular relative position of the two models.

The values of a_1, a_2, V, V_1 and also the stresses in the ship side structures of the collided ship were experimentally recorded by appropriate apparatus.

This apparatus can be described as follows (see Section 1.2.2 relating to the above-mentioned paper by Prof. Spinelli - "Protection du compartiment du réacteur nucléaire contre les abordages. Résultats d'essais sur modèles"):

- accelerations a_1 and a_2 were recorded by gravity-type accelerometers, with a sensitive element formed by a mass having elastic suspension and transmitting its movements to a potentiometer;
- speeds were recorded by means of an electric signal every 50 cm covered by the carriage carrying the colliding bow model and every 25 cm covered by the struck model (when not rigidly mounted);
- stresses were recorded by Budd C6-131 strain gauges (120 $\Omega \pm 0.2$; gauge factor 2.5 $\pm 0.5\%$), photoengraved, autocompensated for temperature and placed on the side of a Wheatstone bridge.

Signals from the pick-up points were utilized by means of stabilized supply lines and amplification lines with Visigraph-type recorders, which detected the signals received on a photographic paper strip through mirror galvanometers. The stabilized lines were used for supplying and detecting signals from the potentiometric pick-up, while the amplification lines were employed for recording with the Wheatstone bridge.

Diagrams of the accelerations, speeds and stresses recorded in test No. 8 by means of the instruments described above are plotted in Fig. 5 as an example.

The speed of the carriage carrying the striking bow could obviously be changed by varying the height from which it was released.

However, it should be noted that the highest carriage speed possible with the apparatus described was about 6.1 m/sec, i.e., 11.86 knots.

In order to obviate this limitation, as will be explained in Section 4, the colliding mass was altered to give an energy, with a speed of 6 m/sec, equal to the energy obtainable with the design speed and the original colliding mass.

In tests No. 1-6 and 8 the carriage to which the ship side model was attached was not rigidly mounted.

The drift plan of the struck ship was represented on the model by two sets of three drift fins, one on each side, immersed in water tanks so as to give a ratio of λ^2 between the drift plan of the full-size ship and the total area of the fins.

Tests No. 7 and 9-14 were carried out with the ship side model fixed to a rigidly mounted frame to avoid transverse motion.

3.3 Material and Connections of Model Elements - Their Assembly

3.3.1 Physical tests on the steel materials for the models

The material used in the models was normal hull steel Fe-42 (previously named Aq-42), having a tensile strength of between 41 and 50 kg/mm^2, elongation on standard specimen $\geq$ 15%, sulphur limit S$\leq$0.05%, phosphorus limit P$\leq$0.05%.

Prior to starting the construction of the first couple of models, several strength tests were performed on specimens taken from galvanized, tinned and normal steel plate.

Specimens having a thickness of between 0.6 and 2.8 mm were tested and tensile strengths of between 42.5 and 52.2 kg/mm^2 recorded.

In one case only, that of a normal steel plate 2.2 mm thick, a tensile strength of 40.6 kg/mm^2 was recorded.

The elongation values observed were always much higher than the minimum required.

Good results were obtained from bending tests up to 180^{o}.

Newly acquired materials intended for the construction of models after the initial ones were always subjected to tests for control purposes in order to ensure that the material characteristics did not vary.

3.3.2 Joining of model elements

Joints for the assembly of all the structural elements constitute one of the most important problems when building models, since the low thickness of the plate renders electric welding difficult. Moreover, it must be noted that, in electric welding, it is also difficult to adhere to the geometrical proportion of the 1/15 ratio with respect to the dimensions of the deposited metal. This fact could alter the value of the section modulus in built-up welded elements.

In the case of the models for tests No. 1-9 only the joints of the outer plating of the bow and of the ship side and the joints of the main deck over the ship side were electrically welded; the thickness of plating used for these elements was about 1 mm.

Tensile strength and bending tests were performed on two specimens having the dimensions showed in Fig. 6, Cito 1.5 mm electrodes being used in one case and Valens 1.5 mm in the other.

When tested to determine their tensile strength, the specimens failed at 48.7 kg/mm^2 in the first case and at 50.8 kg/mm^2 in the second.

Photo 1 shows the result of a 180^{o} bending test on one specimen.

When models for the initial seven collision tests were constructed, tin soldering was used for the main and platform deck plating of the entire bow model, the bulkheads and all material less than 1 mm thick; the method consisted in soft soldering involving fusion of the metal alone. The same process was employed for joining the frames, side stringers, deck girders and bulkheads stiffeners to the associated plating, and also for connecting built-up structural elements. Care was taken to use a fairly large quantity of deposited metal in order to obtain the strongest possible joints, as was subsequently verified by testing.

The following types of solders were previously tested:

1) Sn 50% - Pb 50%
2) Sn 52% - Pb 48%
3) Sn 55% - Pb 45%
4) Sn 60% - Pb 40%

Solder No. 2 also contained a very small quantity of bismuth.

Solders containing higher percentages of tin permit quicker soldering work at a relatively low temperature.

In view of the low thickness plating employed, it is very important to avoid distortions due to high temperatures. Nevertheless it is not advisable to reduce the quantity of lead below a certain limit, since this could also reduce the efficiency of the alloy.

The tests performed led to the adoption of the Sn60/Pb40 solder, which was a satisfactory solution in view of the problems touched on above.

The normal thin plates were tinned before soldering, and the elements to be joined were previously cleaned with a hydrochloric acid solution.

Photo 2 shows the result of a 180° bending test on a tin soldered T-specimen (see also Fig. 7). This test proved satisfactory in that no dissociation of the tin coat from the steel was ascertained.

Photo 3 illustrates the result of a bending test performed by means of a press acting half-way along the specimen shown in Fig. 8.

At a 50° angle a crack started in the soldering of the web at the point of the load.

One bending test was also performed on a cruciform structure, shown in Fig. 9.

Photo 4 shows the result of this test: at a 30° angle a crack started in the flange of a cross bar, close to a soldered joint.

The tests showed that, if a large quantity of deposited metal is used, the soldered joints are sufficiently strong.

On the other hand, low tensile strength values were recorded when tin soldered butt joints were tested.

Some tests carried out at Naples University had given values of about 2 kg/mm^2, but tensile strengths close to that of the base (steel) material were obtained by using overlapped soldered joints having a width of overlap equal to 20-30 times the plate thickness.

For this type of joint "equivalent stresses" were calculated from the ratio breaking load from test/specimen cross section.

This overlapped joint system was employed for soldering the plating of the platform decks and bulkheads up to collision test No. 9 inclusive.

The models built for test No. 10 were entirely electrically welded, in order to avoid dissociation of tin soldered structural elements, since in some isolated instances joints had become detached in the initial tests as a result of the collision shock. Copper plate steel wire supplied by Arcos Co. was chosen, 0.6 mm in diameter. This type of welding, performed with a CO_2 shield, gave very satisfactory results, even in the case of small welds.

Models built for tests No. 11-14 were not only entirely electrically welded, but they were also heat-treated at 650°C to avoid residual tensile stress at the welds.

3.3.3 Workshop assembly of model elements

In order to avoid deformation of the plating during welding, two wooden frames were set up, on the first of which the plating of the bow model (see Photos 5 and 6) and on the second the plating of the ship side (see Photo 7) were screwed.

With the aid of these frames the models could be turned over during assembly and placed in the appropriate position to simplify the work.

In addition, when some minor modifications were adopted, they were then used in the construction of all the subsequent models.

To join elements to which access from inside the model was impracticable because of lack of space (e.g., connecting of floors to the double bottom plating), slot tin soldering of the type shown in Fig. 10 was used.

Photo 8 shows the satisfactory result of a bending test on the specimen in Fig. 10.

3.3.4 Mounting of models on carriage

The bow models were fixed in the workshop to a fastening frame which was then, at the point where the test was going to be performed, attached to the head plate of a carriage running on rails. This frame is shown in Fig. 11.

The frame and its fixing bolts were fully proportioned on the basis of an anticipated force of impact. In point of fact no serious damage to the frame structure or to the model-mounting components was noted throughout the tests.

As is shown in Fig. 11, in all the collision tests (except for No. 8) rubber distance pieces were fitted between the model fastening frame and the headpiece of the carriage. These were intended to absorb, according to the model scale, an elastic deformation corresponding to that absorbed by the whole length of the striking ship plus the deformation of the struck ship side as due to the collision impact. The deformation over the length of the striking ship is:

$$\Delta L = \frac{F \lambda^2 L}{AE} ,$$

where F is the force of impact in tonnes, λ the scale of the model, L the length of the colliding ship in metres, A the area of the longitudinal elements calculated at the mid-ship section in m^2 and E the modulus of elasticity of the hull material. The above two elastic deformations amount to about 5 cm in full size and 3.5 mm on the model scale.

In order to fix the ship side model to the head plate of its carriage, a fastening frame was designed, consisting of one end plate and two side plates, perpendicular to the first and suitable stiffened, as shown in Fig. 12.

When the first and second tests were performed, the ship side model was rigidly attached in the workshop to the side plates of the model-holding structure, but in the second test the model became detached from both side plates as a result of the collision shock. Consequently, when the ship side model was mounted on the frame prior to the third test, rubber pieces were interposed between the transverse end sections of the ship side model and the side plates of the holding structure to permit elastic rotation of the transverse end sections of the model, as happens in an actual collision owing to bending of the struck hull.

This improvement appeared to be very successful, so that it was adopted in all the subsequent tests.

Fig. 13 shows another improvement. It consists in the fitting of appropriate pins for measuring elastic deformations of the transverse end sections in the ship side model relative to the side plates of the holding structure.

4. DESCRIPTION OF TESTS PERFORMED

A summary of the results of all the tests performed is given in the general table at the end of this report.

4.1.1 Test No. 1

The bow model used for this test is shown in Fig. 14 and in Photo 9.

The bow of a 53,450 tdw tanker was considered to be equal to the bow of the nuclear-powered tanker examined in the intermediate Ansaldo design, normal (i.e., not strengthened) type, with a waterline entrance angle of 64^{o}. A full load displacement of 69,200 tonnes was assumed.

The ship side was built in accordance with the hull structure of the nuclear-powered tanker designed by Ansaldo, in the area of the reactor space, with a barrier against collision in the form of three side platform decks fitted between the side shell plating and a watertight longitudinal bulkhead at a distance of 10.02 m from the longitudinal plane of symmetry of the ship (about 5 m from the shell plating at the full summer load waterline). A second longitudinal bulkhead, bounding the reactor space, is located 7.515 m from the centre line (7.985 m from the shell plating, measured on the main deck).

This last longitudinal bulkhead is more or less entirely independent of the side anticollision structure, in accordance with the requirements imposed by most Classification Societies (see Section 2).

The ship side structure described above is shown in Figs. 15 and 16 and in Photo 10.

The struck ship was assumed to be fully loaded, i.e., having 69,200 tonnes displacement, equal to that of the colliding ship.

The recorded collision speed was 4.68 m/sec, i.e., 9.1 knots. The recorded combined speed of the models after impact was 1.87 m/sec.

It is known that (see Section 3.2):

$$(m_1 + m_2 + dm)\ V = m_1\ V_1 + m_2\ V_2$$

whence:

$$dm = \frac{m_1\ V_1 + m_2\ V_2}{V} - (m_1 + m_2),$$

where

m_1 = mass of the striking ship's model =

$$\text{ship displacement} \times \frac{1}{\lambda^3 \times 9.81} = 2.1 \text{ tonnes}$$

m_2 = mass of the struck ship's model (excluding the mass of entrained water) = 2.1 tonnes

V_1 = speed of the striking ship = 4.68 m/sec

V_2 = speed of the struck ship = 0

V = combined speed of ships after impact = 1.87 m/sec

The mass of the water entrained by the collided ship can be calculated as follows:

$$dm = \frac{2.1 \times 4.68}{1.87} - (2.1 + 2.1) = 1.05$$

This value is 50% of the value of m_2, so that the virtual collided mass is:

$m_2' = m_2 + dm = 3.15$ tonnes.

The acceleration values and all the data relating to this test are given in the general summary table.

Fig. 17 and Photos 11 and 12 show the conditions of the bow and of the ship side after collision.

It appears very clearly that practically all the deformation work was absorbed by the bow (within a distance of 10 m measured from the fore end), while on the ship side only slight buckling could be seen.

From these results it was inferred that the anticollision structure was relatively strong compared with the bow structure, which proved to be weak and lacking in characteristics required for the scheduled tests.

4.1.2 Test No. 2

On the basis of the results reported above for test No. 1, it was decided to provide for the second collision test a stronger bow model relating to a 53,450 tdw tanker, as for test No. 1, but with an ice-strengthened bow having a smaller waterline entrance angle (38° instead of 64°), and the collision speed was raised to about 12 knots (it was actually 6.1 m/sec i.e., 11.85 knots). The new bow model is shown in Fig. 18. The ship side model was identical to that used for test No. 1.

Both the colliding and the struck ship were assumed to be in full load condition as in test No. 1.

In test No. 2 the mass of entrained water (dm) was 43% of the mass of the struck ship, i.e., somewhat less than in test No. 1.

As is shown in Fig. 19 and also in Photos 13 and 14, the bow damage was less extensive than in test No. 1, while heavier ship side damage occurred.

After this test the ship side model became detached from the side plates of the holding frame, as has already been stated in Section 3.3.4.

Bow damage extended to 3 m from the fore end, i.e., less than one-third of the length found after test No. 1, while the ship side structure was damaged up to 7.80 m (for the full-size ship), but without fracturing of the outer shell plating.

4.1.3 Tests No. 3 and 4

In tests No. 3 and 4 the ship side structure of a normal-type, fully loaded 53,450 tdw tanker (test No. 3) and the previously tested ship side structure specially designed against collision and pertaining to a similar laden nuclear-powered tanker (test No. 4) were compared when struck by the same bow under the same impact conditions. The displacement of both ships was 69,200 tonnes.

The ship side structure used in test No. 3 is shown in Fig. 20.

The striking bow was identical in both cases and the same as the one used in test No. 2, namely an ice-strengthened bow for a fully laden 53,450 tdw tanker, with a waterline entrance angle of 38°. The collision speed was set at about 10 knots; the actual recorded speed was 5.08 m/sec (9.9 knots) in both cases.

As an improvement in this test, the ship side model was attached to the side plates of the holding structure by means of interposed rubber pieces (see Section 3.3.4). As has already been stated, this system was adopted in all the following tests.

The mass of the entrained water was calculated in test No. 3 to be 45% and in test No. 4 31.5% of the mass of the struck ship.

The length of the bow affected by collision damage was about the same in both cases (5.70 and 5.50 m respectively from the fore end), but the penetration of damage into the ship side was 9.00 and 7.20 m in tests No. 3 and 4 respectively.

Furthermore, while a hole 8.10 m high 1 m wide (full-size ship) was made in the outer shell plating in test No. 3, only a deformation of the outer shell plating was observed in test No. 4.

These tests clearly established the efficiency of the anticollision barrier (see Figs. 21 and 22 and Photos 15-18).

4.1.4 Test No. 5

For test No. 5, the bow and side models chosen were identical to those used in test No. 4, but, while the colliding ship was again assumed to be in full-load condition, the struck ship was considered to be in ballast (displacement 50,000 tonnes instead of 69,200).

The recorded collision speed was 4.5 m/sec (8.76 knots). The entrained water was 50% of the mass of the struck ship.

The collision energy was entirely absorbed by plastic deformation of the ship side structure to a depth of 6.60 m and by fractures which occurred in the outer shell and in the main deck on both sides of the point where the ship side had been damaged as a result of the collision. These fractures reached a limit of about 3.20 m transversely on the deck and of about 7.50 m vertically on the outer shell (full-size ship).

The bow was almost undamaged.
For this test (No. 5) see Fig. 23 and Photo 19.

4.1.5 Test No. 6

The objectives of this test were different from those of the preceding and following tests. The aim was to verify whether, after reproducing by means of models all the conditions of an actual collision, the results of the actual collision and of the model collision could be compared.

The characteristics of the ships that had collided were as follows:

a) Colliding ship

Length 140 m
Breadth 18.90 m
Depth 10.46 m
Displacement (in ballast) 11,000 tonnes
Draught at the above displacement 5.60 m
Speed at time of collision 10.7 knots
Waterline angle of entrance 48°

b) Struck ship (12,960 tdw tanker)

Length 134 m
Breadth 19.50 m
Depth 10.62 m
Displacement (in ballast) 8,500 tonnes
Draught at the above displacement 4.62 m
Speed at time of collision about 0

Models for this test were built on a 1/10 scale on the basis of original drawings of the two ships.

The angle of impact was fixed at 76° on the basis of data given in a report on the collision.

The relative position of the two models as arranged for the collision test and the damage caused by the collision shock are shown in Figs. 24 and 25.

While the damage caused to the bow model could be readily compared with that to the colliding ship, and was very similar (see Photos 20 and 21), the struck model suffered much less damage than the ship side of the struck ship (see Photos 22 and 23).

This discrepancy between the damage undergone by the struck ship and that observed in the ship side model can be accounted for by several factors, the most important ones being:

(a) The speed of the colliding ship might have been higher than that reported by the Captain.

This hypothesis, which could be justified in some cases of collision, would explain the small amount of damage to the ship side model, but cannot account for the fact that the bow damage is at the same time very similar to the actual damage to the colliding ship.

(b) The speed of the struck ship might have been higher than that reported by the Captain.

For obvious reasons, this hypothesis might be more justified than the previous one.

The fact that the struck ship may have been under way at the moment of impact certainly could explain the more extensive fracturing of the side plating.

It should be noted that the structure of the ship side model was damaged to a depth of 4 m (full size), but only a small fracture to the side plating occurred, near the bottom of the model and equivalent to a full-size opening 0.30 m wide and 2.10 m high.

In the actual collision the depth of penetration of the bow was about 4 m, which is equal to the depth of deformation recorded on the ship side model; the larger dimensions of the hole in the side plating of the actual ship might be due to the speed of the struck ship, which was more than zero.

(c) The similarity theory between the model and the full-size ship is not completely valid.

The tests carried out by several researchers (see Section 2) were performed with model scales varying from 1/20 to 1/7.5; the results obtained from struck ship side models are satisfactorily equivalent.

However, this lack of validity of the similarity theory would only apply to the ship side structure, since the damage suffered by bow models is generally equal to that caused to the bows of ships that are actually involved in collisions, as can be seen in the photos frequently published in the technical press.

4.1.6 Test No. 7

The models chosen for this test were identical to those used in test No. 5. The displacements were also identical, i.e., 69,200 tonnes for the fully laden colliding ship and 50,000 tonnes for the struck ship, which was in ballast. The same collision speed was also adopted (4.5 m/sec or 8.76 knots).

It should be pointed out that the carriage holding the ship side model in this test was rigidly mounted to avoid transverse motion after the collision, while in previous tests this motion had been permitted (see Section 3.2).

In this case V was obviously zero. Before testing, it was therefore necessary to consider the mass of water entrained by the struck ship (50%, as had been calculated after test No. 5) and, consequently, to modify the mass of the colliding model as follows:

The mass of the colliding ship is $\frac{69{,}200}{9.81} = 7{,}050$ tonnes

corresponding to a model mass:

$$m_1 = \frac{7{,}050}{\lambda^3} = \frac{7{,}050}{15^3} = 2.1 \text{ tonnes}$$

Also

$$m_2 = \frac{50,000}{9.81 \times 15^3} = 1.51 \text{ tonnes}$$

The following formula must then be considered:

$$\frac{1}{2}\, m\, V_1^2 = \frac{1}{2}\, m_1\, V_1^2\, \frac{m_2'}{m_1 + m_2'}$$

where m is the modified mass of the colliding model and the other symbols have the notation given in Section 3.2.

In this case:

$$m_2' = m_2 + dm = m_2 + 0.50\, m_2 = 2.26 \text{ tonnes}$$

From the above expression m can be calculated:

$$m = \frac{m_1\, m_2'}{m_1 + m_2'} = \frac{2.1 \times 2.26}{2.1 + 2.26} = 1.09 \text{ tonnes (mass)}$$

The total weight of the carriage carrying the bow was consequently defined as:

$$1.09 \times 9.81 = 10.7 \text{ tonnes (weight)}$$

As regards the damages suffered by the models, the results of test No. 7 were thus practically identical to those observed after test No. 5. The colliding bow showed only heavy buckling, while plastic deformations and fractures in the ship side structures absorbed almost all of the collision energy, reaching a depth of 6.60 m (full size). A fracture 9.60 m high in the side plating was noted, affecting the main deck transversely for a depth of 3.90 m (full size).

The results of this test are shown in Fig. 26 and Photos 24 and 25.

Comparison of the results of tests No. 5 and 7 revealed that the carriage of the ship side model could always be rigidly mounted, provided that the total weight of the carriage and the bow model on it was suitably altered.

4.1.7 Test No. 8

On this occasion, and for the last time, the carriage carrying the ship side model was free to move laterally after the collision.

The bow and ship side models were identical to those used in test No. 3, i.e.,

- Ice-strengthened bow of a fully loaded 53,450 tdw tanker (displacement 69,200 tonnes);
- Ship side in way of cargo tanks of a fully loaded 53,450 tdw tanker, normally structured, i.e., without anticollision barrier (displacement 69,200 tonnes).

The following variations were adopted with regard to test No. 3:

- Collision speed 5.5 m/sec (10.7 knots) instead of 5.08 m/sec (9.9 knots);
- Bow model fixed rigidly to the frame, i.e., without interposed rubber pieces.

For this test, a speed of the struck ship different from zero was simulated by arranging the position of the models and the angle of impact appropriately. With the models still positioned at 90° to each other, but rotated through an angle of 24° 30' about a vertical axis (see Fig. 27), a recorded carriage speed of 10.7 knots (5.5 m/sec) corresponded to a speed of 9.7 knots for the striking bow and 4.4 knots for the struck ship.

The length of the bow damaged by the collision was about 6 m from the fore end (full size), compared with 5.70 m in test No. 3. The forward part of the bow, however, showed a different type of deformation, because the structures were badly crushed and bent towards the port side, as was to be expected (see Figs. 27 and 28 and also Photos 26 and 27).

The ship side was damaged to a depth of 4.60 m (full size), compared with 9 m in test No. 3. No fractures occurred in the side shell plating.

It is interesting to note the small quantity of entrained water recorded in this test, namely, 19% of the mass of the struck ship; this low figure is comparable with the 21% recorded in test No. 6, where the models collided at an angle of 76°.

4.1.8 Test No. 9

The models were identical to those employed in test No. 4 i.e.,

- Ice-strengthened bow of a fully loaded 53,450 tdw tanker (displacement 69,200 tonnes);
- Ship side of a fully loaded nuclear –powered tanker (displacement 69,200 tonnes)

The following variations were adopted in test No. 4:

- Collision speed 8.22 m/sec (16 knots) instead of 5.08 m/sec (9.9 knots);
- Ship side carried on a rigidly mounted carriage.

The mass of water entrained by the struck ship was considered to be 45% of its own mass.

The absorbed energy can be calculated as follows:

$\Delta E = \frac{1}{2} m_1 V_1^2 - \frac{1}{2} (m_1 + m_2') U^2$ (for symbols see Section 3.2);

where:

$m_1 = 2.1$ tonnes

$m_2' = m_2 + dm = 2.1 \times 1.45 = 3.05$ tonnes

$U = m_1 V_1 / m_1 + m_2' = 3.35$ m/sec

In the case of test No. 9, the value of ΔE for the model is:

$$\Delta E = \frac{1}{2} \times 2.1 \times 8.22^2 - \frac{1}{2} (2.1 + 3.05)\ 3.35^2 = 42.1 \text{ tonnes x m.}$$

Since a carriage speed of more than 6.1 m/sec (11.86 knots) could not be attained with the testing equipment available, it was decided to limit the speed to 6 m/sec; the struck model being rigidly mounted (V = 0), the value of m (modified striking mass) was calculated as follows:

$$\Delta E = 42.1 = \frac{1}{2} m \times 6^2 \qquad \text{whence:} \quad m = \frac{2 \times \Delta E}{6^2} = \frac{2 \times 42.1}{36} = 2.34 \text{ tonnes}$$

Consequently the total weight of the carriage and the bow model was:

$2.34 \times 9.81 = 23$ tonnes

The results of this test, shown in Fig. 29 and in Photos 28 and 29, are compared here with those of test No. 4:

		Test No. 4	Test No. 9
Speed of colliding ship (knots)		9.9	16
Length of the bow affected by collision damage from stem (full size)	(m)	5.50	9.00
Depth of damage to the ship side (full size)	(m)	7.20	8.25

Furthermore, whereas in test No. 4 there were no fractures of the side shell plating, in test No. 9 a hole was caused in the plating as high as the ship's depth and 1.50 m wide (full size). A small fracture in the intermediate longitudinal bulkhead and a very small split in the longitudinal bulkhead bounding the reactor space were also found.

4.1.9 Test No. 10

A model of the bow of a passenger liner of the "Bremen" type was used. The draught was 7.92 m, corresponding to a displacement of 17,050 tonnes; the collision speed was 16 knots (8.22 m/sec).

This type of bow, having a waterline angle of entrance of about 20°, was identical to the one chosen by the Deutsche Werft for the test carried out at Hamburg (see Section 1.2.3).

The side structure of the struck ship was that of the fully loaded nuclear-powered tanker designed by Ansaldo (displacement 69,200 tonnes).

The carriage carrying the ship side model was mounted rigidly, this system always being employed in the subsequent tests.

For the reasons described in Section 4.1.8, the mass of the striking bow had to be modified in view of the fact that the maximum carriage speed was 6 m/sec.

From the formulas:

$$\Delta E = \frac{1}{2} m_1 V_1^2 - \frac{1}{2} (m_1 + m_2') U^2 \quad \text{and} \quad m_2' = m_2 + dm$$

assuming $dm = 0.45 \times m_2$

and since $m_1 = 0.515$ tonnes and $m_2 = 2.1$ tonnes, the following values of m_2', U and ΔE can be calculated:

$$m_2' = 1.45 \times 2.1 = 3.05 \text{ tonnes}$$

$$U = m_1 V_1 / (m_1 + m_2') = 1.19 \text{ m/sec}$$

$$\Delta E = \frac{1}{2} \times 0.515 \times 8.22^2 - \frac{1}{2} (0.515 + 3.05)\, 1.19^2 = 14.9 \text{ tonnes} \times \text{m}.$$

As in test No. 9 (Section 4.1.8), the modified value of the colliding ship's mass was:

$$m = \frac{2\,\Delta E}{6^2} = 0.826 \text{ tonnes,}$$

and the total weight of the carriage and of the bow model was:

$$0.826 \times 9.81 = 8.10 \text{ tonnes.}$$

The results of test No. 10 are shown in Figs. 30 and 31 and also in Photos 30 and 31.

The damaged part of the bow extended for 9.40 m from the stem; the deformation into the ship side reached a depth of 3.90 m, and a hole was made in the shell plating 9 m high and 1.50 m wide (all full-size dimensions).

4.1.10 Test No. 11

Test No. 11 was performed with the following models:

- Ice-strengthened bow of a fully loaded tanker (displacement 69,200 tonnes) of the type previously described.
- Ship side of a fully loaded 53,450 tdw tanker (displacement 69,200 tonnes) in way of the reactor space but without the anticollision protection formed by side platform decks and an added longitudinal bulkhead, i.e., without the special barrier designed by Ansaldo. To replace this protection the thickness of the whole shell plating was increased to 45 mm. The original thicknesses were as follows: shearstrake 42 mm, side plating 26 mm, bottom plating up to the height of the double bottom 40 mm.

All the other parameters (masses, absorbed energy ΔE, speeds, etc.) were identical to those already reported for test No. 9, which was carried out with the same colliding bow against the side of the nuclear-powered ship fitted with anticollision barrier. In particular, the collision speed in both cases was 8.22 m/sec (16 knots).

The data for tests No. 9 and 11 are compared below:

		Test No. 9	Test No. 11
Length of the damaged part of the bow (from forward end)	(m)	9.00	5.65
Depth of damage in the ship side	(m)	8.25	5.70

As was stated in Section 4.1.8, the following damage to the shell plating was observed after test No. 9: an opening in the side shell plating as high as the ship's depth and 1.50 m wide (full-size dimensions), a small fracture in the intermediate longitudinal bulkhead and also a small split in the inner longitudinal bulkhead bounding the reactor space. After test

No. 11, no fracture of the outer shell was observed; on the hull structure only deformations were noted and the inner longitudinal bulkhead was only slightly buckled (see Figs. 32 and 33 and Photos 32 and 33).

Comparison of collision tests No. 9 and 11 (see Figs. 29 and 33) seems to confirm the results obtained by the Yokohama Shipyard in Japan (Section 1.2.1) and the views on the same subject outlined in Section 3.1 of this report.

The damage to the colliding bow extended further in the case of test No. 9, but after test No. 11 the bow model appeared to be completely flattened by the collision impact.

In test No. 11 the increased thickness of the shell plating evidently caused the bow to be flattened over its whole height so that it could not pierce the side shell. On the contrary, in test No. 9 the fracture of the lower side plating, which was of normal thickness, allowed the strongest part of the bow to penetrate the plating during the first stage of the collision, thus causing wider and deeper damage to the side structure.

It should be pointed out that in test No. 11, contrary to what happened in all the other collision tests, the bow-mounting carriage was seen to recoil for a distance of about 1.30 m.

The effectiveness of increasing the thickness of the outer side plating was also demonstrated in test No. 14 (see Section 4.1.12).

4.1.11 Tests No. 12 and 13

Before these tests were performed, facilities were provided for increasing the length of the ship side model to 2.00 m (30 m full size) by removing the water tanks located at both sides of the carriage for the model. This obviously endows the struck model with greater elasticity.

The bow type used for tests No. 12 and 13 (and subsequently for test No. 14) was that for a 130,000 tdw tanker, with a large bulb, as shown in Fig. 34. Ballast conditions were selected for both these tests, with a draught of 7.92 m, corresponding to a displacement of 74,900 tonnes.

In both tests examined here, the ship side model was based on the nuclear-powered tanker designed by Ansaldo, which was fitted with an anticollision barrier and had a side plating of normal thickness. The draught was 12.00 m, corresponding to a displacement of 69,200 tonnes (fully laden).

The collision speed was 6.168 m/sec (12 knots) for test No 12 and 4,883 m/sec (9.5 knots) for test No. 13.

For test No. 12 the mass of the bow model was lowered from 2.26 tonnes to 1.37 tonnes (on the basis of an actual carriage speed of 6 m/sec) as follows:

$$U = \frac{m_1 V_1}{m_1 + m_2'} = \frac{2.26 \times 6.168}{2.26 + 3.05} = 2.63 \text{ m/sec},$$

$$\Delta E = \frac{1}{2} m_1 V_1^2 - \frac{1}{2} (m_1 + m_2') U^2 =$$

$$= \frac{1}{2} \times 2.26 \times 6.168^2 - \frac{1}{2} (2.26 + 3.05)\ 2.63^2 = 24.7 \text{ tonnes x m.}$$

The modified mass of the colliding model was:

$$m = \frac{2 \times \Delta E}{6^2} = \frac{2 \times 24.7}{36} = 1.37 \text{ tonnes.}$$

The total weight of the carriage and the bow model was therefore:

1.37 x 9.81 = 13.44 tonnes

For test No. 13 a carriage speed of 5 m/sec was chosen, instead of 6 m/sec as in test No. 12.

In this case:

$$U = \frac{2.26 \times 4.883}{2.26 \times 3.05} = 2.08 \text{ m/sec}$$

$$\Delta E = \frac{1}{2} \times 2.26 \times 4.883^2 - \frac{1}{2} (2.26 + 3.05)\ 2.08^2 = 15.47 \text{ tonnes x m.}$$

The modified mass of the colliding model was:

$$m = \frac{2 \times \Delta E}{5^2} = \frac{2 \times 15.47}{25} = 1.24 \text{ tonnes.}$$

The total weight of the carriage + bow model was then:

$$1.24 \times 9.81 = 12.1 \text{ tonnes.}$$

The results of tests No. 12 and 13 are as follows:

		Test No. 12	Test No. 13
Collision speed	(knots)	12	9.5
Length of bow affected by collision damage (from forward end of the bulb - full size)	(m)	8.70	7.35
Depth of damage of the ship side (full size)	(m)	9.00	8.10

After test No. 12 a hole 13.80 m high and 5.70 m wide (full-size ship) was observed in the outer shell plating. The width of the damages area was measured between the edges of the fully broken plates, while the deformation of the outer shell affected about 11 m of the ship's length.

There was also a minor fracture in the intermediate longitudinal bulkhead and a small split in the longitudinal bulkhead bounding the reactor space.

After test No. 13 a hole 13.80 m high and 5.50 m wide was observed in the outer shell plating (full-size dimensions), the width being measured between the edges of the broken plates. There was also a smaller fracture in the intermediate longitudinal bulkhead and a slight deformation in the longitudinal bulkhead bounding the reactor space.

The deformation of the outer shell extended for about 10 m of the ship's length (full size).

4.1.12 Test No. 14

Test No. 14 can be regarded as a repeat of test No. 12, with one variation: the thickness of the struck ship's shell plating was uniformly increased to 45 mm, as had already been done for test No. 11. This decision was taken in order to examine once more the collision-resistance of a thicker shell plating.

All the other conditions considered in tests No. 12 and 14 were identical (collision speed, displacements of ships, model mountings, etc.).

All the calculated data in Section 4.1.11 relating to test No. 12 are thus valid for test No. 14.

The effectiveness of the increased thickness of the shell plating was demonstrated very clearly from this test as can be seen by comparing Fig. 36 with Fig. 38 and also Photos 34 and 35 with Photos 38 and 39.

Furthermore, the following results can be compared:

		Test No. 12	Test No. 14
Collision speed	(knots)	12	12
Colliding ship's displacement	(tonnes)	74,900	74,900
Struck ship's displacement	(tonnes)	69,200	69,200
Length of damage on the bow (from forward end of bulb)	(m)	8.70	6.65
Depth of damage in ship side	(m)	9.00	4.55
Longitudinal extent of damage in ship side	(m)	11.00	15.10

In test No. 14, as was also noticed after test No. 11, the flattening of the bow due to the impact against the stout shell plating considerably decreased the bulb's ability to penetrate the ship side structure, so that the damage suffered by the ship side affected a wider area but a smaller depth, compared with the damage caused in test No. 12.

The bow model showed a greater damaged length in test No. 12 than in No. 14. This result also seems to be easily accounted for by the flattening of the forward end of the bow structure in test No. 14, which limited both the penetration of the bow into the ship side, as stated above, and, consequently, the cutting of the bow structure caused by the deck stringer (4.20 m in test No. 12 and 1.70 m in No. 14 - See Figs. 36 and 38).

Finally, the absorbed deformation work, which was the same in both cases compared here (owing to identical masses and speeds) affected a deeper but narrower zone in test No. 12, while in No. 14 a shallower but wider zone was affected, both on the bow and on the ship side models.

This result is a very interesting one for technicians working on the design of an efficient anticollision side protection.

5. SOME FURTHER CONSIDERATIONS ON THE RESULTS OF THE COLLISION TESTS

It might be said that the results of test No. 14 represent a large proportion of the conclusions which were expected from the whole series of tests, revealing as they do the following points:

1) The efficiency of an anticollision barrier is substantially increased if, in addition to the side decks and side stringers, a thicker side plating is employed, so that the colliding bow is crushed and flattened before it has a chance to pierce the side plating.

2) Bulbous bows have a greater tendency to penetrate the struck ship side unless a stronger side plating flattens the bow at first impact, as suggested above. The greater tendency towards penetration is essentially due to the longitudinal structural elements in the bulb.

The pronounced anticollision effectiveness of a thicker side plating was also demonstrated in test No. 11 (side structure of a normal tanker, with thicker side shell) compared with test No. 9 (anticollision structure formed by platform decks, but with side shell of Rule thickness). The results appeared to be more favourable in the case of test No. 11, the other test conditions being equal (speed, displacement, etc.).

Another interesting result of the test lies in the fact that, in a number of cases, the damage to the longitudinal bulkhead bounding the reactor space would have been much worse if the bulkhead had been structurally connected to the anticollision structure of the sides. This consideration fully proves the validity of the Classification Societies' requirement to the effect that the longitudinal bulkhead must be made structurally independent of the side structure.

However, it is perfectly obvious that the effectiveness of an anti-collision barrier can only be demonstrated within well-defined speed and displacement limits.

No anticollision barrier in a nuclear ship of average dimensions, no matter how well designed, could safely withstand a collision with a ship of several hundred thousand tons, even in the case of a relatively low collision speed.

Nevertheless it may be safely stated that, in the vast majority of cases of collision which may be anticipated today, a well-designed anticollision barrier may keep the damage to the struck ship within tolerable limits.

6. TENTATIVE CORRELATION OF COLLISION RESULTS

6.1 Actual Collisions

In general, the correlation of collision data has been attempted in the past by relating the volume of material affected by the collision to the energy of the two ships at the time of impact. While the second term (energy) can be fairly well defined, both in the case of actual collisions at sea and, even better, in the case of model tests, the first term (volume of material affected by the collision) is very difficult to evaluate, mainly because it is dependent upon the judgment of the individuals evaluating the collision damage. Since the collision does not actually "destroy" material, but simply fractures, crushes, bends or otherwise displaces the structures, each structural component absorbs a different amount of collision energy, depending upon the type of external action to which it has been subjected.

Even though only part of the damaged structure actually absorbs collision energy (namely, the part which has been bent, fractured or crushed), while other parts of the same structure are simply displaced in space, it nonetheless seems reasonable as a first approximation to evaluate the entire volume of material affected by the collision, on the assumption that there is a more or less constant ratio between the parts which have actually absorbed energy and those which have merely been displaced from their original position. This approach is also dictated by the practical impossibility of evaluating the ratio of the two parts to any significant degree of accuracy.

The following considerations are based on the total volume of material, namely, both the volume affected in the bow of the colliding ship and the volume affected in the side of the struck ship are considered together.

Fig. 4-C-20 in the reference mentioned in Section 3.1 (Gibbs & Cox Inc. Report PB 173602.1) shows a relation between the displaced volume of steel in both ships and the total energy absorbed in the collision. The relation can be described by a straight line equation:

$$E = 1{,}213\ V - 61{,}300$$

where:

E = energy (ft x tons)
V = volume of material (ft^3)

The data on which this relation is based were obtained by analysing a a number of actual collisions at sea. In metric units, the above equations becomes:

$$E = 0.01322 \times V - 18{,}900$$

where:

E = energy (tonnes x m)
V = volume of material (cm^3)

The same equation applied to 1:15 models gives:

$$e = 0.01322 \times v - 5.6$$

where:

e = energy (tonnes x m)

v = volume of material (cm^3)

6.2 Collision Tests with Ship Models

The same approach (consideration of the entire volume of material affected by the collision) was used as a first means of analysing the results of the collision tests on models, on the basis of tests No. 8-14 (see line A in Fig. 39). The line relating to Gibbs & Cox Inc. Report PB 173602.1 reduced to metric units and to the size of the models is also shown in this figure. It should be noted that the volume of material shown by this line presumably includes plates only, and disregards frames, longitudinals, webs, etc.

When the procedure suggested by Minorsky in the article quoted in Section 3.1 was applied to the models, a considerable scatter of the points occurred (see Fig. 40); also, a very low volume of material seems to be effective in absorbing the collision energy. It was thought that the Minorsky procedure was unnecessarily crude for our purposes and it was thus abandoned.

Line B in Fig. 41 shows a second attempt at correlation: in this case, the volume of plating and stiffeners normal to the line of impact was ignored entirely. The test points show a somewhat greater scatter than in the previous case.

A third attempt at correlation is represented by line C in Fig. 42; in this case, the volume of material V represents:

(a) 100% of the material which underwent crushing, fracturing, etc. In other words, this is the material which was subjected to loads in excess of its ultimate tensile strength.

(b) 65% of the material which underwent only minor damage, material which was bent without major fractures or without being crushed, etc. This represents, in a broad sense, the material which reacted elastically, or which reached a load somewhat less than its ultimate tensile strength.

The percentage of 65% was selected arbitrarily, even though it is supposed to represent broadly the ratio of the elastic limit (yield point) to the ultimate tensile strength of the material.

It may be seen that this last procedure perhaps involves the least scatter of the test points.

Fig. 43 shows a comparative plot of the three lines A , B and C

6.3 Distribution of Deformation Work between the Bow of the Colliding Ship and the Side of the Struck Ship

Lines A , B and C in Fig. 43 represent the relation between the total collision energy absorbed and the volume of material affected by the collision, under the conditions illustrated in the preceding section.

It is also interesting to note the percentages of total energy which have been absorbed by the bow and the ship side respectively. These were calculated from the volume of material affected by the collision in the bow and the side respectively, as a percentage of the total volume.

The results of these computations (tests No. 8-14) are summarized in the table overleaf.

Test No.	Collision Speed (Knots)	(A)		(B)		(C)		Notes
		B	S	B	S	B	S	
8	9.7	24.7	75.3	41.4	58.6	25.4	74.6	Struck ship speed 4.4 knots Carriage speed 10.7 knots
9	16.0	23.1	76.9	39.2	60.8	23.2	76.8	
10	16.0	27.9	72.1	50.2	49.8	28.3	71.7	
11	16.0	13.2	86.8	27.9	72.1	13.3	86.7	No special protection but thicker side plating
12	12.0	7.7	92.3	15.2	84.8	6.8	93.2	
13	9.5	13.3	86.7	21.3	78.7	12.3	87.7	
14	12.0	37.3	62.7	70.3	29.7	37.5	62.5	Same condition as for test No. 12, but thicker side plating

Columns A , B and C refer to the three basic assumptions made in evaluating the volume of material affected by the collision (see Section 6.2), namely:

A : 100% of the volume of material affected is included;

B : plates and stiffeners normal to the line of impact have been disregarded;

C : 100% of crushed and/or fractured structures, plus 65% of deformed structures.

Columns B and S show the percentage of energy absorbed by the bow and the side respectively.

It may be noted from above table that the percentages relating to conditions A and C are in close agreement, while those for condition B are quite different.

In particular, under condition B it may be seen that the energy absorbed by the side structure varies from 84.8% in test No. 12 to 29.7% in No. 14. Since the only difference between these two models was the increased side shell thickness in model No. 14 as compared with model No. 12, it seems obvious that this difference must be responsible for the large variation in energy apparently absorbed by the side in the two tests. It follows that the basic assumption underlying condition B (plates and stiffeners normal to the line of impact disregarded) is not truly representative of reality, since test No. 14 has clearly proved that the side shell plating has a very effective role in resisting the bow penetration. Comparison of the percentages in columns A and C for tests No. 12 and 14 shows more consistent results.

It should also be noted that test No. 14 shows the minimum percentage of energy absorbed by the side and the maximum percentage of energy absorbed by the bow, by comparison with all the other test results; this fact seems to prove the high degree of effectiveness of the anticollision structure used in this case.

The damage suffered by the side in test No. 11 was wider but shallower than that caused to the side in test No. 9; both tests were performed at the same collision speed and mass, and therefore energy. This fact is almost certainly accounted for by the increased side shell thickness used in test No. 11. However, in spite of the relatively shallow penetration, the percentage of energy absorbed by the side is relatively high; it follows that an increased side shell thickness alone is only partially effective in limiting collision damage.

The high percentage of collision energy absorbed by the side in tests No. 12 and 13 illustrates the potential danger represented by bulbous bows. While the values for the side in tests No. 8, 9 and 10 (normal bows) vary between 72 and 77%, in tests No. 12 and 13 (bulbous bows) they are in excess of 92%. In the case of test No. 14, as was stated above, the damage caused by the bulbous bow was reduced owing to the increased side shell thickness (from 26 to 45 mm, ship size).

7. CONSIDERATIONS ON THE STRUCTURAL STRENGTH OF THE COLLIDING BOW

The results of the collision tests, particularly those relating to tests No. 12, 13 and 14, where a bulbous bow was used (supposedly belonging to a large tanker), show that the flattening and crushing of the colliding bow is accompanied by a considerable reduction in the depth of the damage to the side structure.

Even though this report is concerned with the design of an effective anti-collision structure for the sides of a nuclear vessel, the possibility of taking suitable steps to limit the potential danger represented by the bows of larger vessels (particularly bulbous bows) should not be neglected. These steps should be aimed at designing bow structures capable of absorbing a relatively large deformation (flattening) in the event of collision, without, of course, jeopardizing the strength characteristics required for normal operation at sea.

This concept has already been debated in the technical literature. To emphasize the importance of the subject, it may be noted that, if the bow is encouraged to deform transversely in the event of collision, the damage to the bow in the longitudinal direction is normally reduced, as has been shown by the results of the collision tests.

Economic considerations relating to the cost of the repair of collision damage also militate in favour of the suggestion put forward above. It may in addition be pointed out that, while the damage suffered by the colliding ship (with traditional bow structures) are such that the buoyancy of the vessel is seldom endangered, the damage to the struck ship may cause the vessel to sink or, in the case of a nuclear ship, may lead to extremely dangerous situations if the damage affects the reactor containment vessel.

In conclusion, it would be desirable if the international authorities concerned could consider the possibility of requesting that bow structures be made less rigid longitudinally than is the practice at present.

This requirement would tend to reduce the potential danger inherent in a collision at sea, with advantages both for the struck ship and for the colliding ship; these advantages would, of course, be all the more valuable when the struck ship is nuclear-powered.

15 December 1970

FIGURES

COLLISION EXPERIMENTAL APPARATUS IN THE YOKOHAMA SHIPYARD

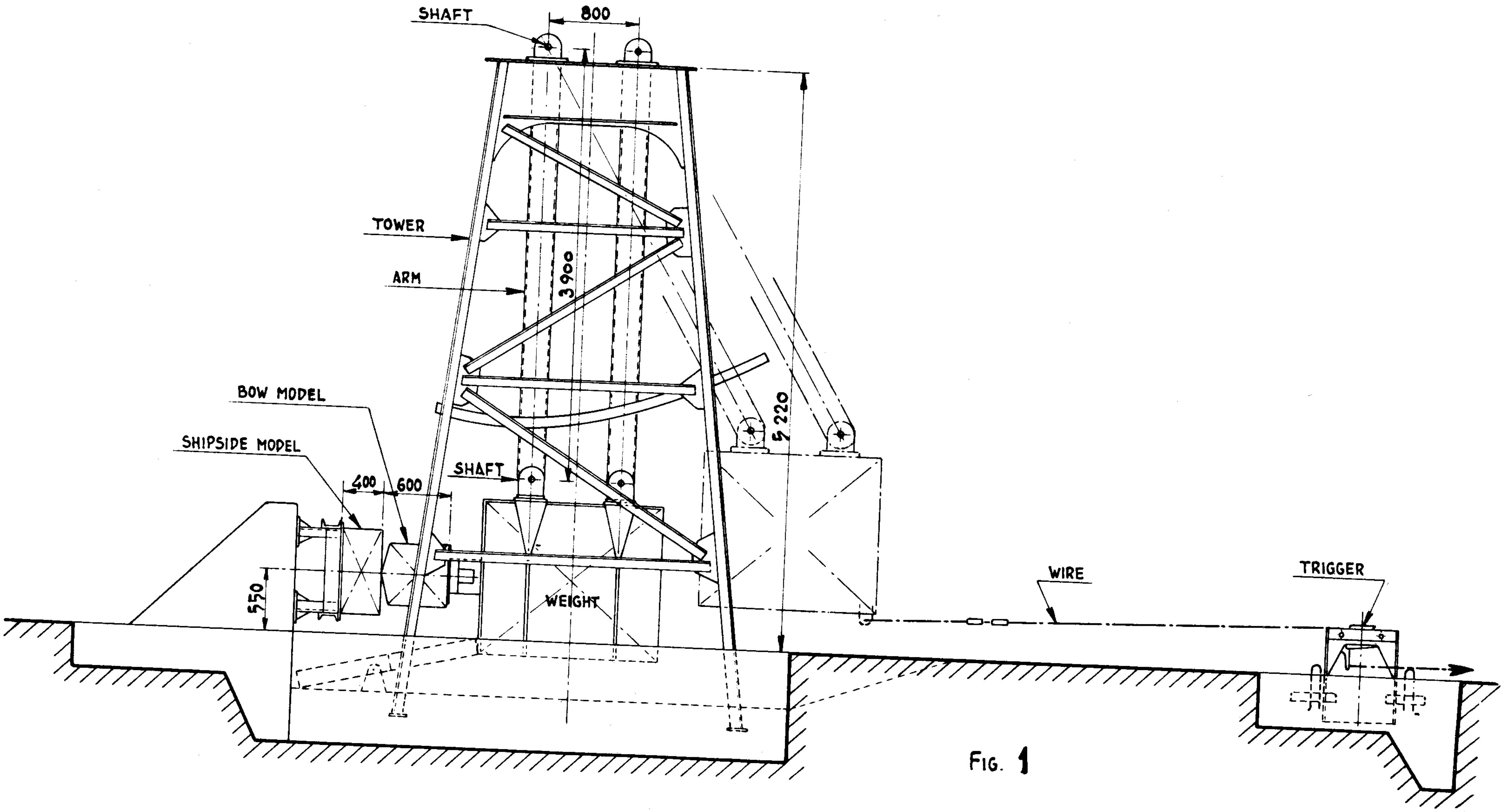


FIG. 1

INSTALLATION FOR COLLISION TESTS BETWEEN MODELS – C.N.E.N. STUDY CENTRE AT THE UNIVERSITY OF NAPLES

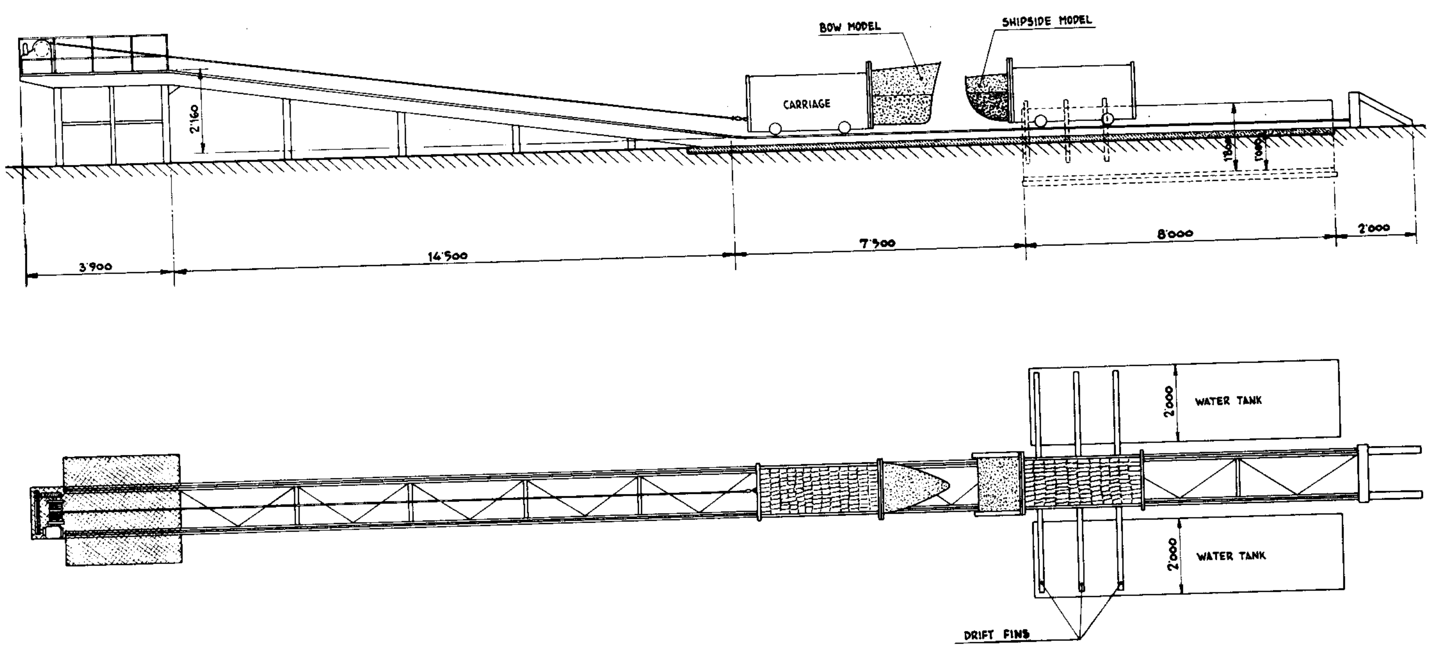
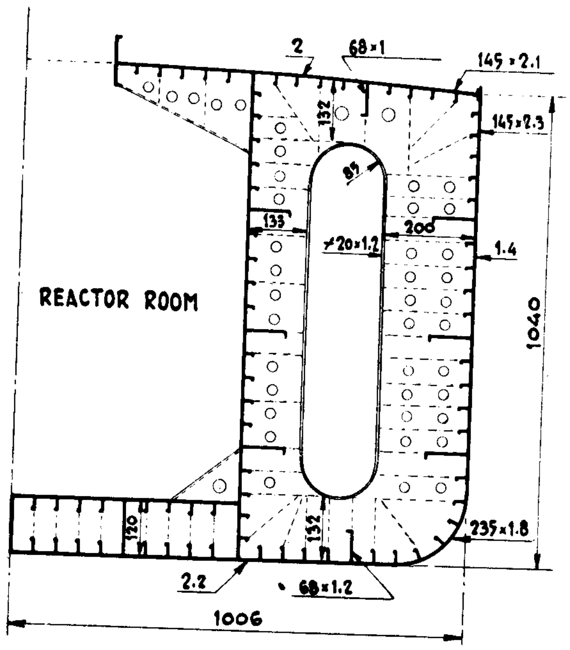
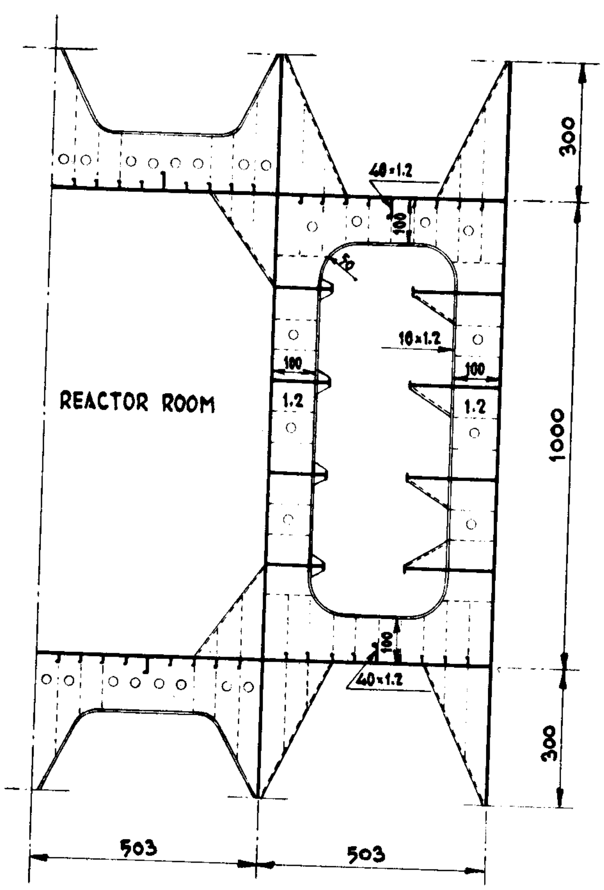


FIG. 2

SHIPSIDE MODEL T 3



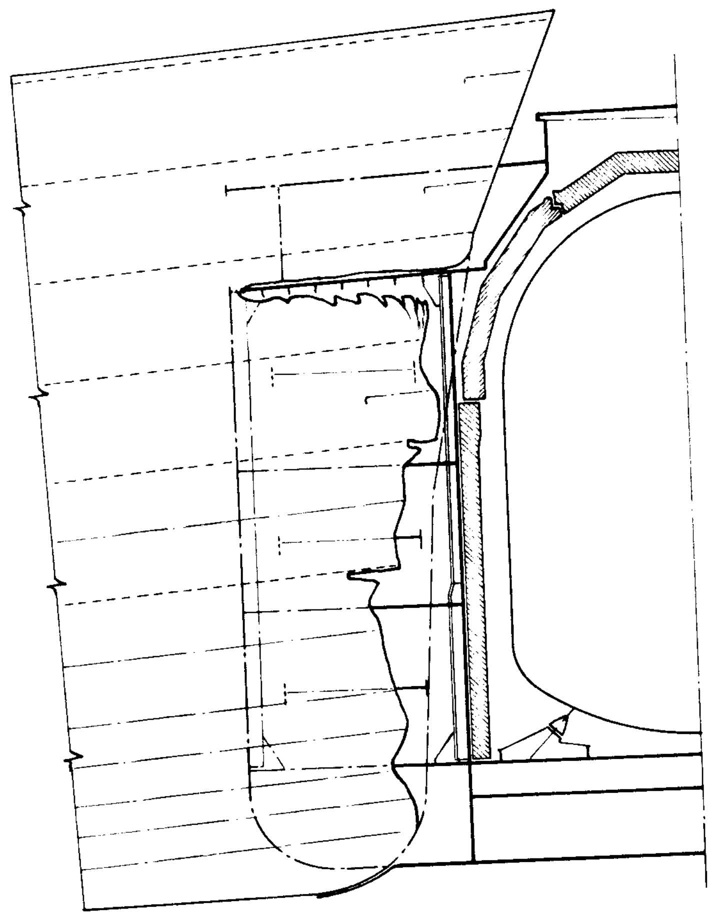


MARKED FIGURES ARE MODEL DIMENSIONS

FIG. 3

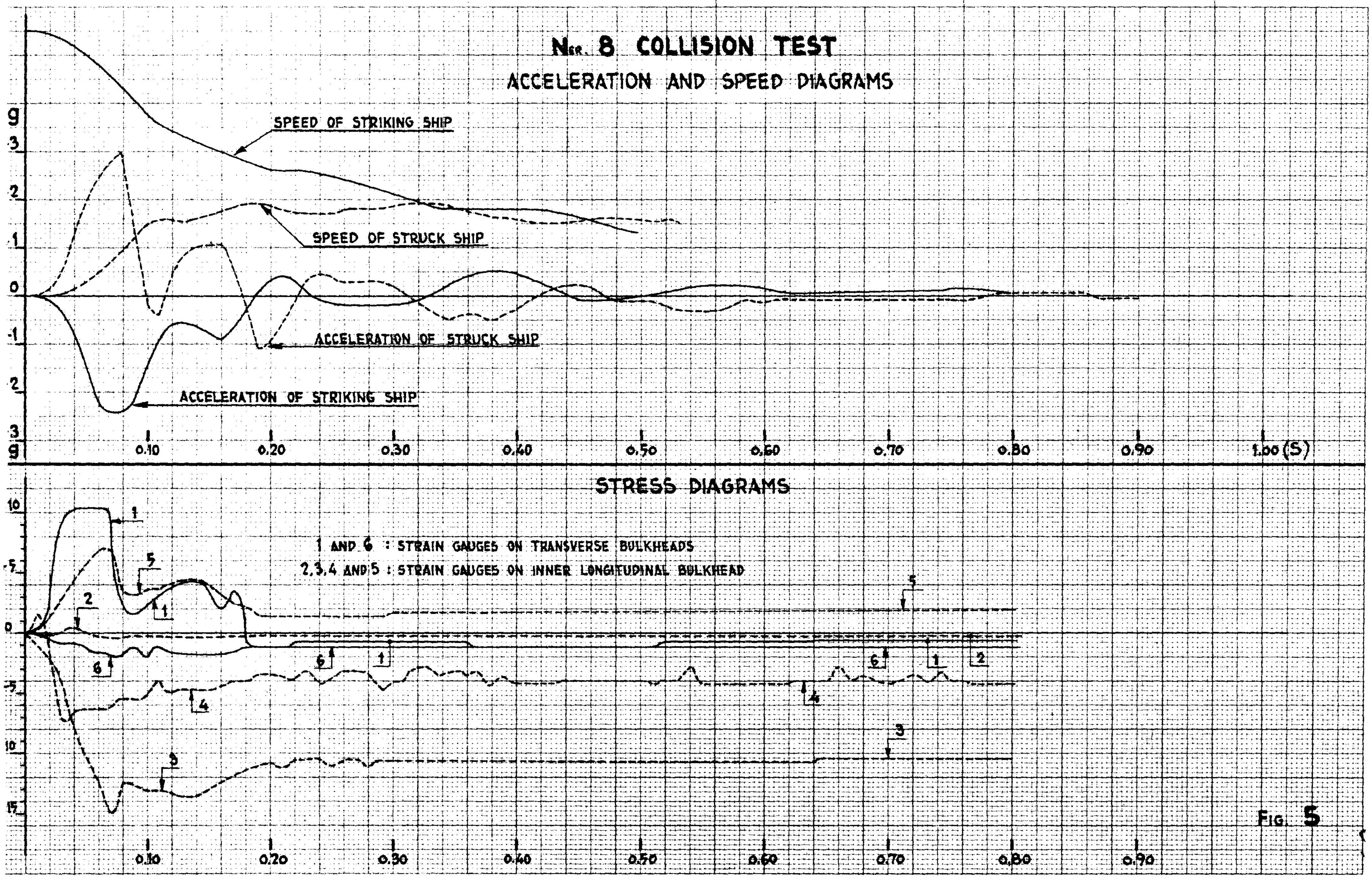
COLLISION TEST BY "GKSS" AND "DEUTSCHE WERFT A.G."

HAMBURG



(SEE "KERNTECHNIK, ISOTOPENTECHNIK UND - CHEMIE" YEAR 1967 - Nr. 8)

FIG. 4



Ner. 8 COLLISION TEST
ACCELERATION AND SPEED DIAGRAMS
SPEED OF STRIKING SHIP
SPEED OF STRUCK SHIP
ACCELERATION OF STRUCK SHIP
ACCELERATION OF STRIKING SHIP
g
3
2
1
0
-1
2
3
g
0.10
0.20
0.30
0.40
0.50
0.60
0.70
0.80
0.90
1.00 (S)
STRESS DIAGRAMS
1 AND 6 : STRAIN GAUGES ON TRANSVERSE BULKHEADS
2,3,4 AND 5 : STRAIN GAUGES ON INNER LONGITUDINAL BULKHEAD
10
5
0
5
10
15
1
2
3
4
5
6

FIG. 5

SAMPLES FOR WELDING AND SOLDERING TESTS

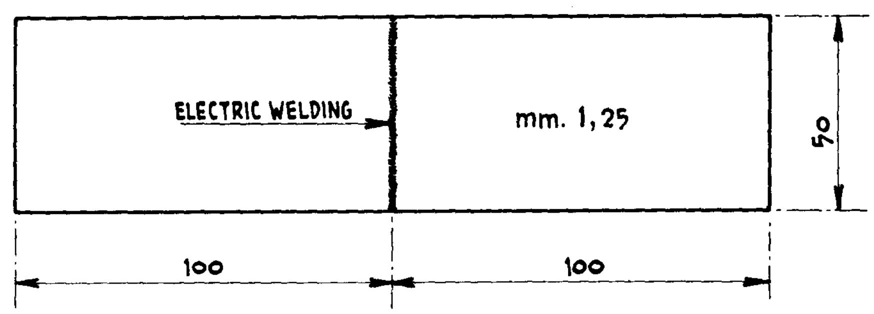


FIG. 6

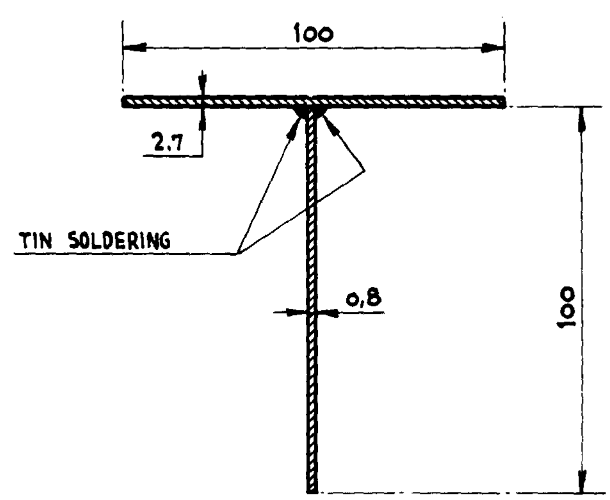


FIG. 7

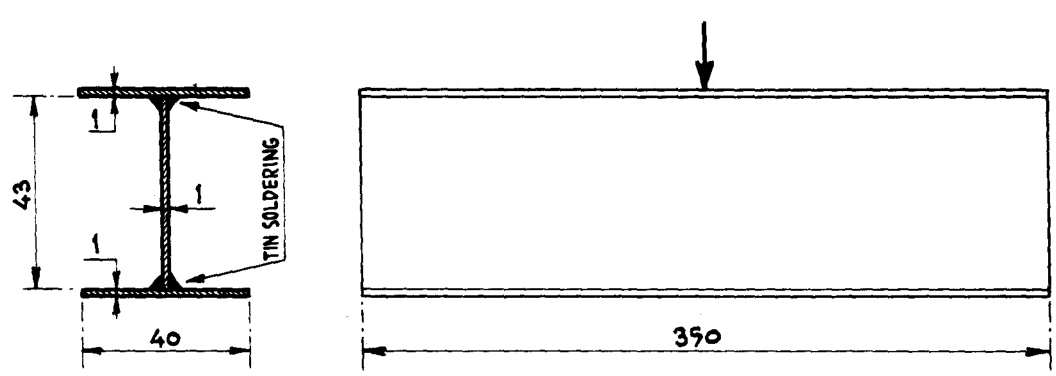


FIG. 8

SAMPLES FOR TIN SOLDERING TESTS

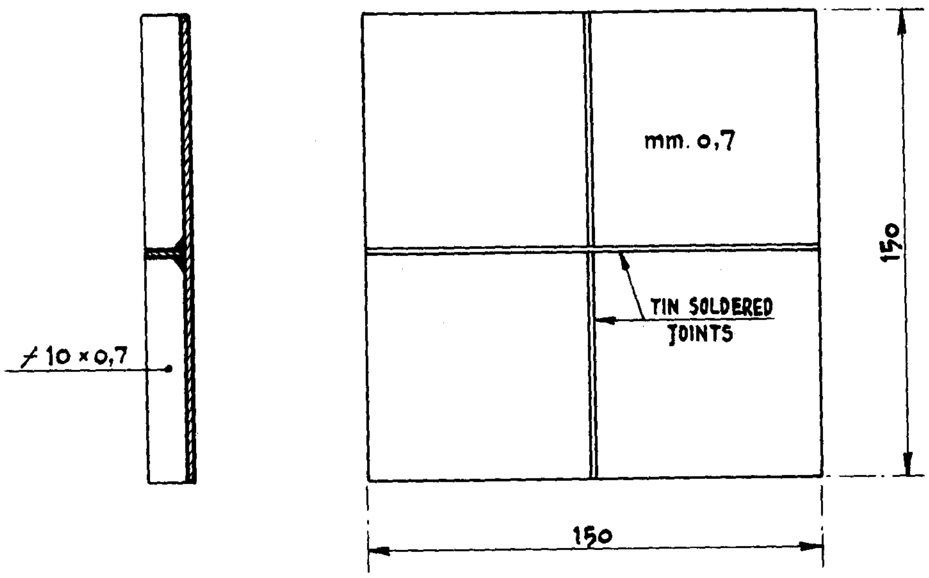


FIG. 9

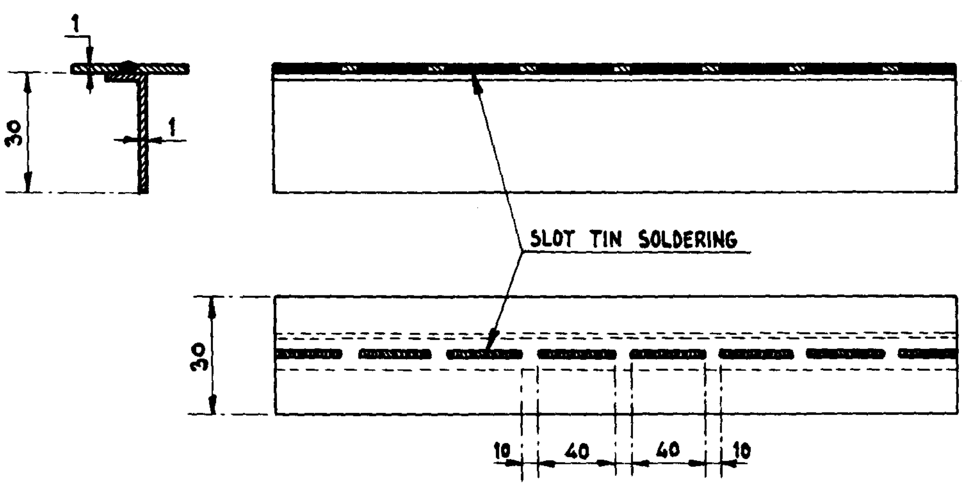
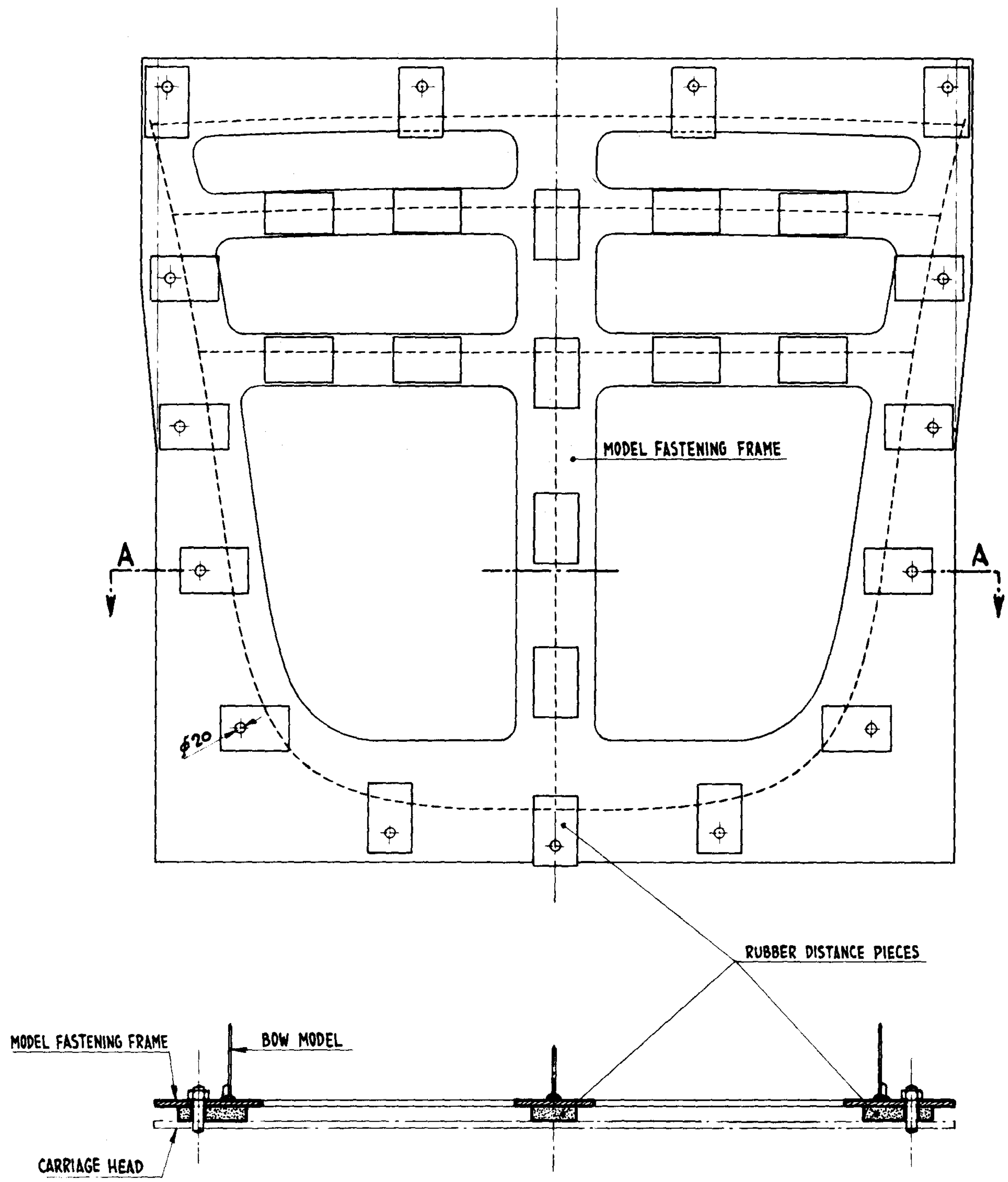


FIG. 10

ELASTIC FASTENING FOR BOW MODEL



A - A SECTION

FIG. 11

FASTENING OF THE SHIPSIDE MODEL

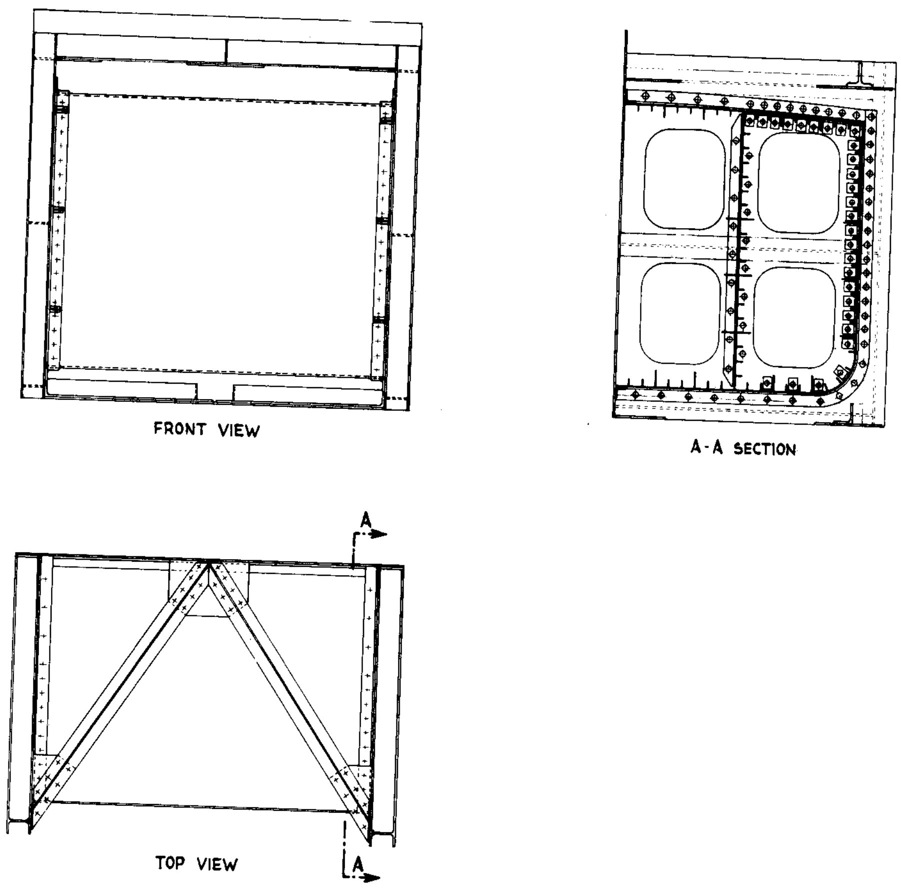
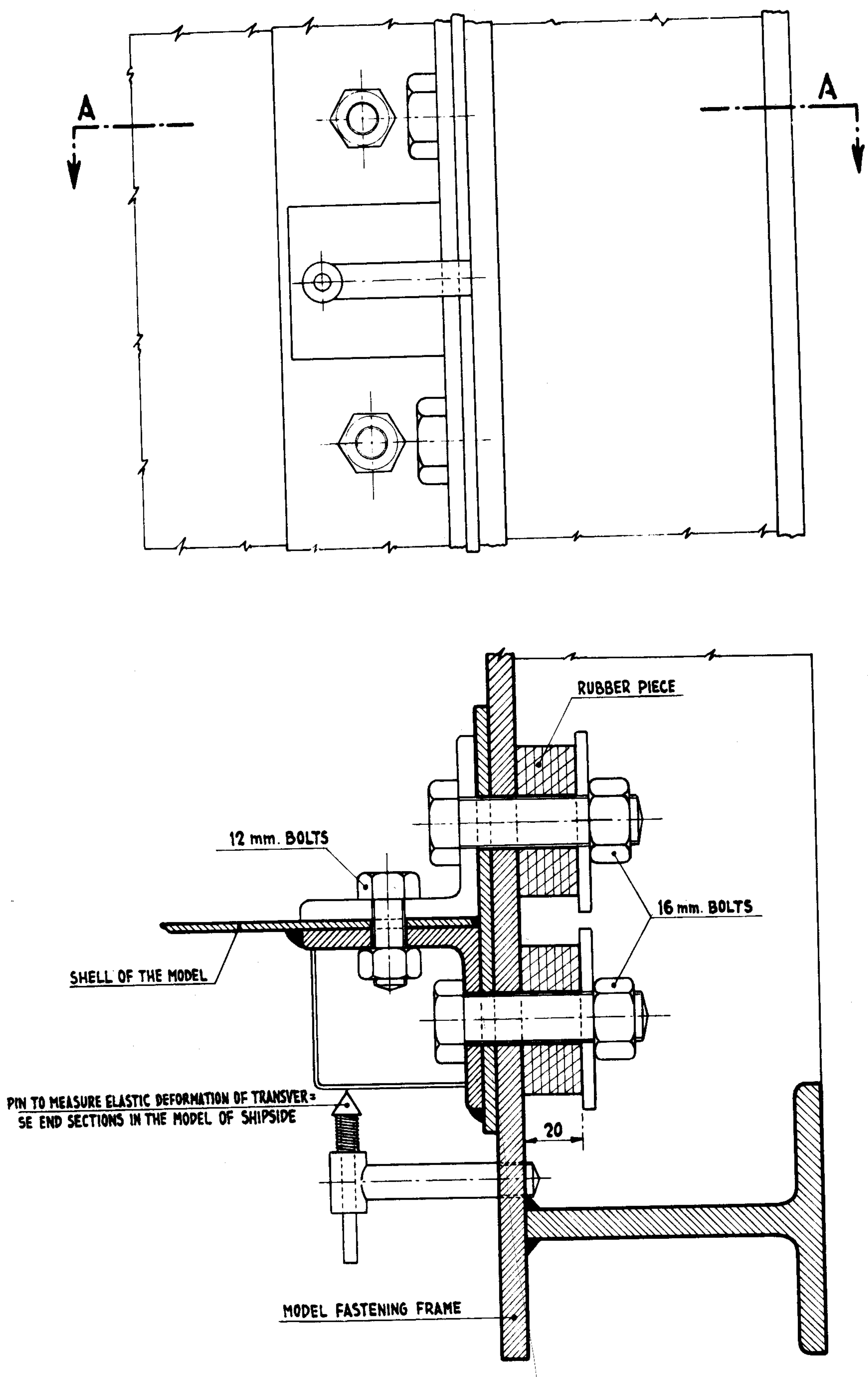


Fig. 12

ELASTIC FASTENING FOR SHIPSIDE MODEL



A-A SECTION

FIG. 13

BOW MODEL FOR THE 1st COLLISION TEST

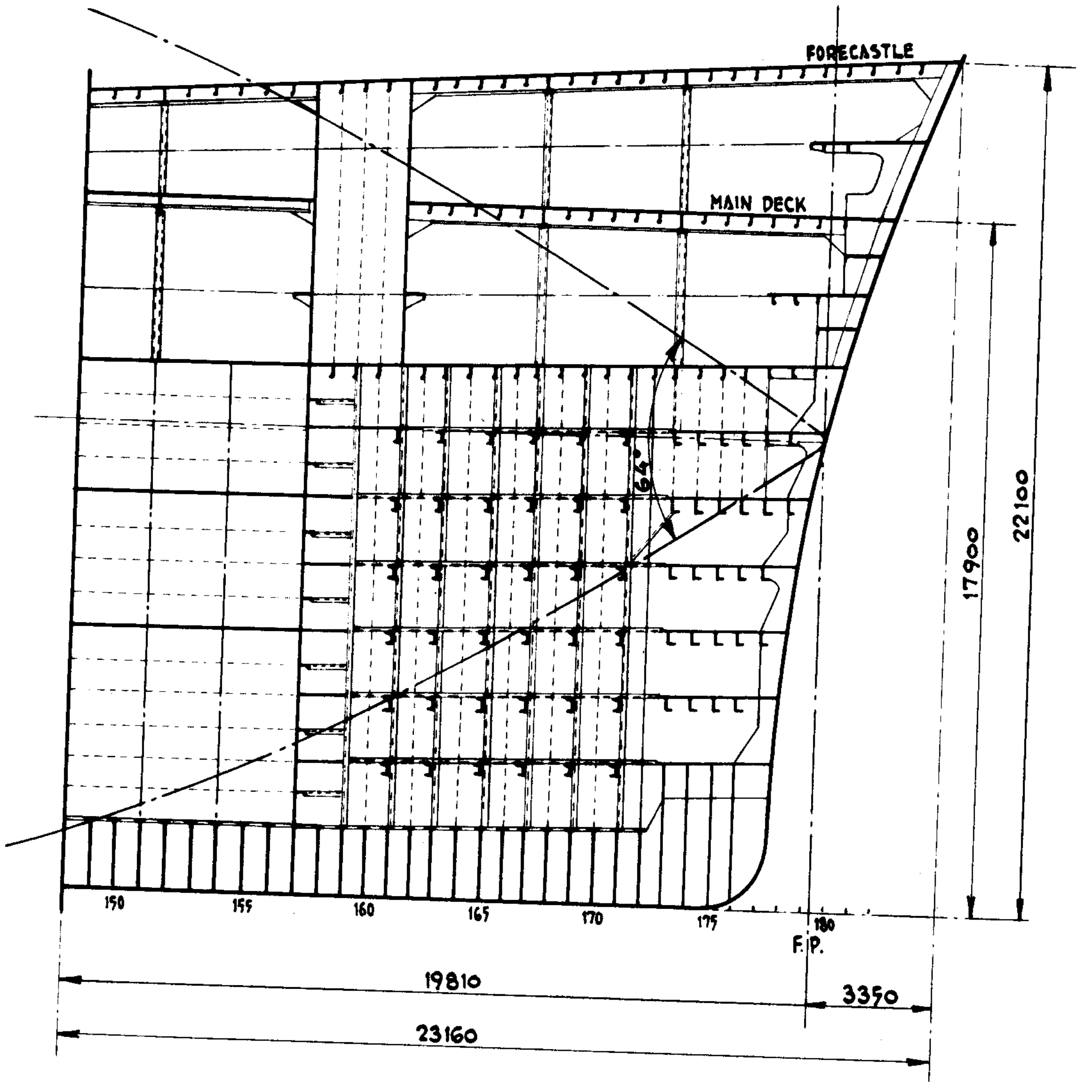
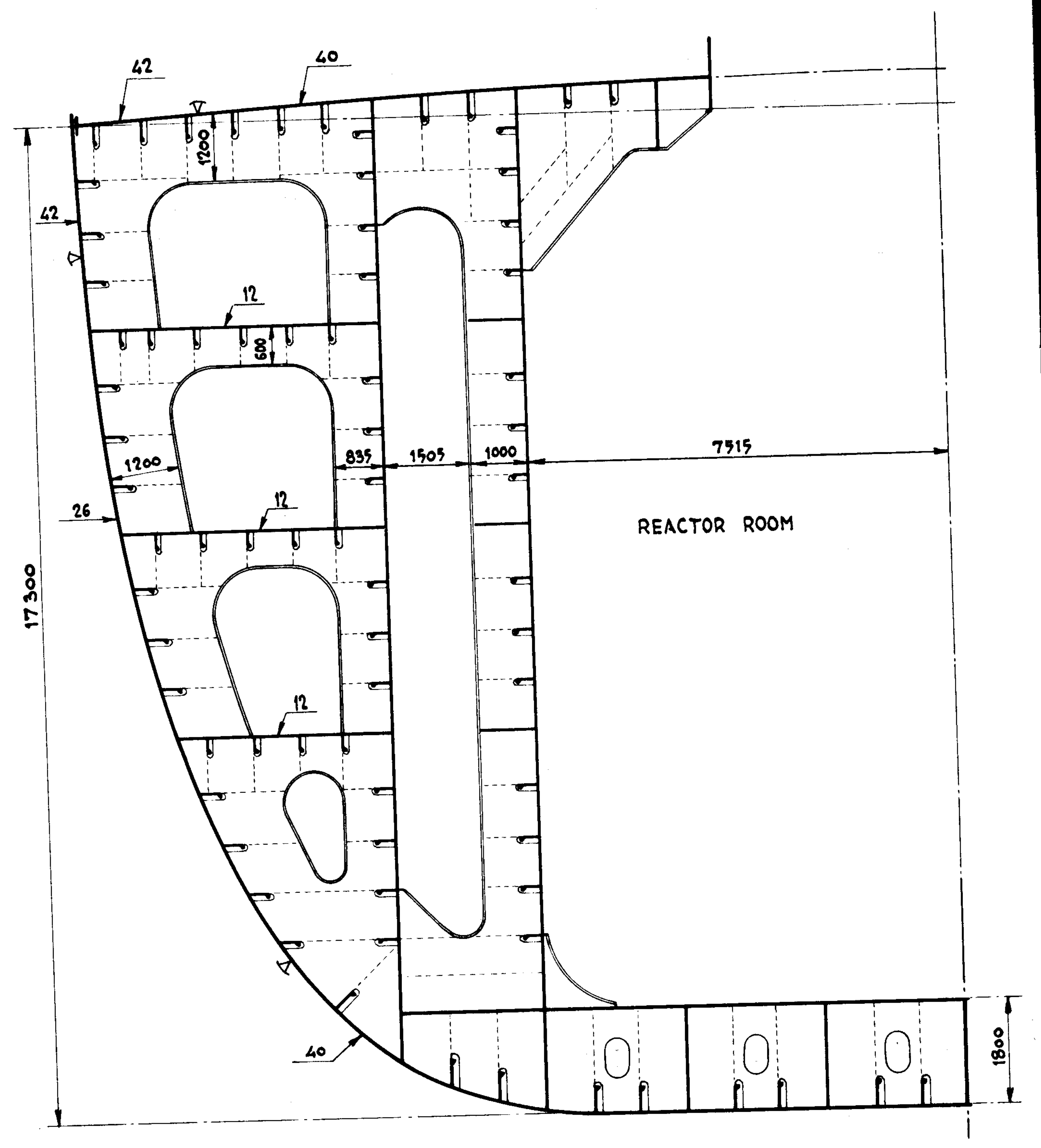


FIG. 14

SHIP CROSS SECTION WITH COLLISION BARRIER



MARKED FIGURES ARE FULL SIZE SHIP DIMENSIONS

FIG 15

SHIPSIDE MODEL FOR THE 1st COLLISION TEST

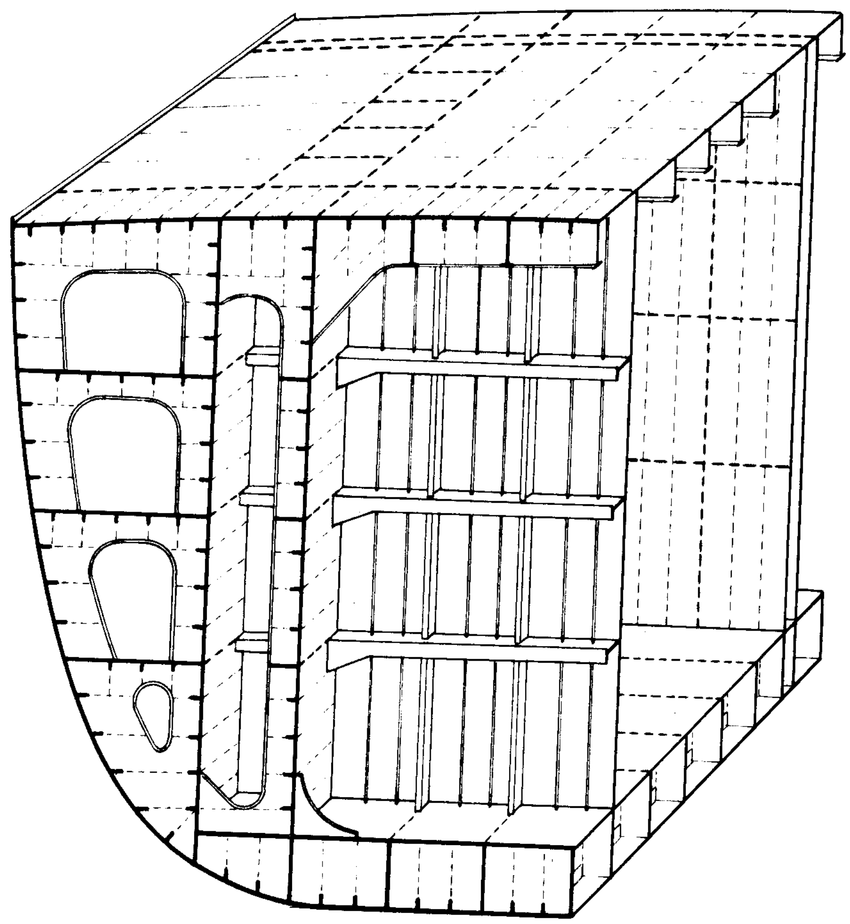


FIG. 16

BOW AND SHIPSIDE MODELS AFTER 1st COLLISION TEST

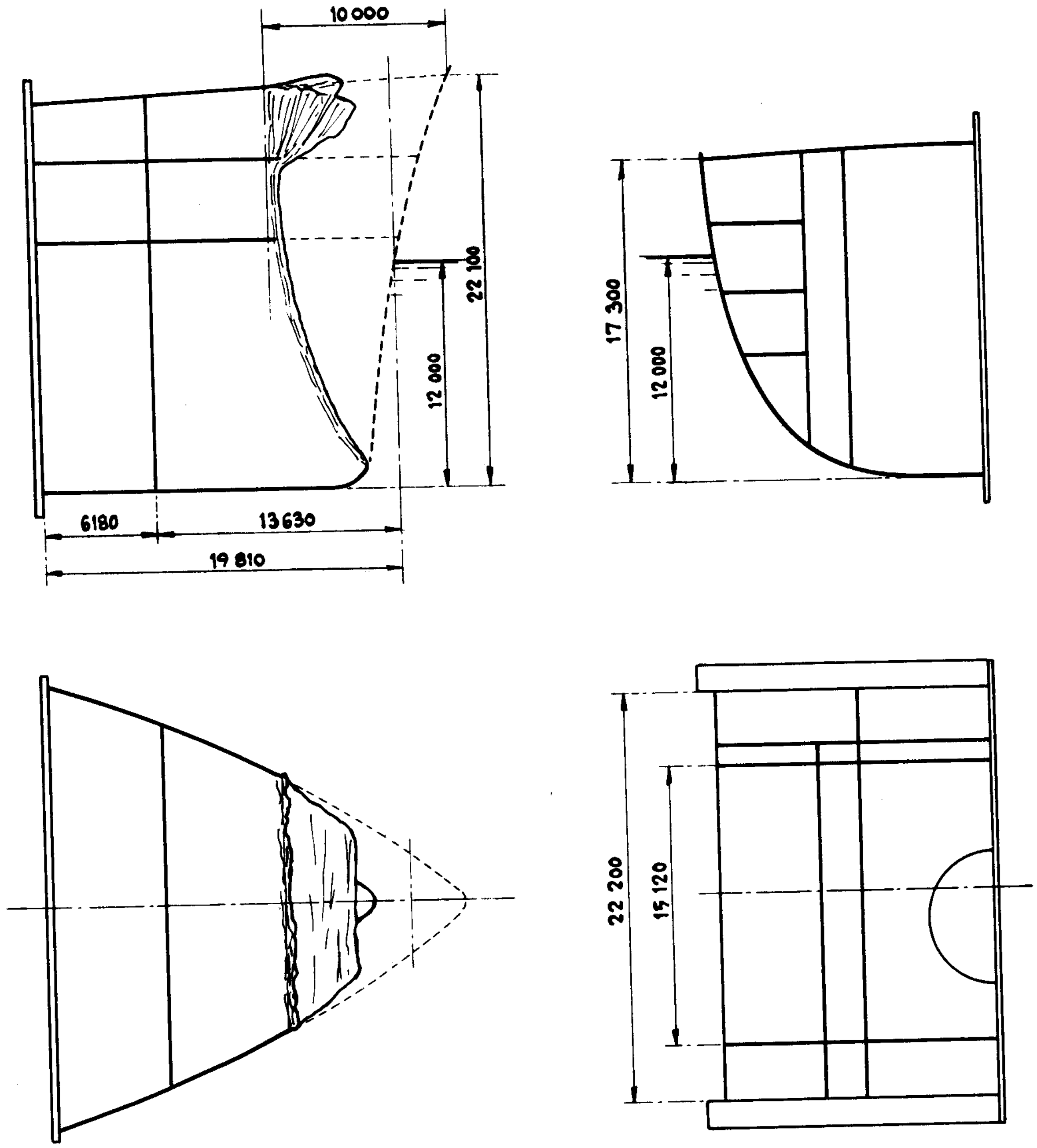


FIG. 17

BOW MODEL FOR THE 2ND COLLISION TEST

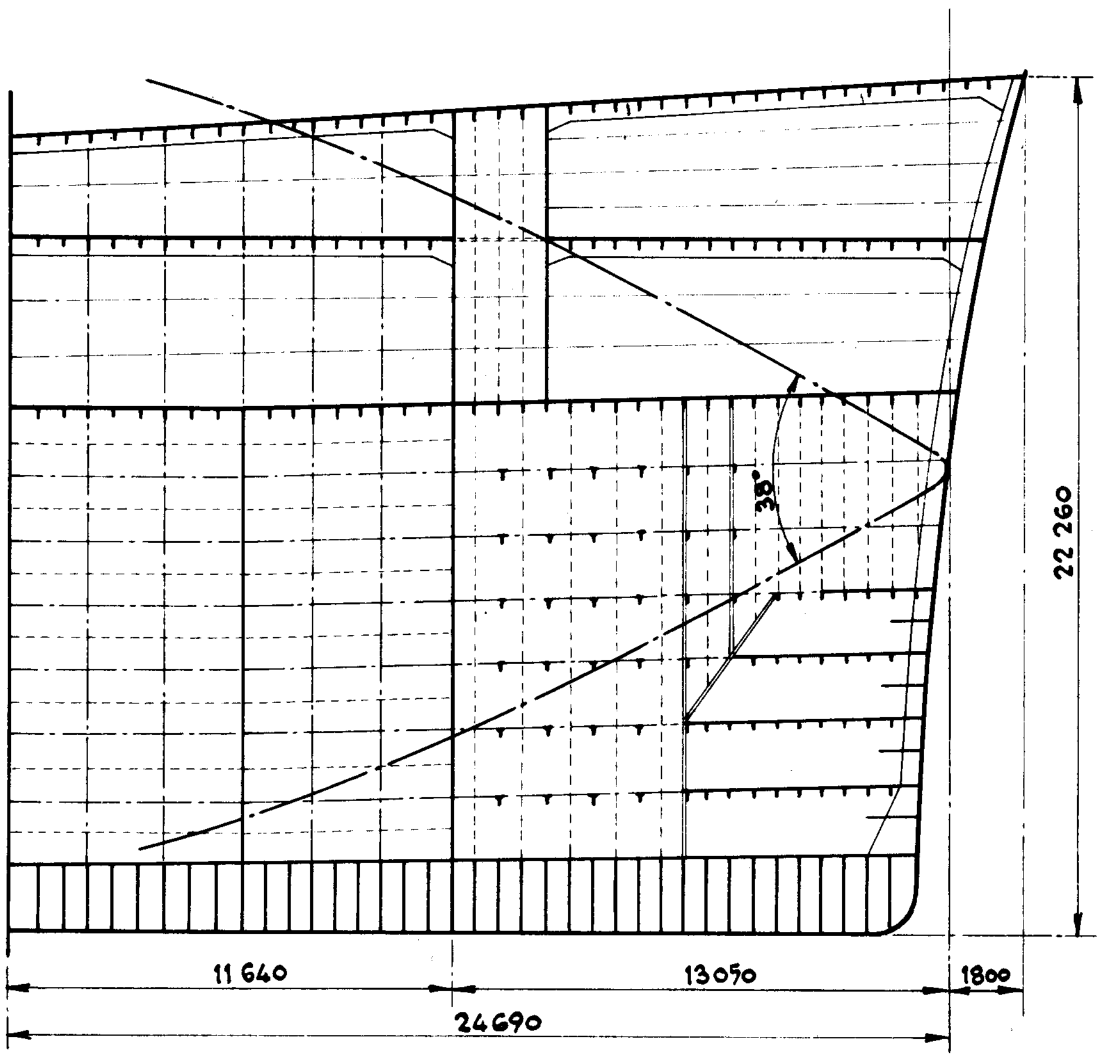


FIG. 18

BOW AND SHIPSIDE MODELS AFTER 2ND COLLISION TEST

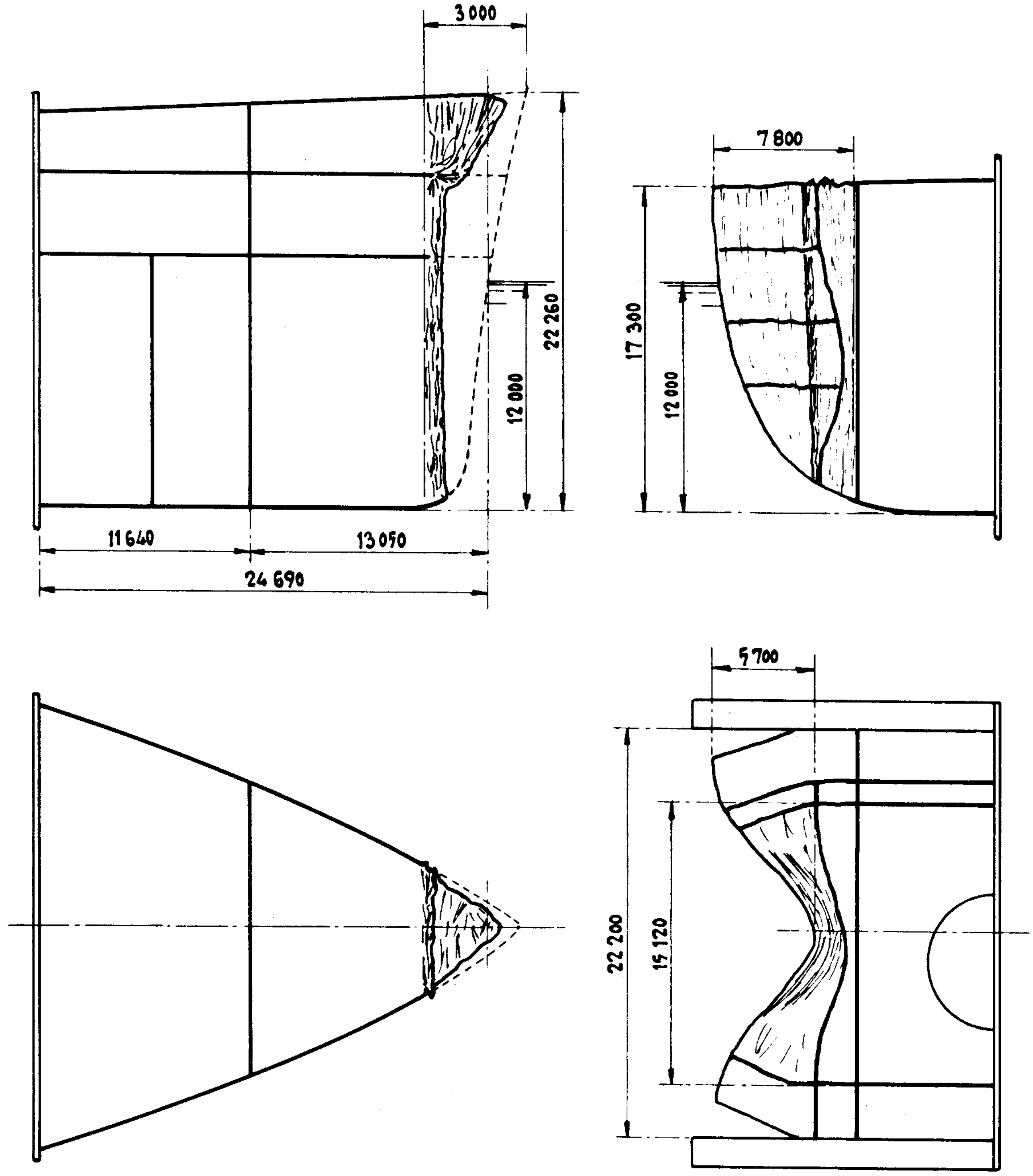


FIG. 19

SHIPSIDE MODEL FOR THE 3RD COLLISION TEST

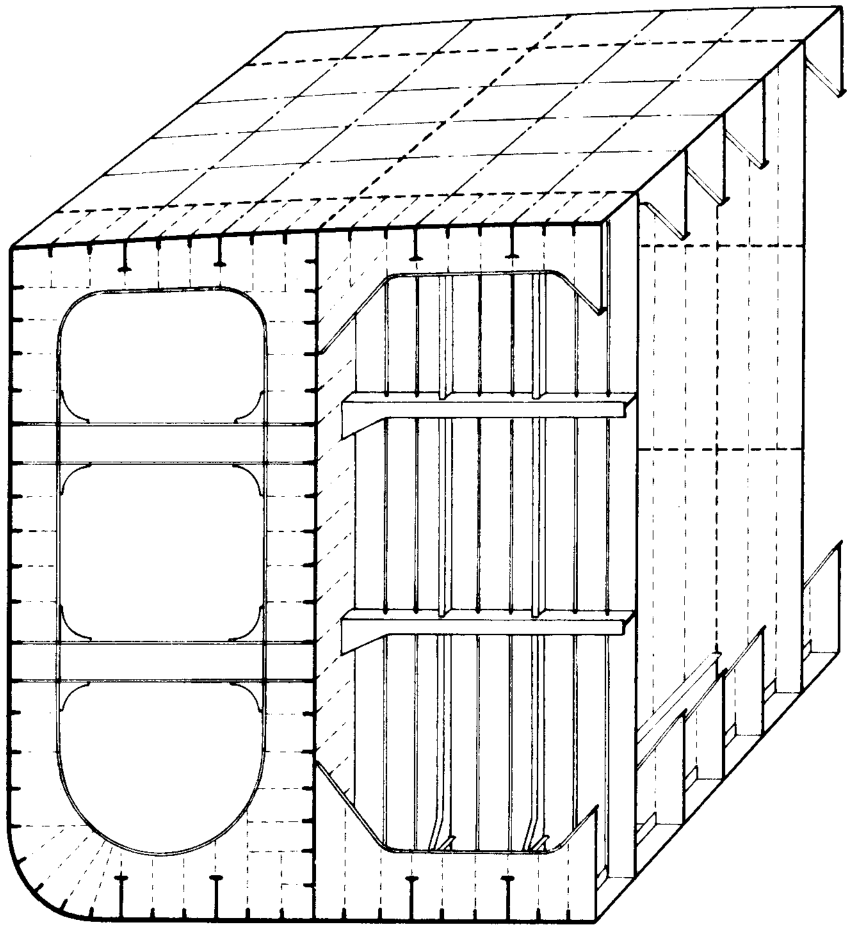


FIG. 20

BOW AND SHIPSIDE MODELS AFTER 3RD COLLISION TEST

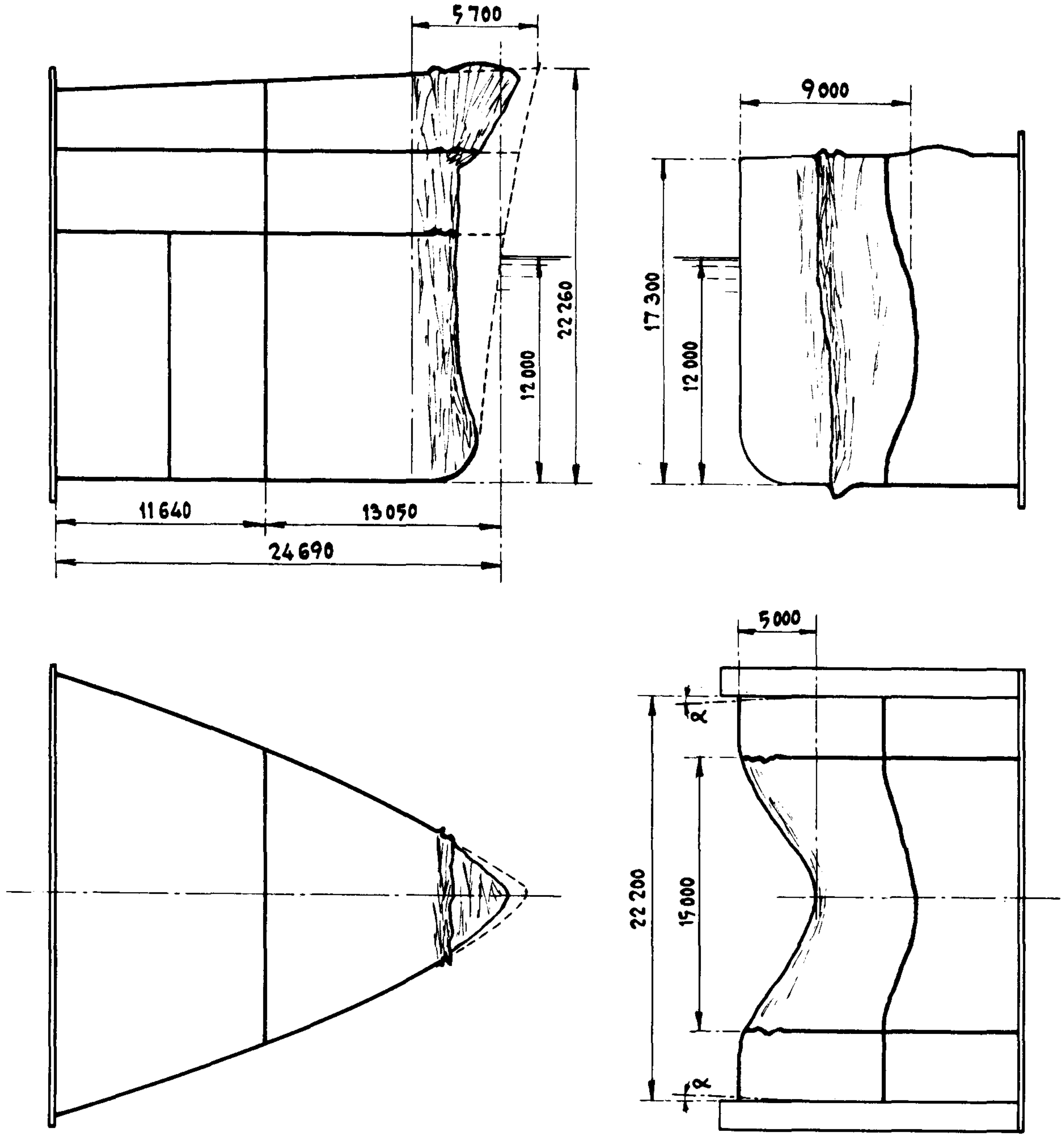


FIG. 21

BOW AND SHIPSIDE MODELS AFTER 4TH COLLISION TEST

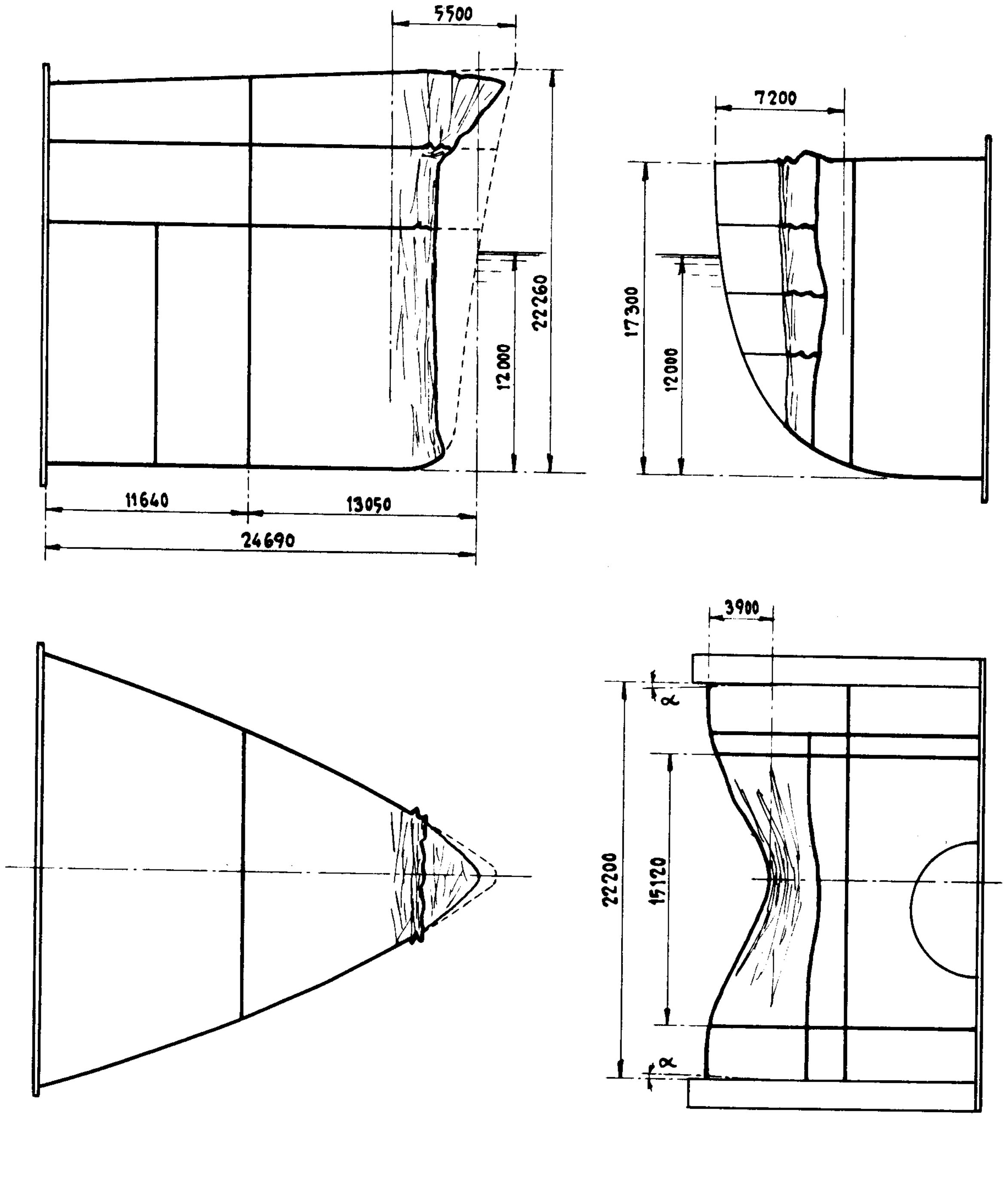


FIG. 22

BOW AND SHIPSIDE MODELS AFTER 5TH COLLISION TEST

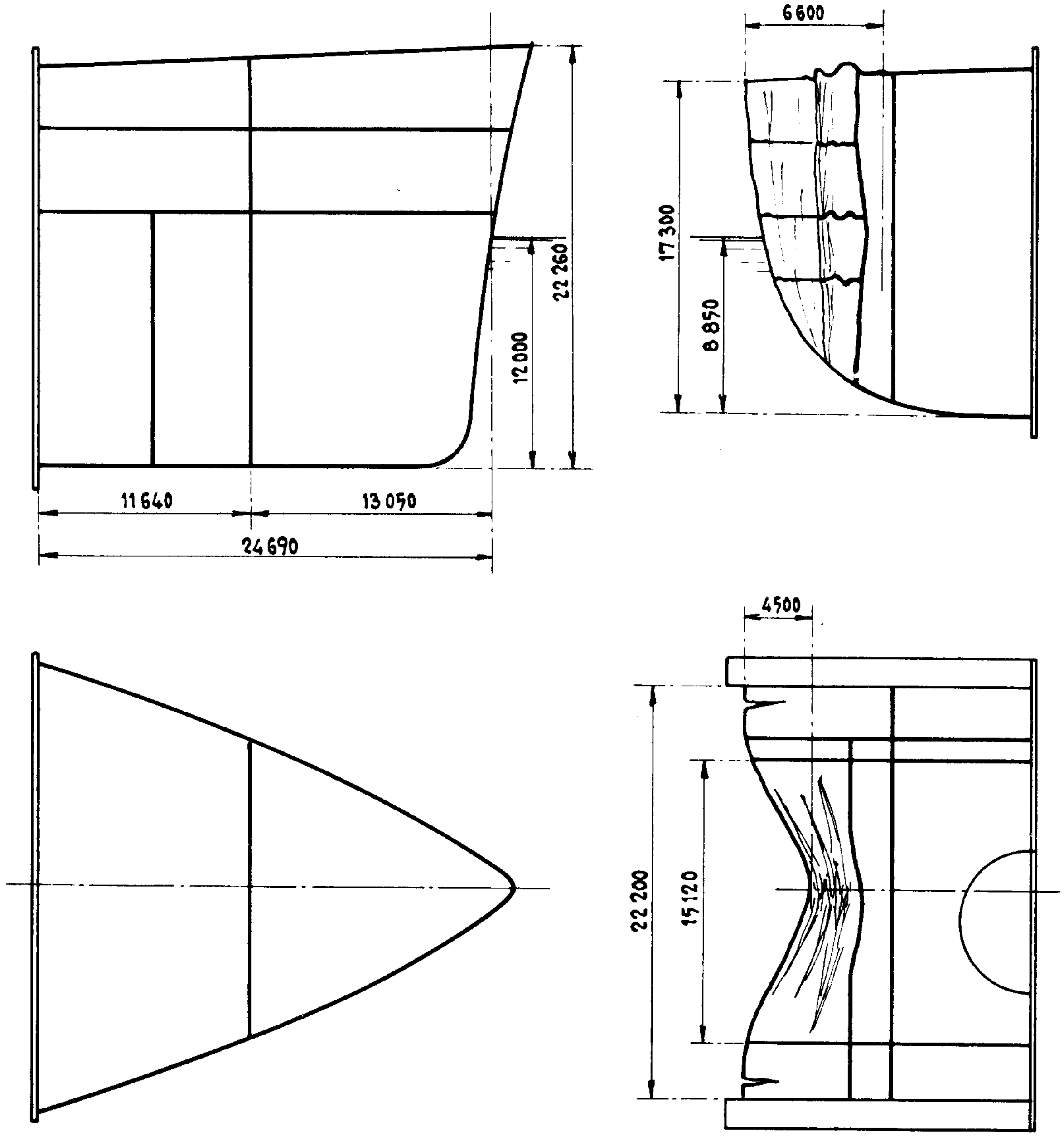


FIG. 23

BOW AND SHIPSIDE MODELS BEFORE 6TH COLLISION TEST

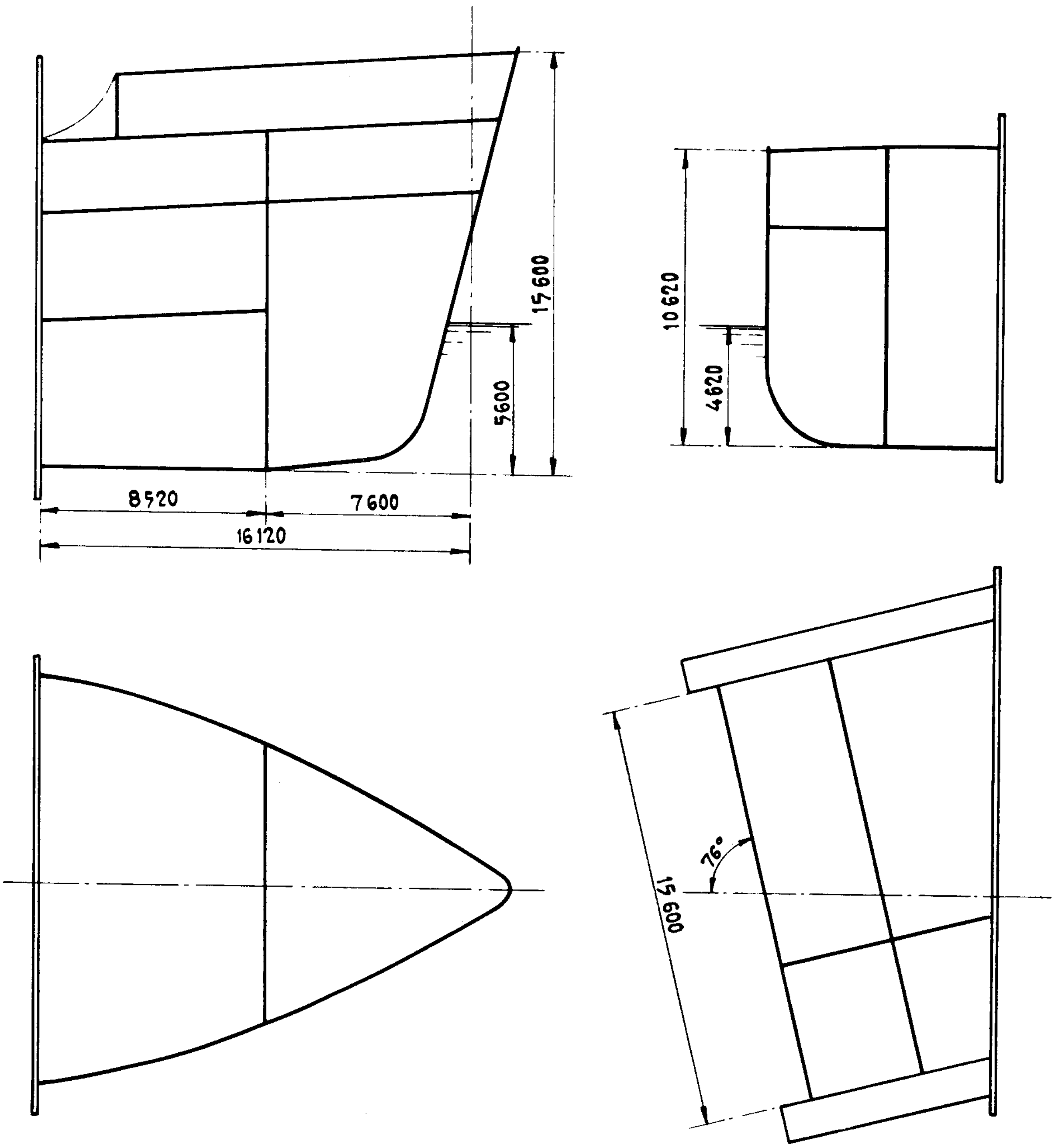


FIG. 24

BOW AND SHIPSIDE MODELS AFTER 6TH COLLISION TEST

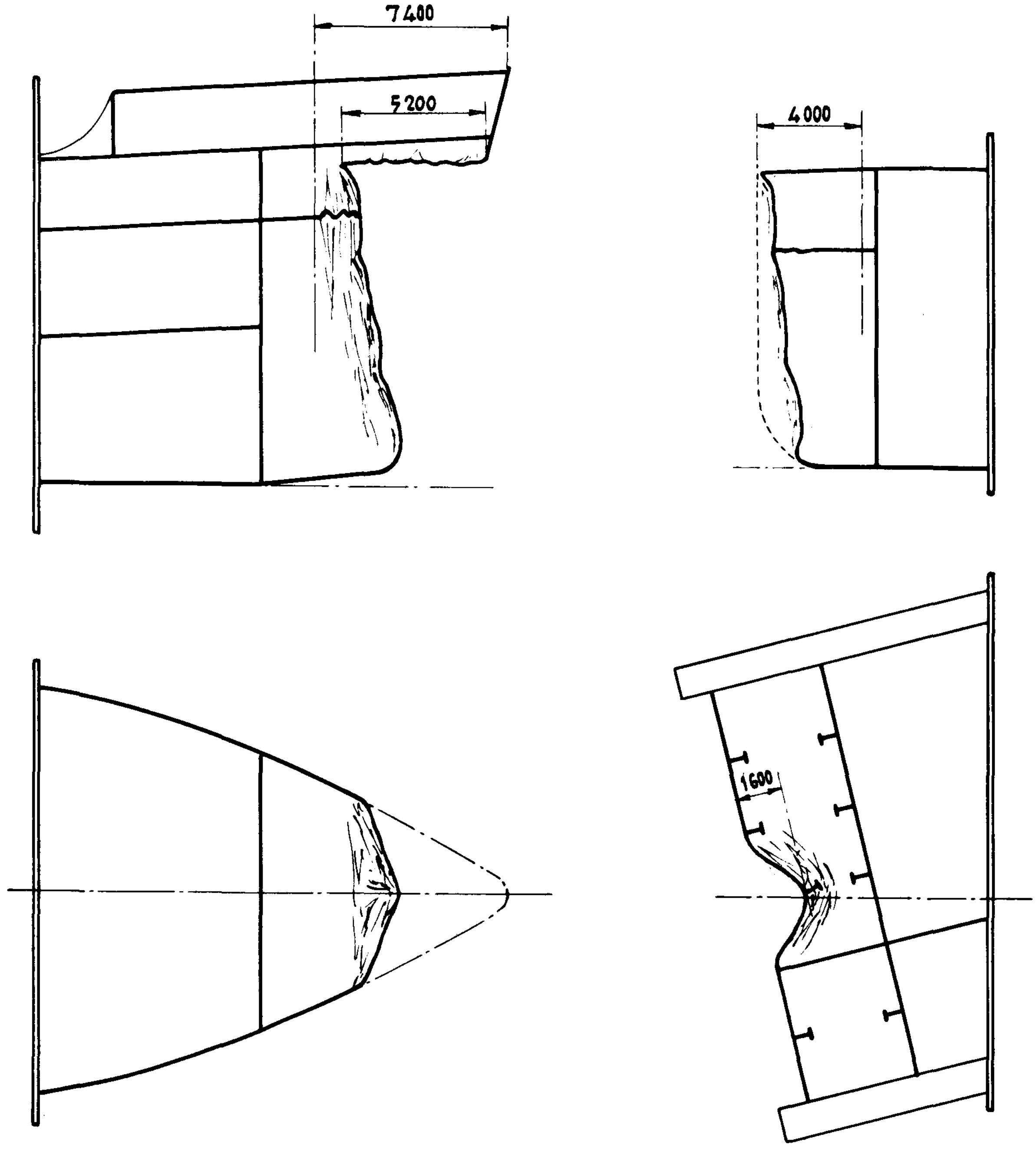


FIG. 25

BOW AND SHIPSIDE MODELS AFTER 7TH COLLISION TEST

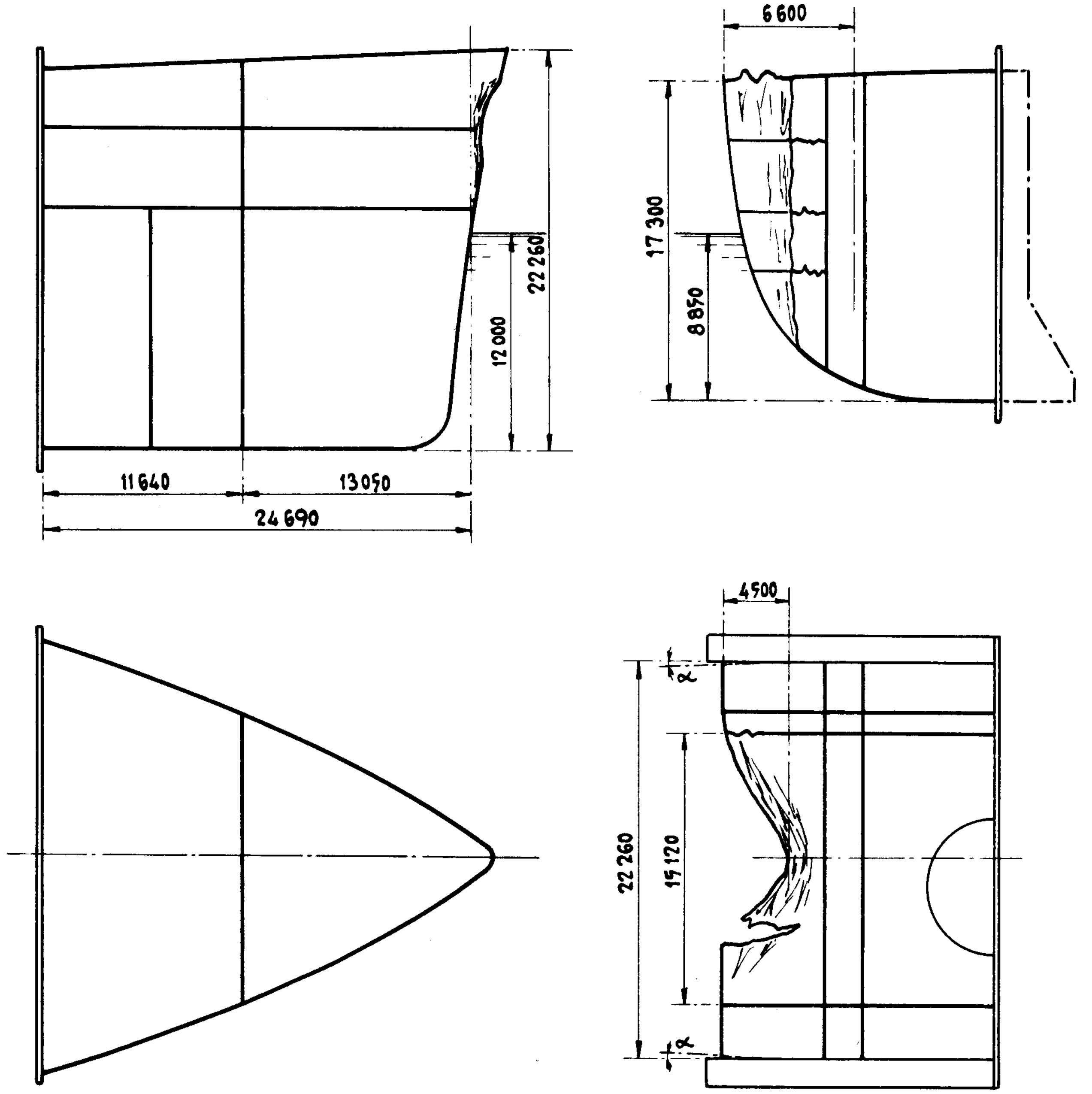


FIG. 26

BOW AND SHIPSIDE MODELS BEFORE 8TH COLLISION TEST

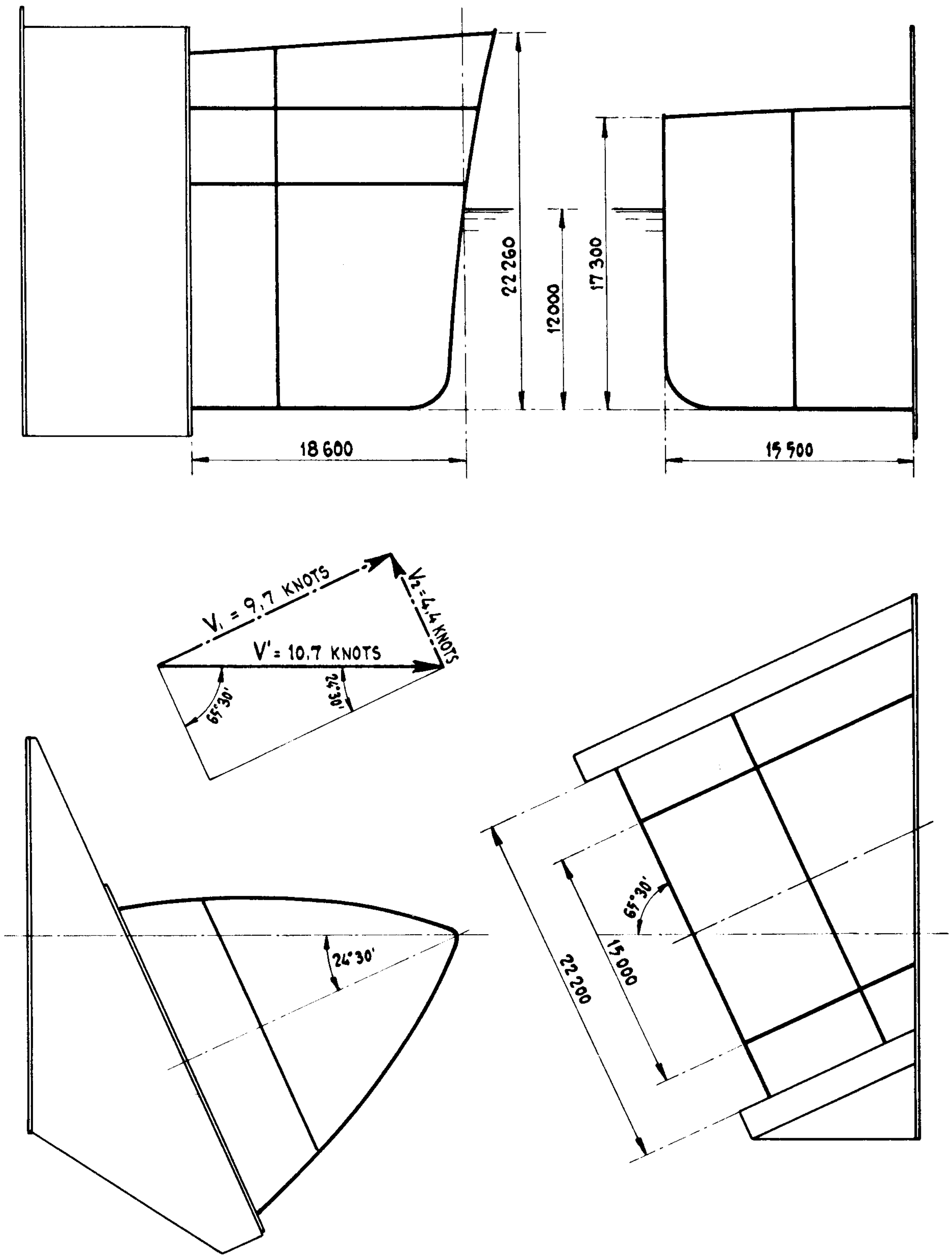


FIG. 27

BOW AND SHIPSIDE MODELS AFTER 8TH COLLISION TEST

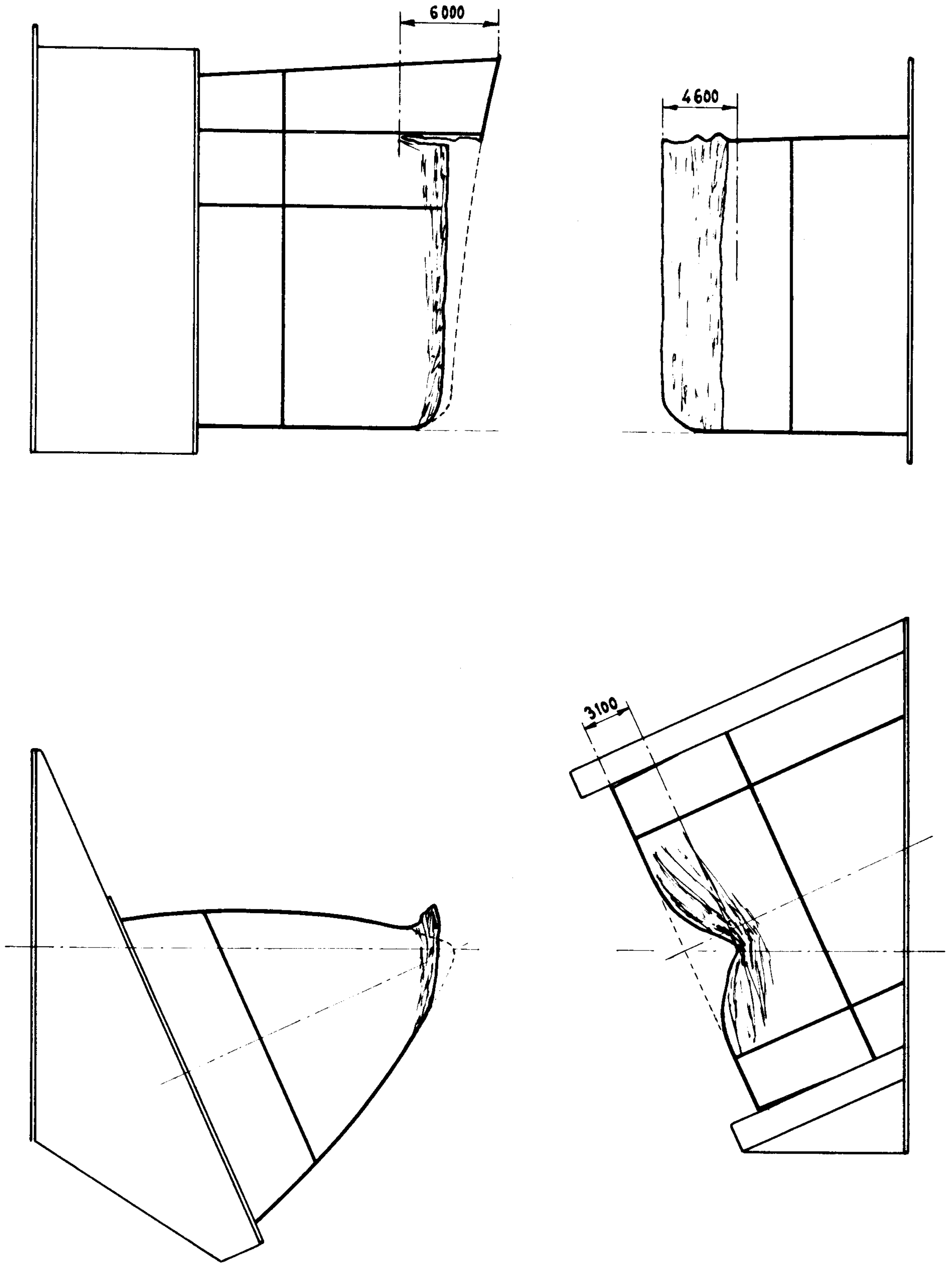


FIG. 28

BOW AND SHIPSIDE MODELS AFTER 9TH COLLISION TEST

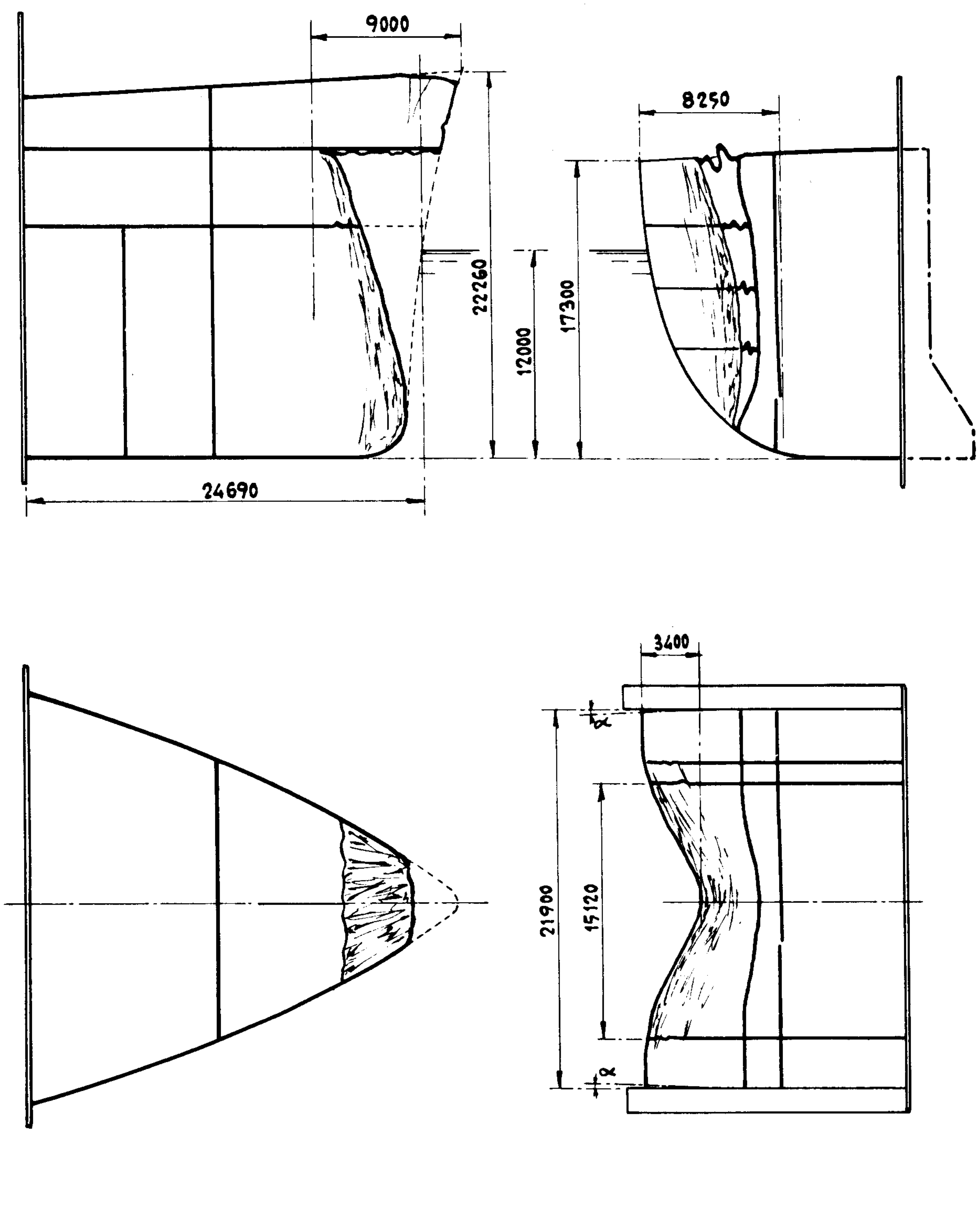


FIG. 29

BOW AND SHIPSIDE MODELS BEFORE 10TH COLLISION TEST

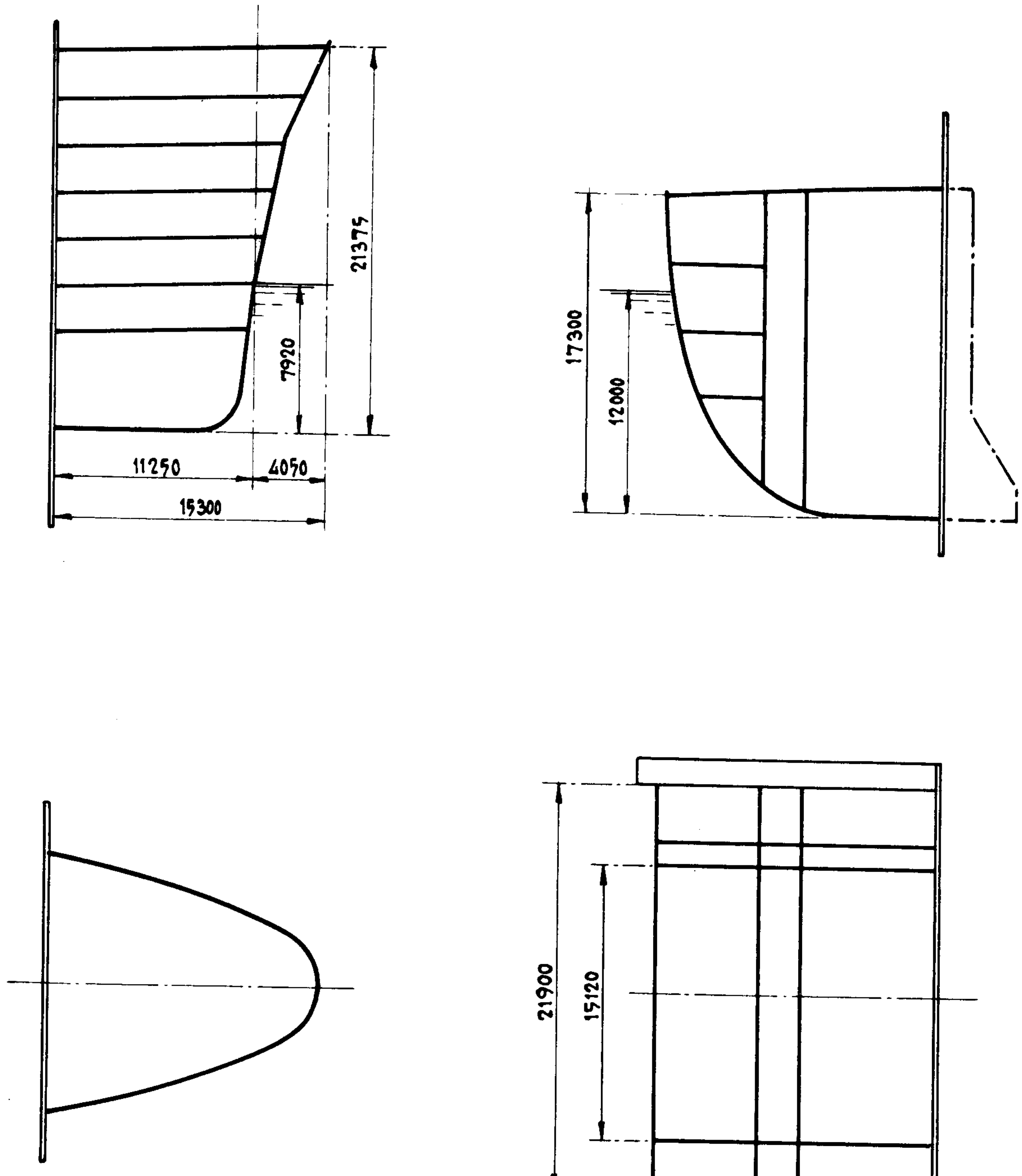


Fig. 30

BOW AND SHIPSIDE MODELS AFTER 10TH COLLISION TEST

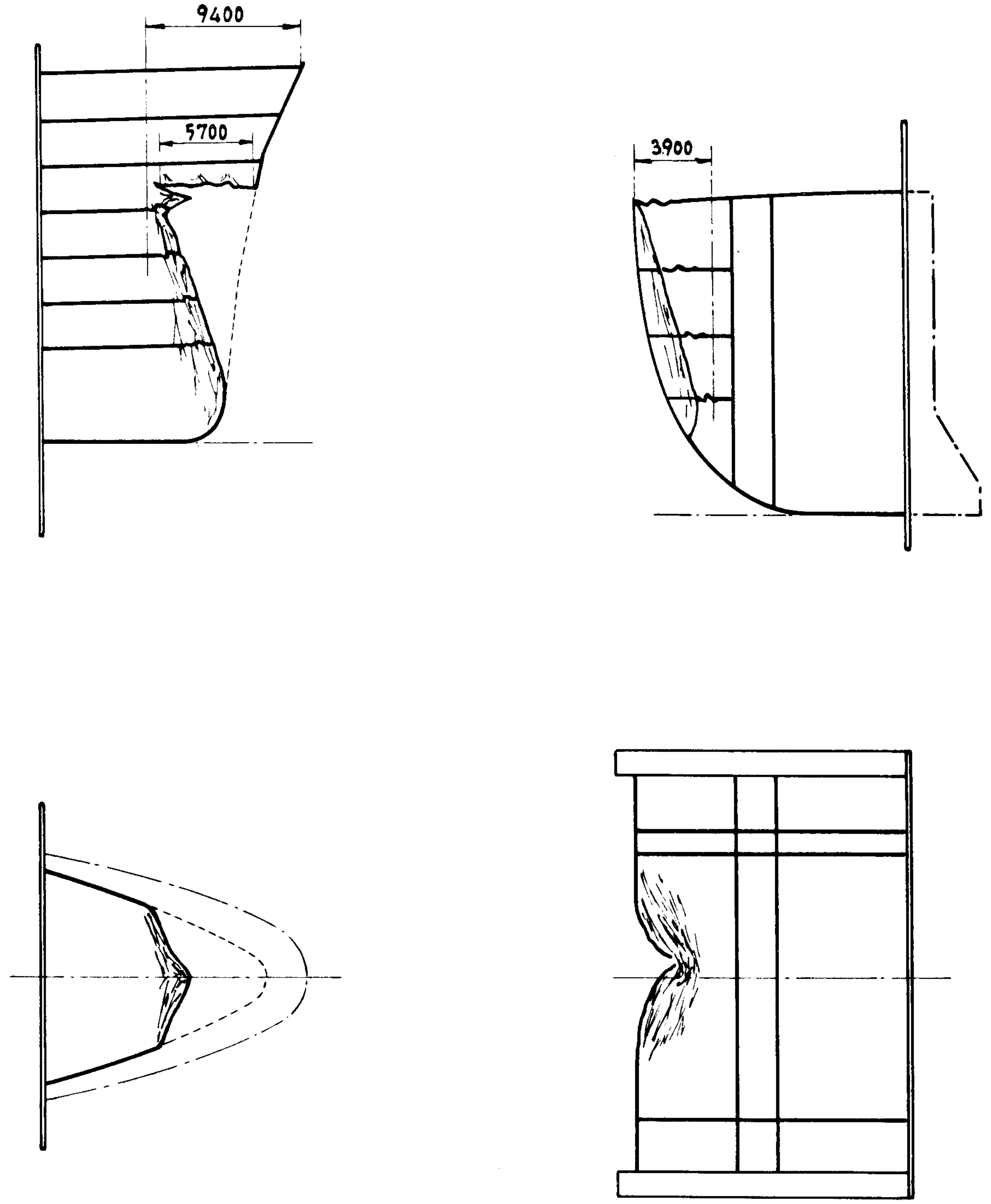


FIG. 31

BOW AND SHIPSIDE MODELS BEFORE 11TH COLLISION TEST

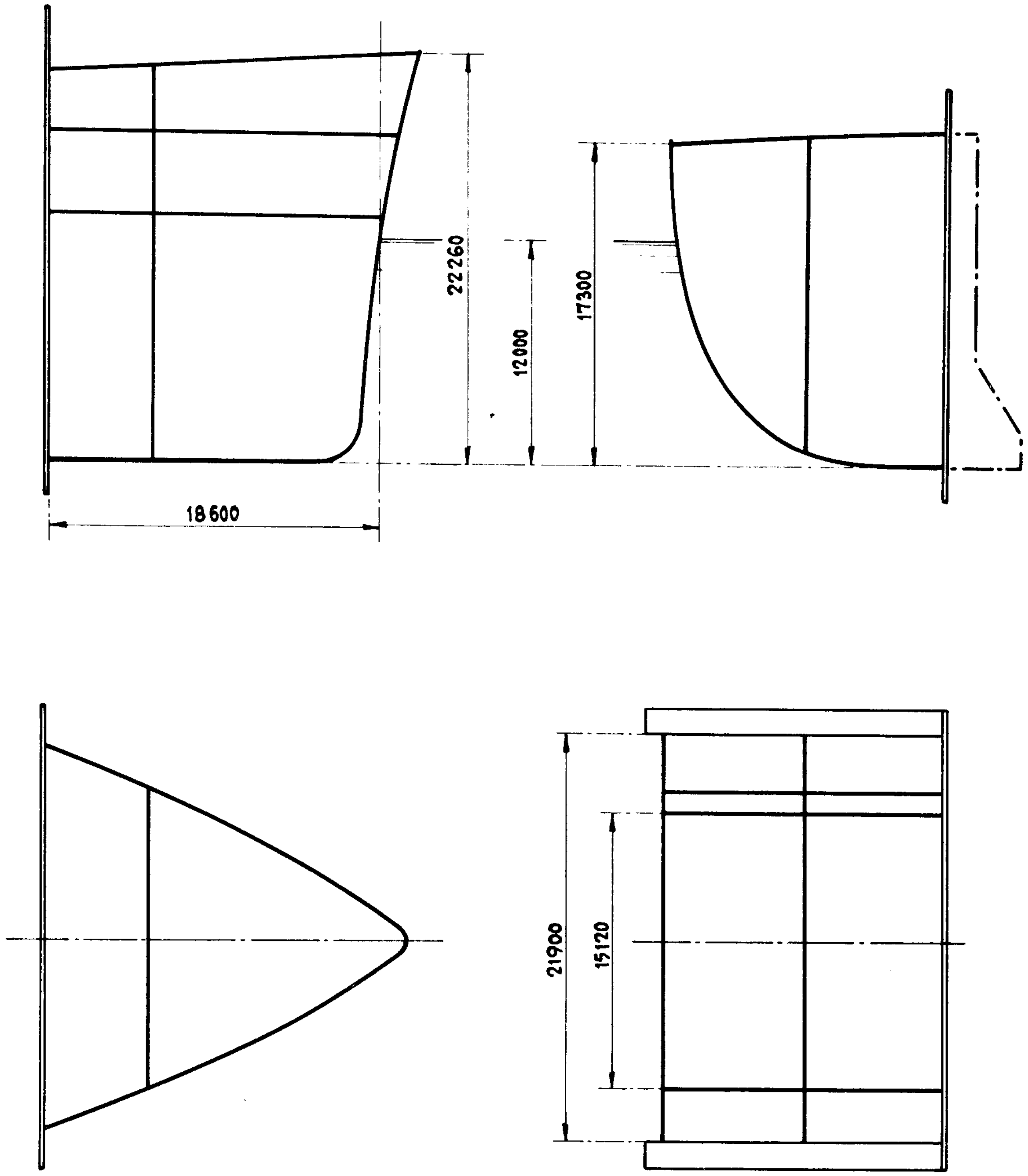


FIG. 32

BOW AND SHIPSIDE MODELS AFTER 11TH COLLISION TEST

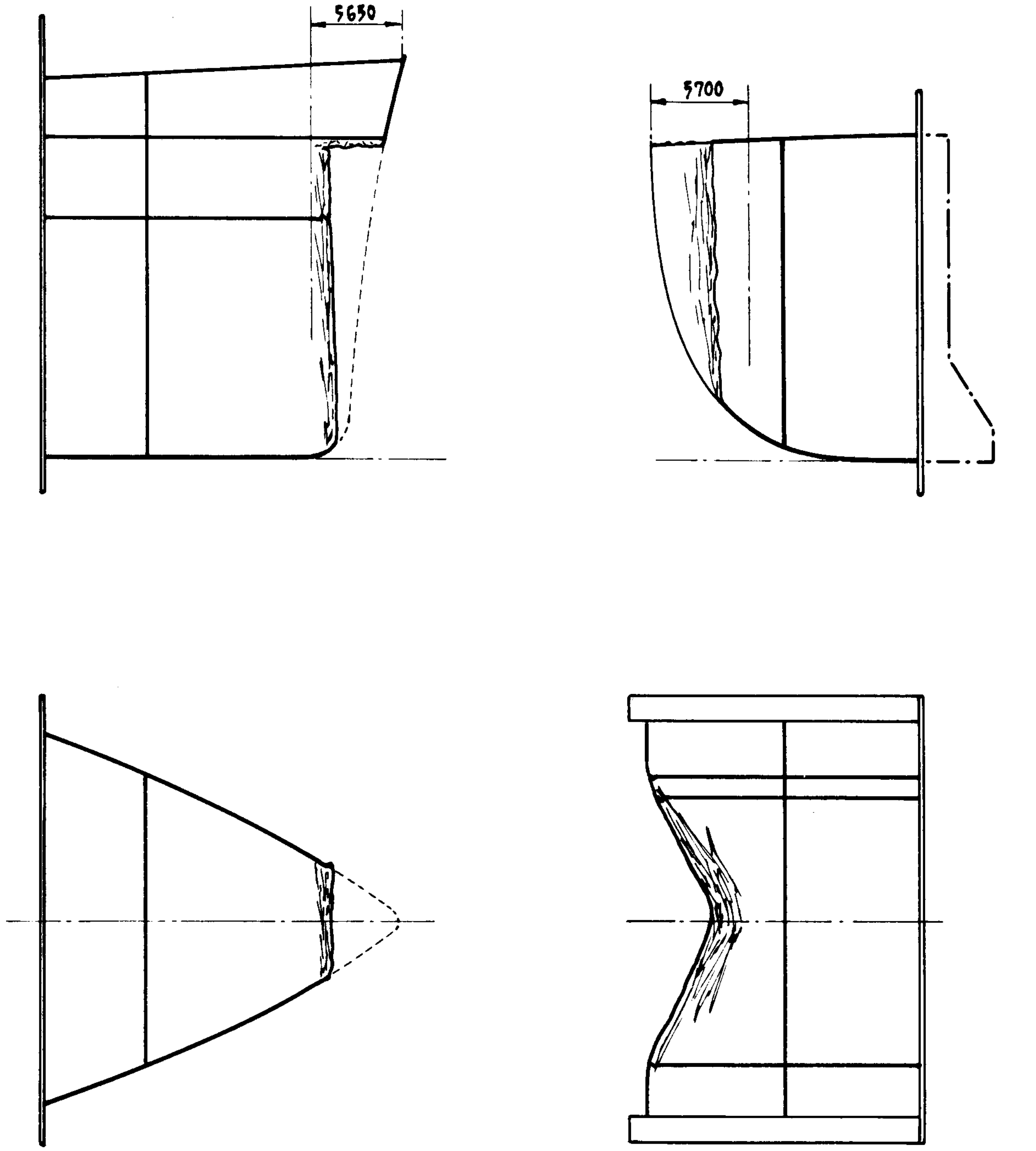
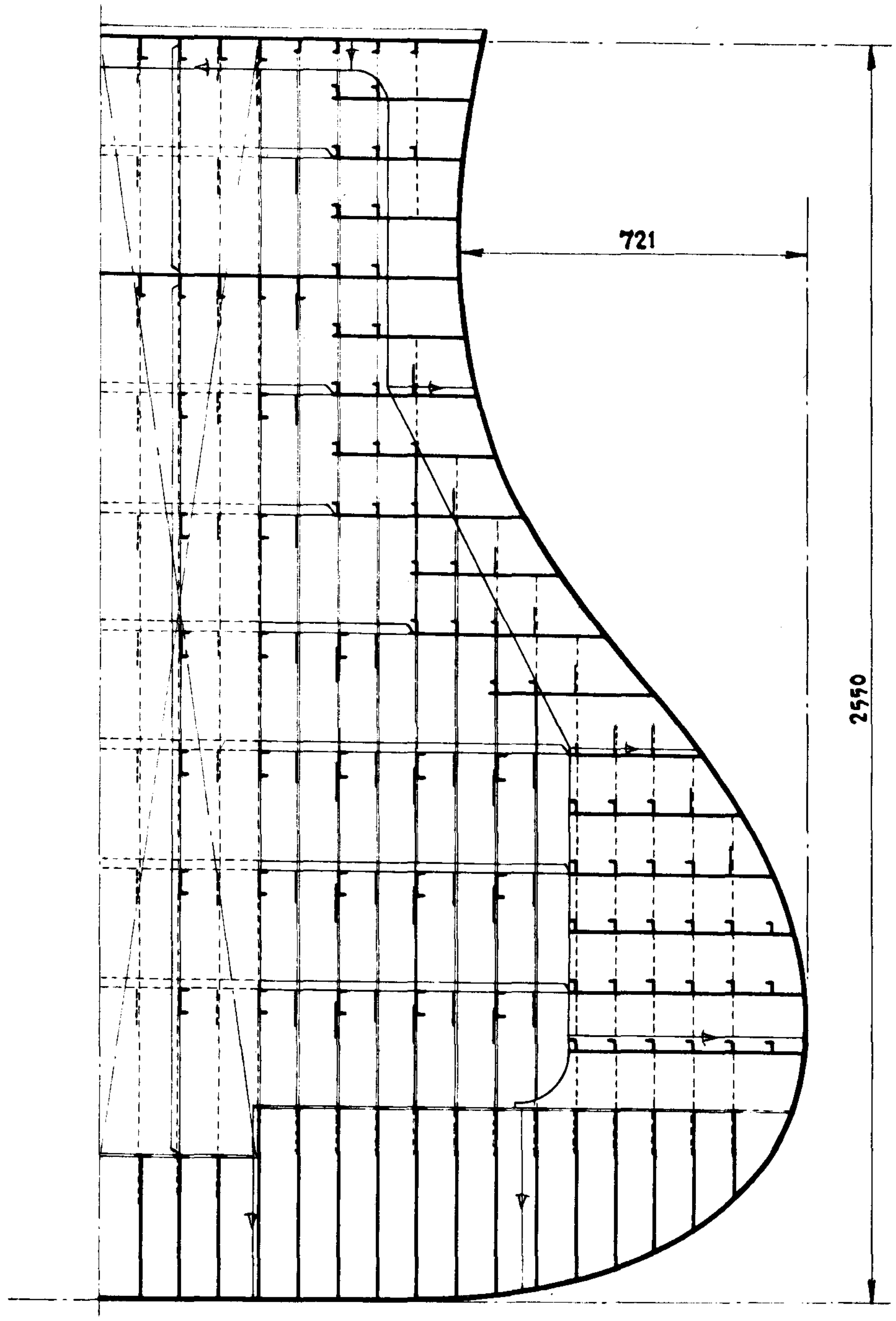


FIG. 33

BOW MODEL FOR 12TH, 13TH AND 14TH COLLISION TESTS



MARKED FIGURES ARE MODEL DIMENSIONS

FIG. 34

BOW AND SHIPSIDE MODELS BEFORE 12TH COLLISION TEST

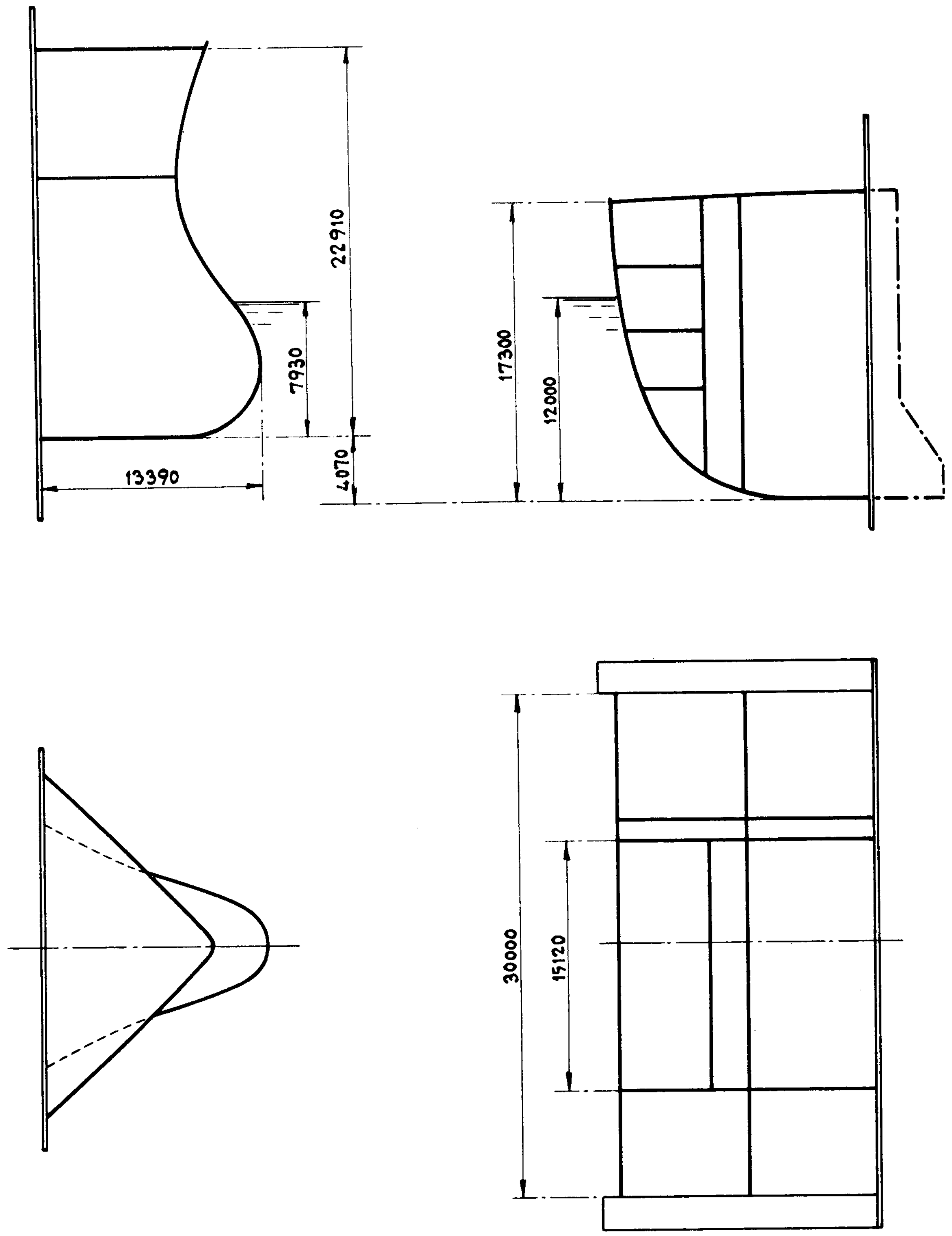


FIG. 35

BOW AND SHIPSIDE MODELS AFTER 12TH COLLISION TEST

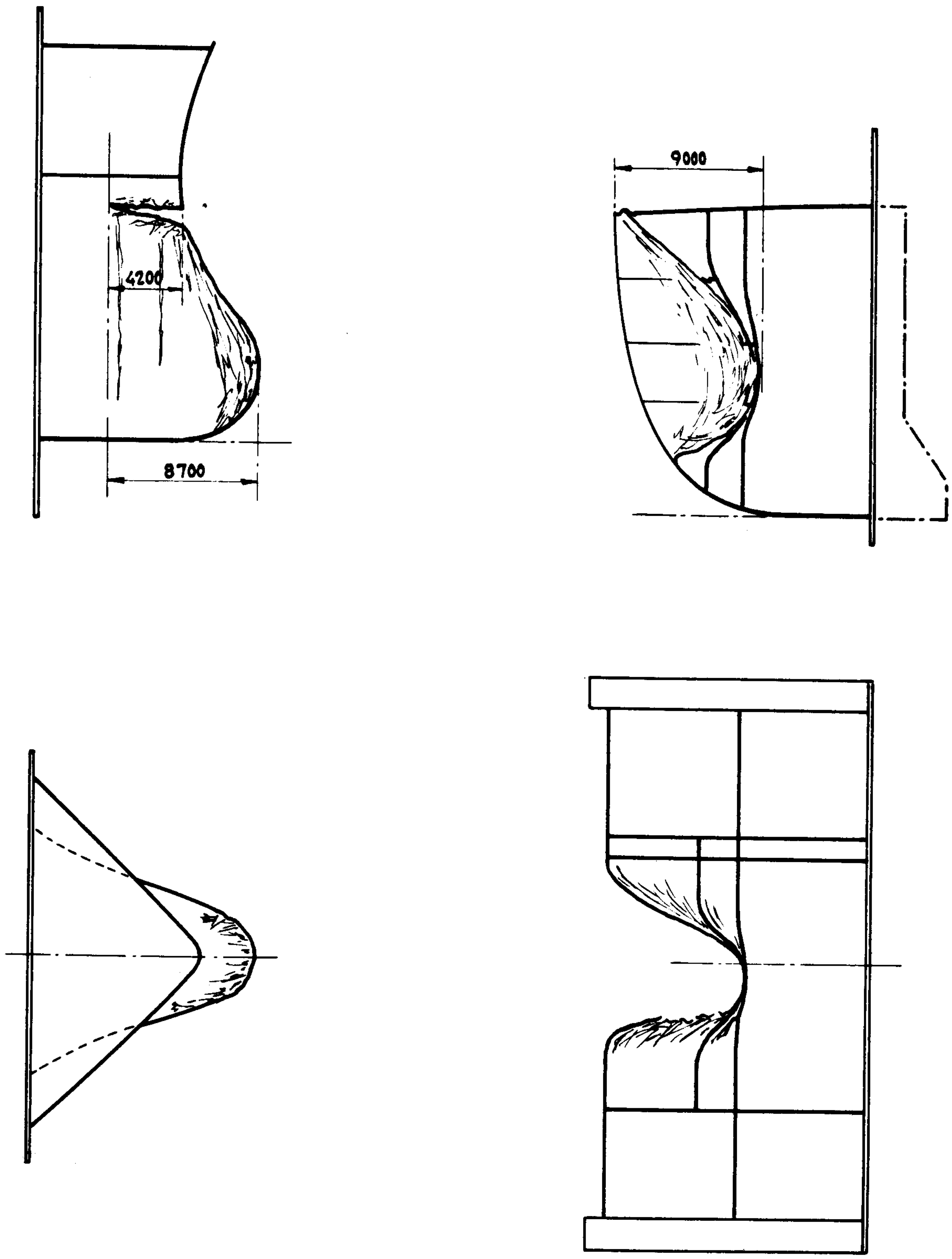


FIG. 36

BOW AND SHIPSIDE MODELS AFTER 13TH COLLISION TEST

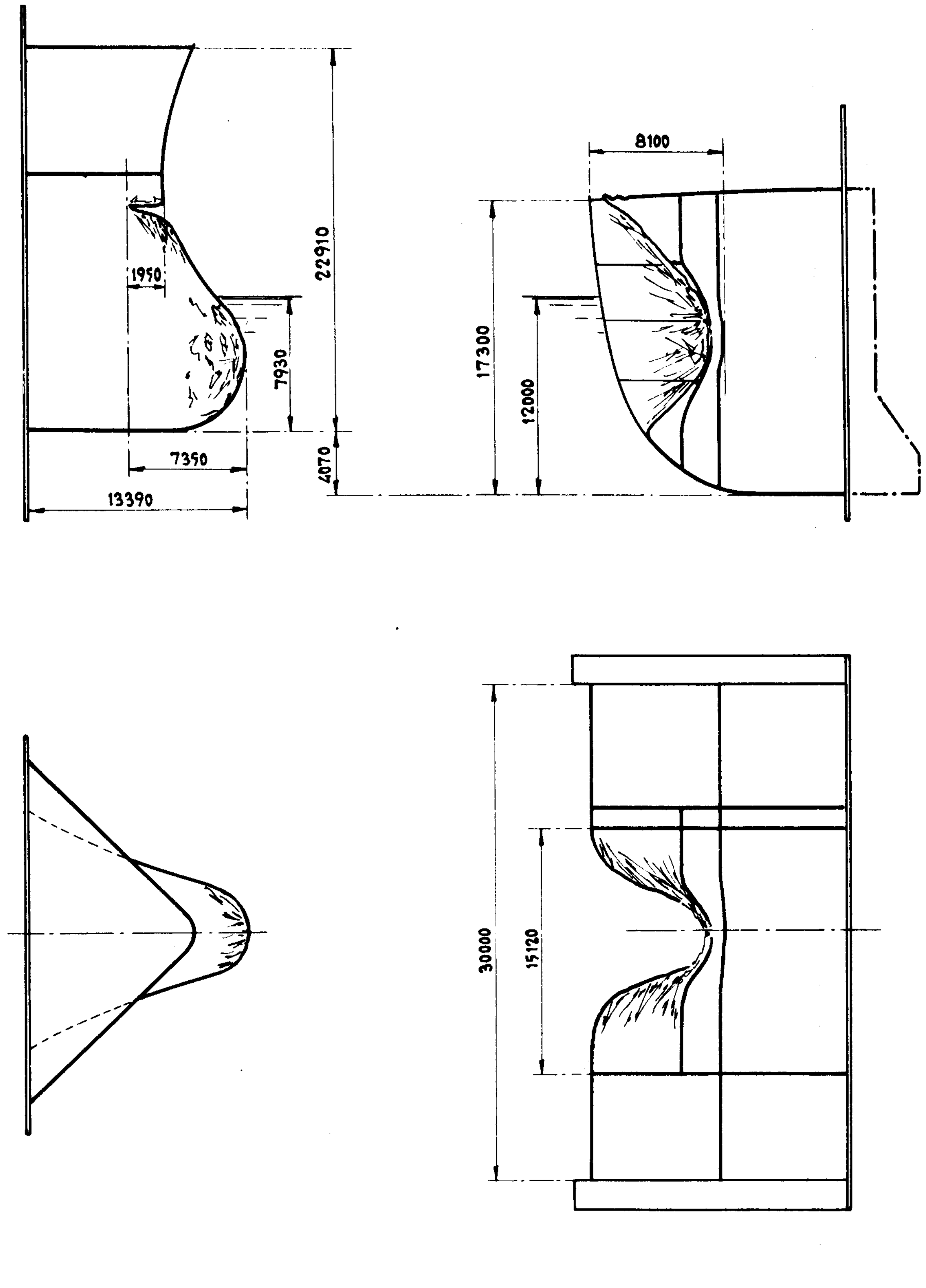


FIG. 37

BOW AND SHIPSIDE MODELS AFTER 14TH COLLISION TEST

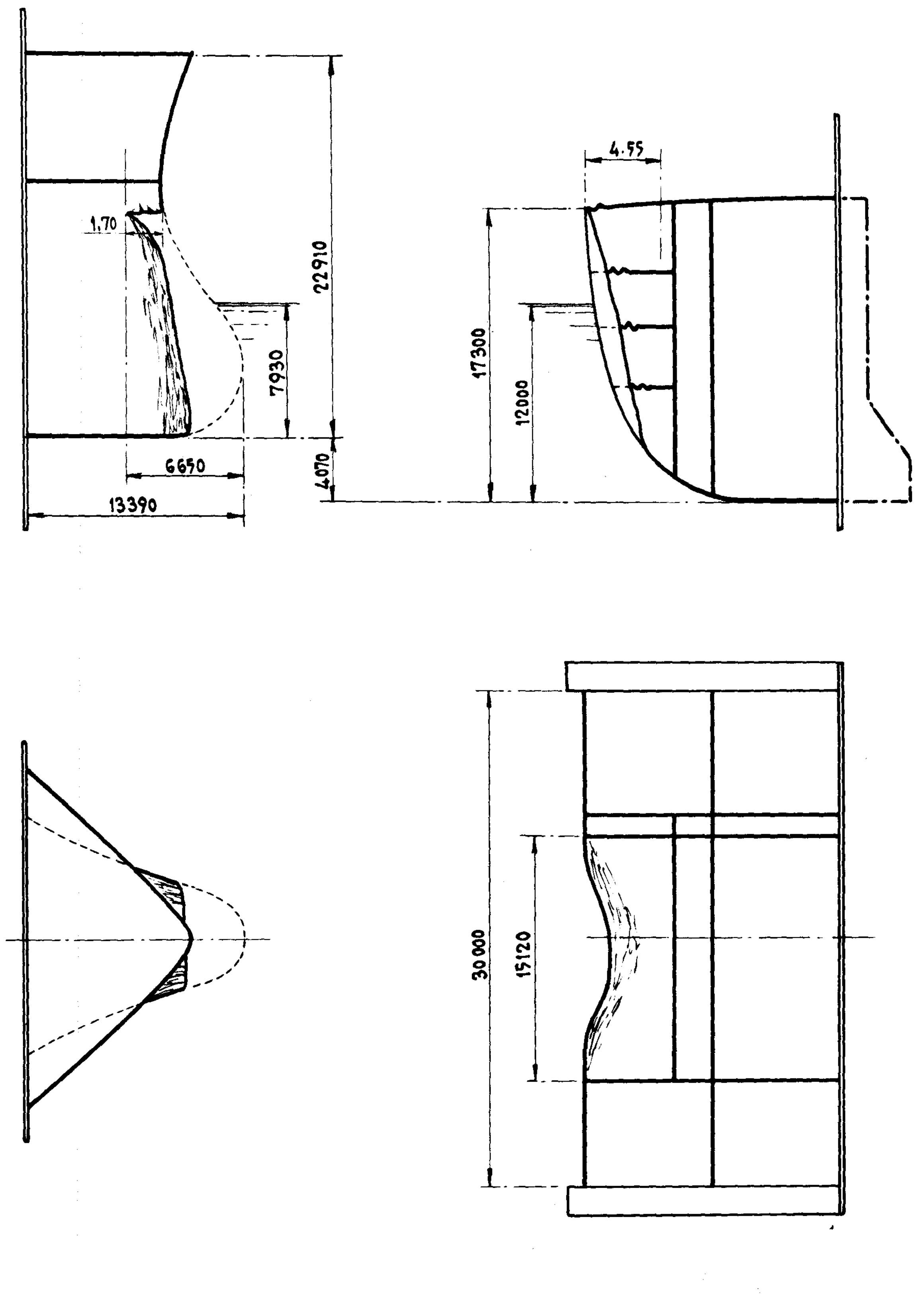


FIG. 38

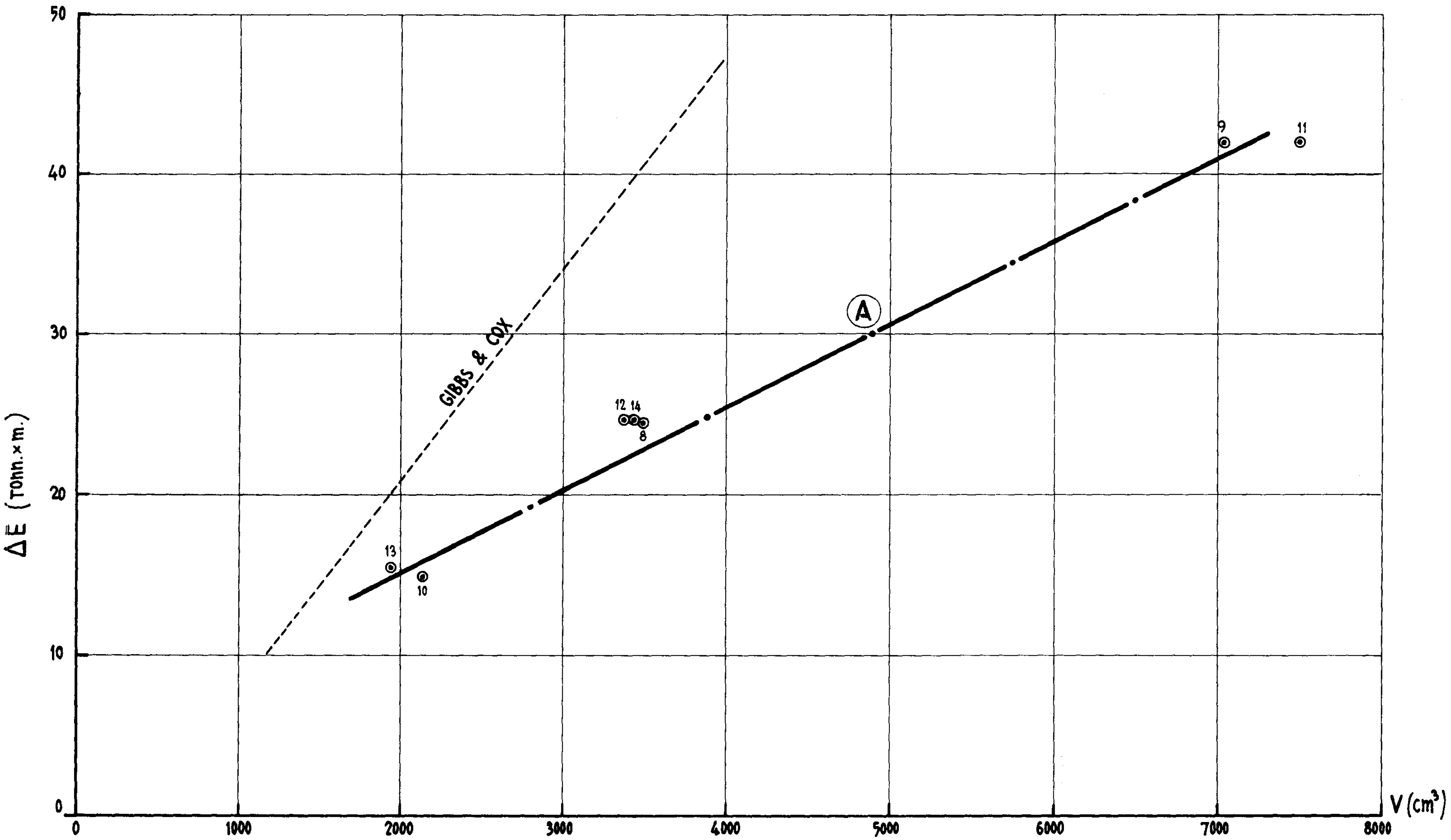


Fig. 39 RELATION BETWEEN VOLUMES OF MATERIAL AFFECTED BY THE COLLISION AND DISSIPATED ENERGY
Total volume of material

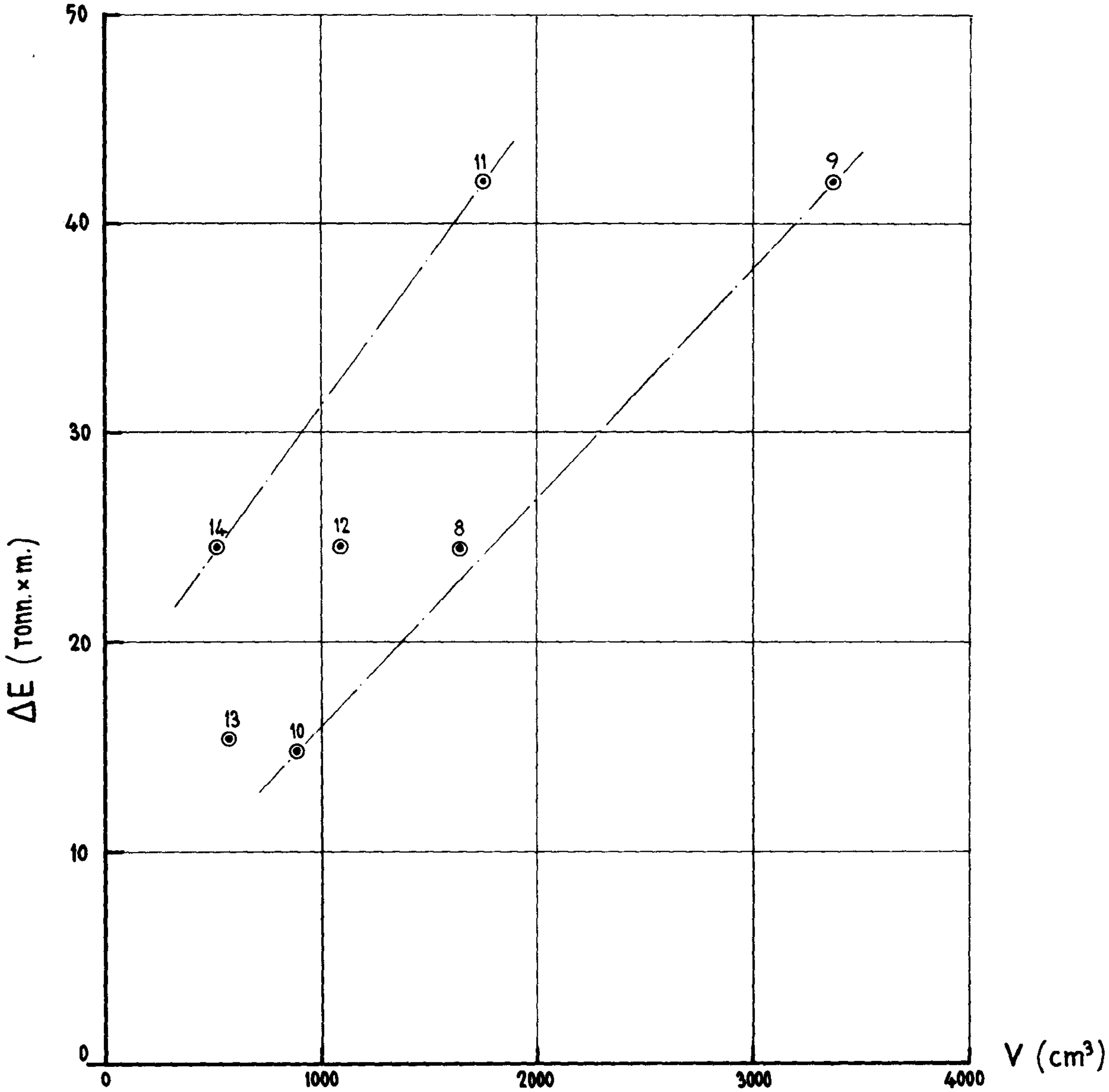


Fig. 40 RELATION BETWEEN VOLUMES OF MATERIAL AFFECTED BY THE COLLISION AND DISSIPATED ENERGY
Volumes according to Minorsky procedure

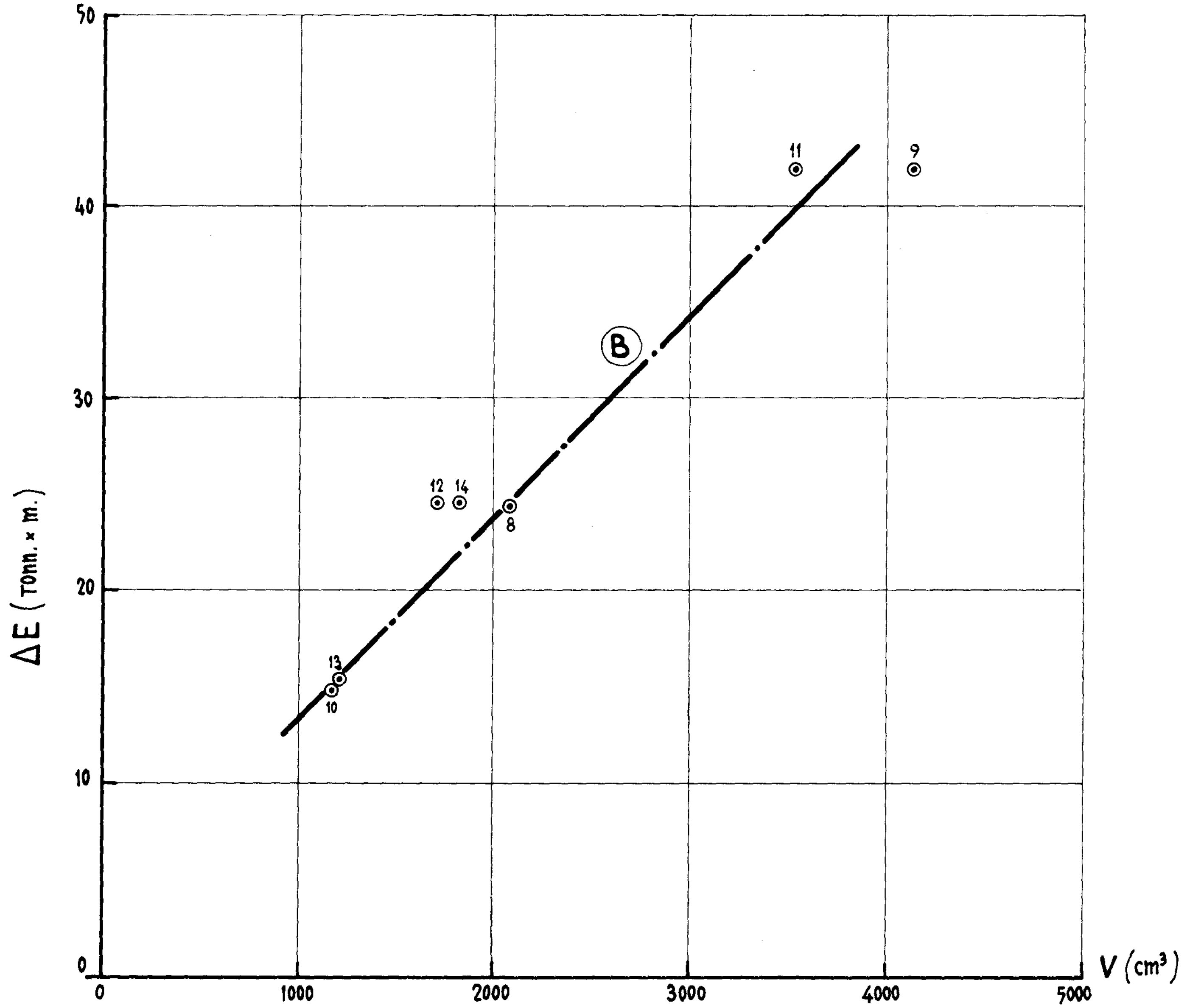


Fig. 41 RELATION BETWEEN VOLUMES OF MATERIAL AFFECTED BY THE COLLISION AND DISSIPATED ENERGY
Excluding plating and stiffeners normal to line of impact

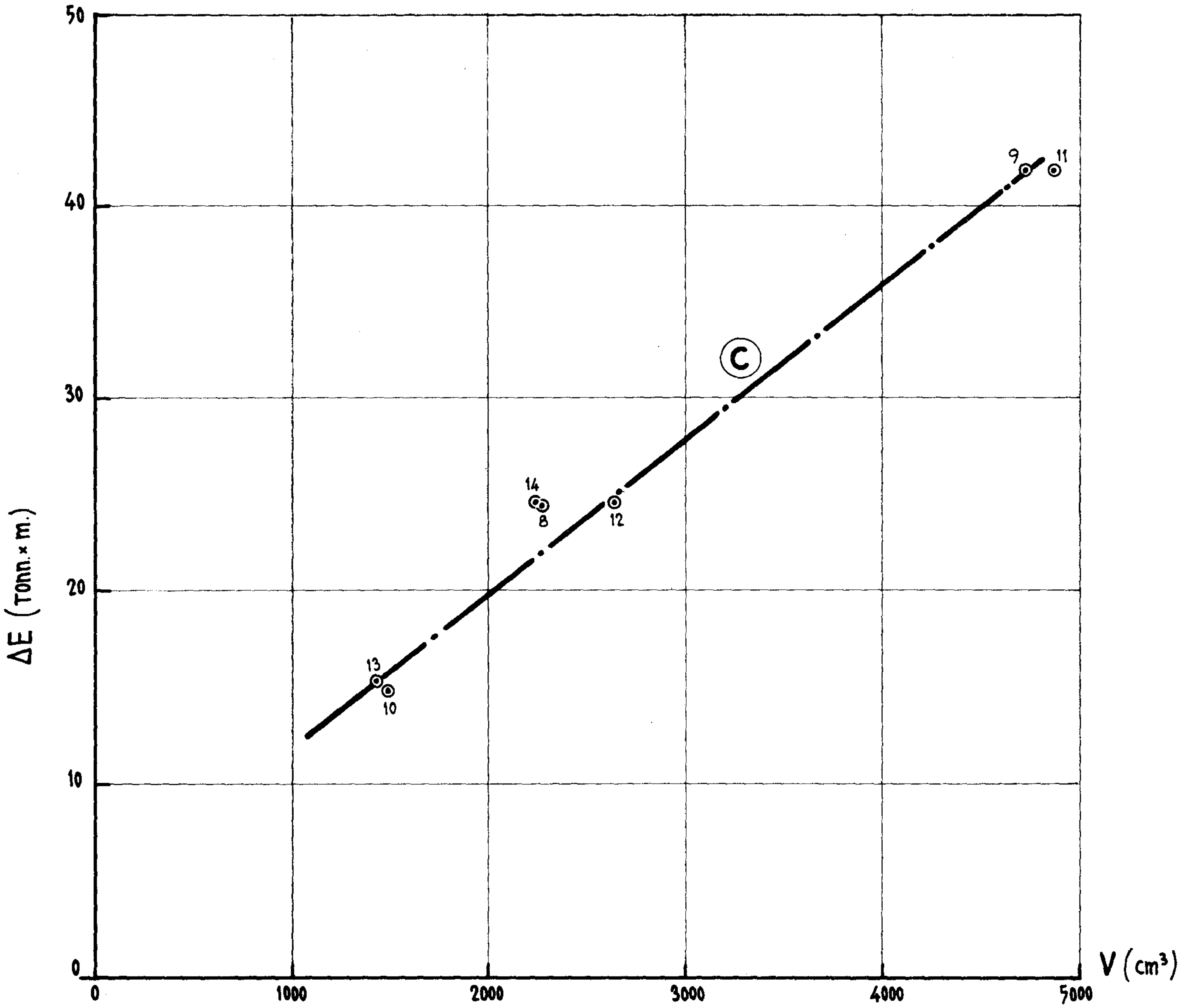
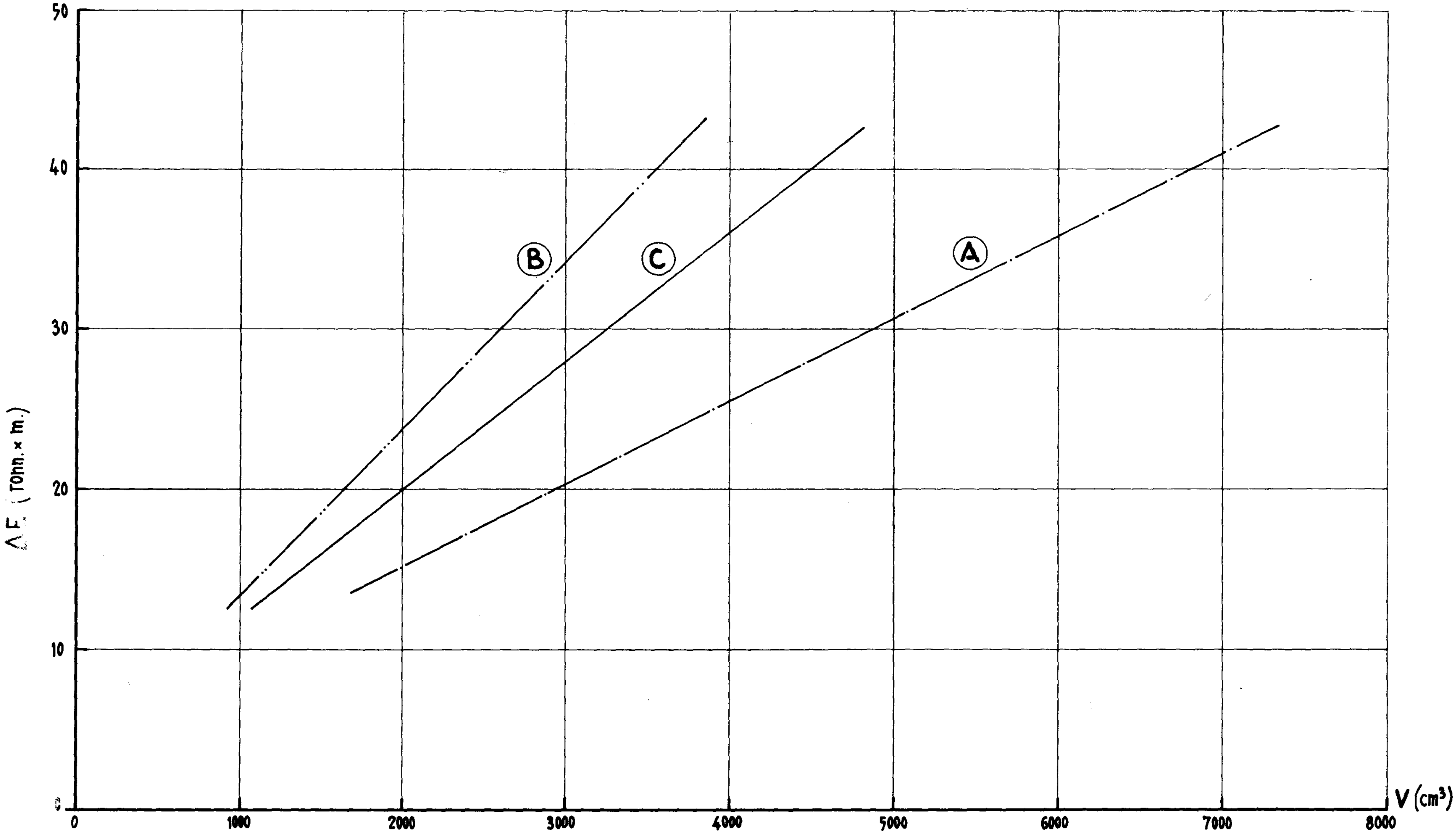


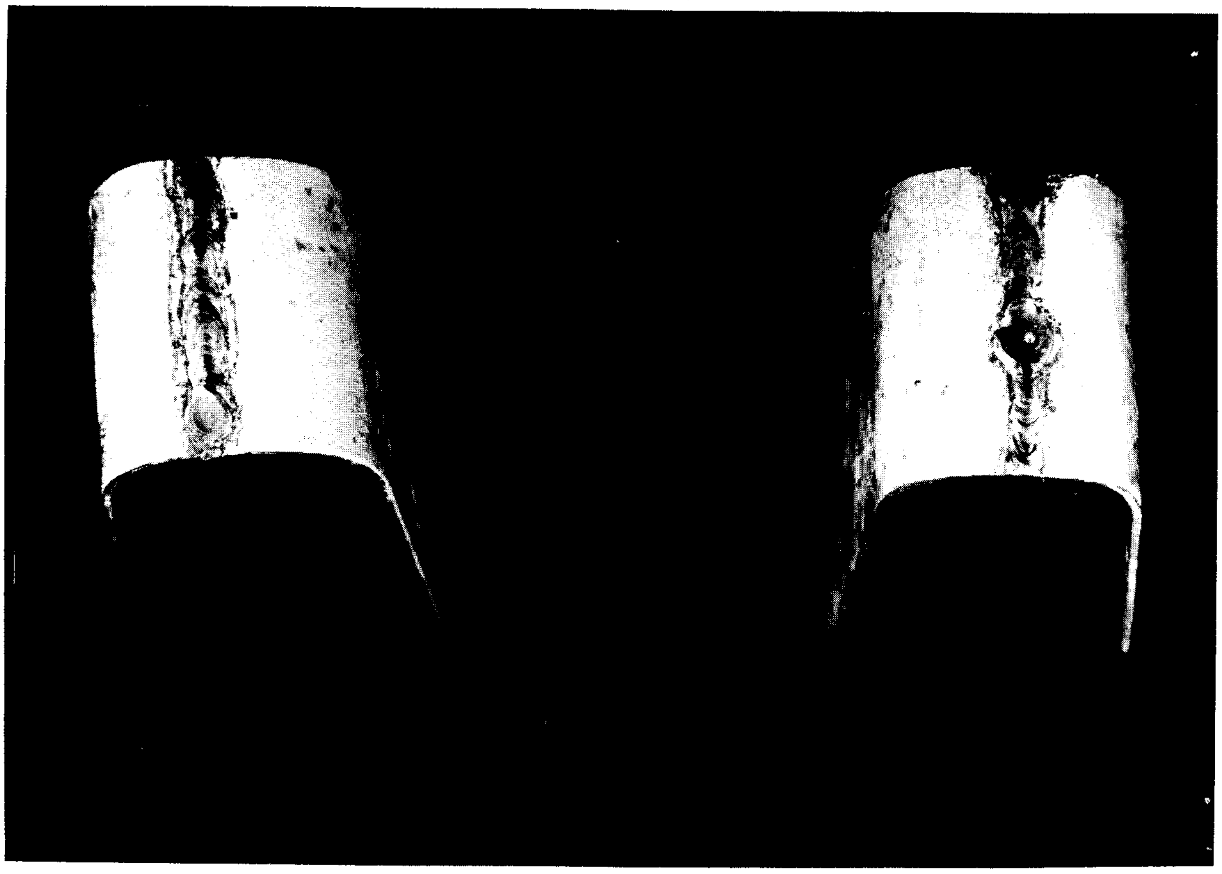
Fig. **42** RELATION BETWEEN VOLUMES OF MATERIAL AFFECTED BY THE COLLISION AND DISSIPATED ENERGY
100% volume of destroyed material, plus 65% volume of deformed material



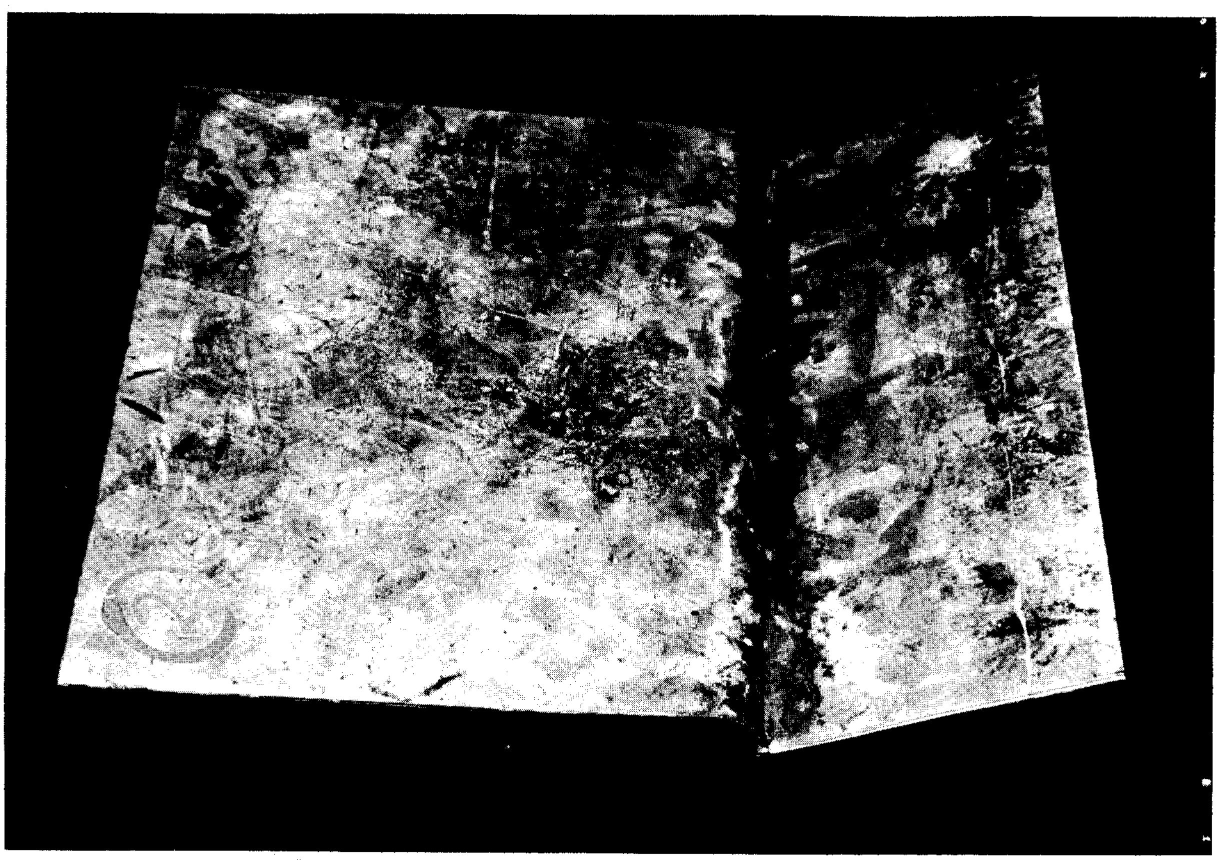
43 RELATION BETWEEN VOLUMES OF MATERIAL AFFECTED BY THE COLLISION AND DISSIPATED ENERGY

Comparis on of A - B - C procedures

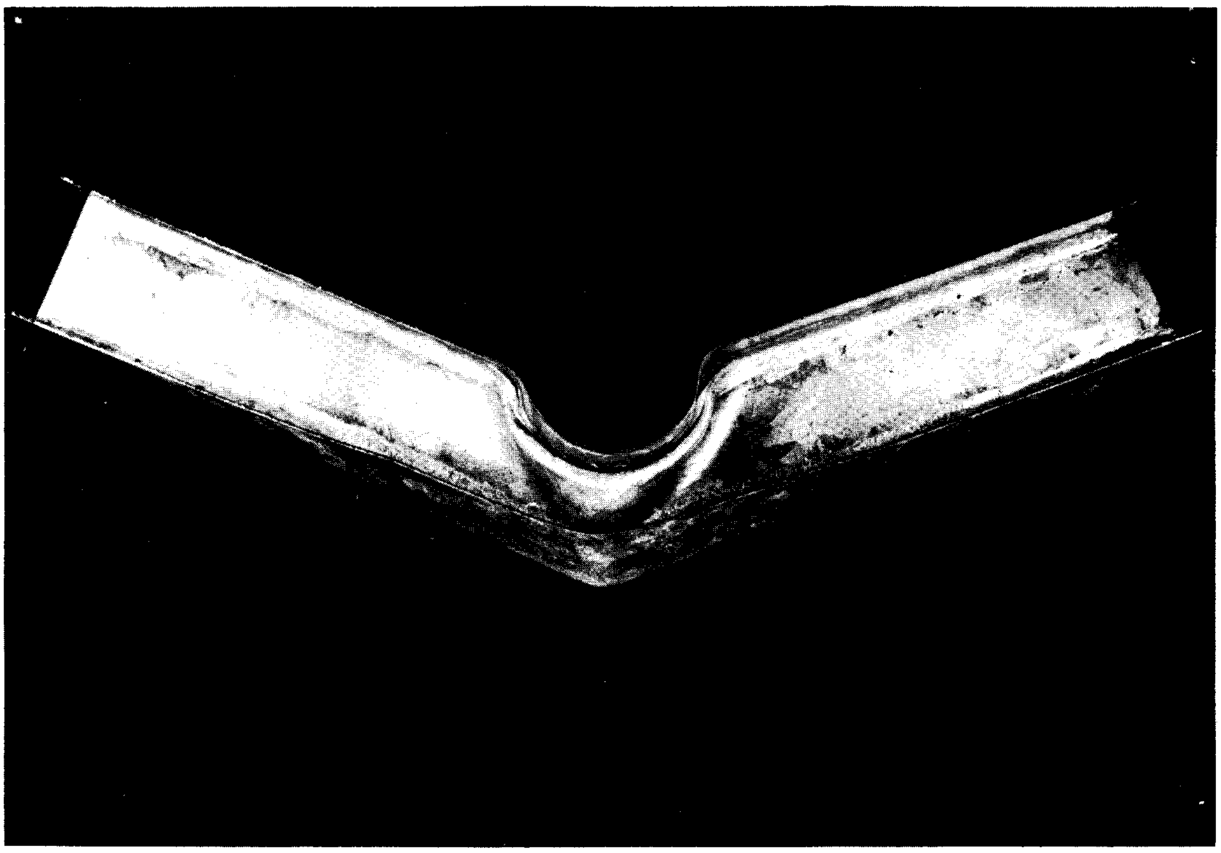
PHOTOGRAPHS



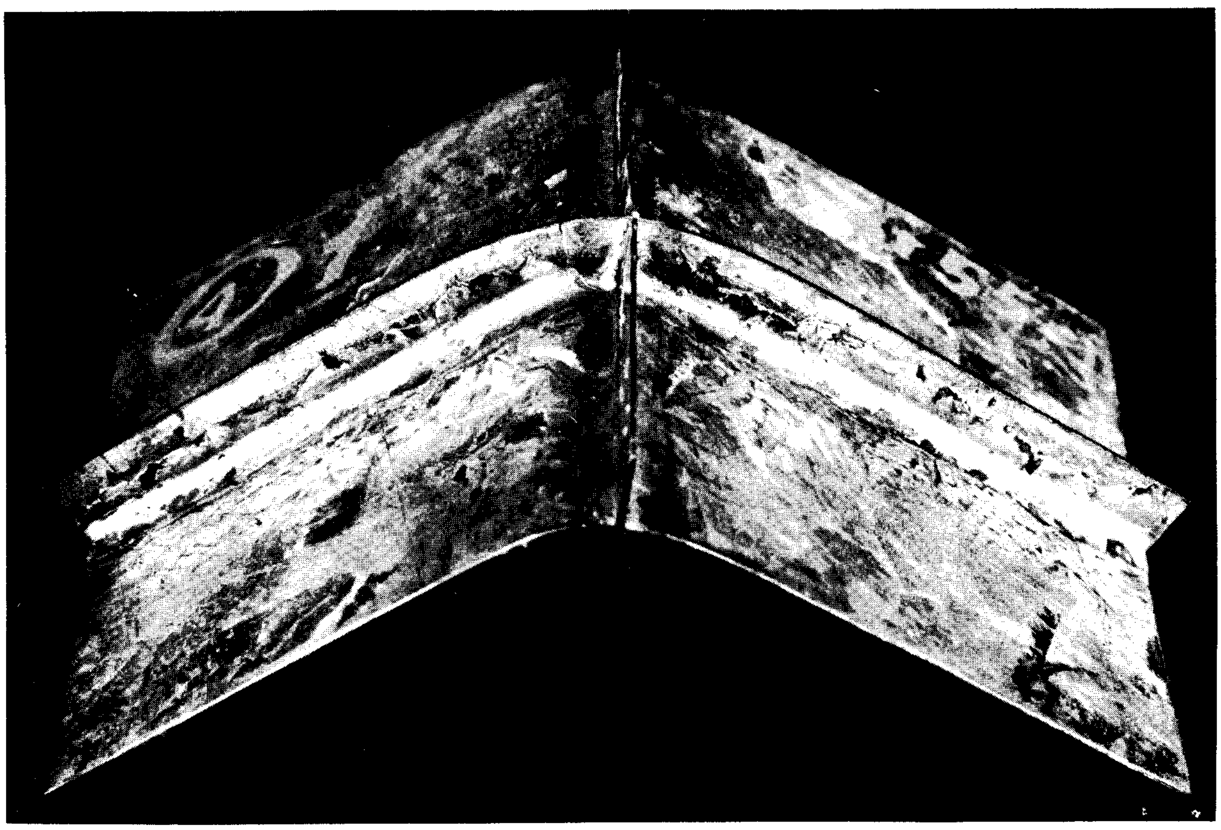
1. 180° bending test of an electrically welded specimen



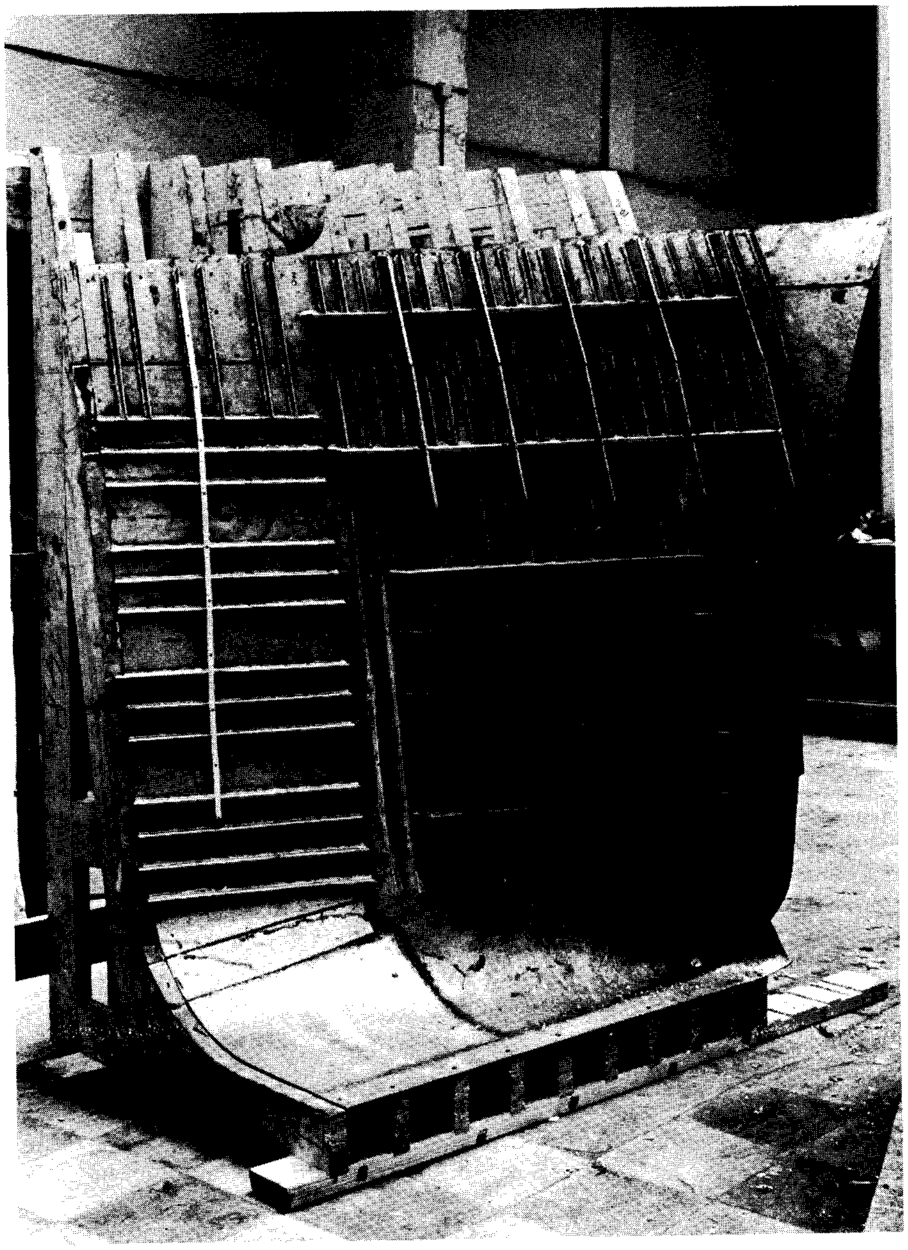
2. 180° bending test of a tin soldered T specimen



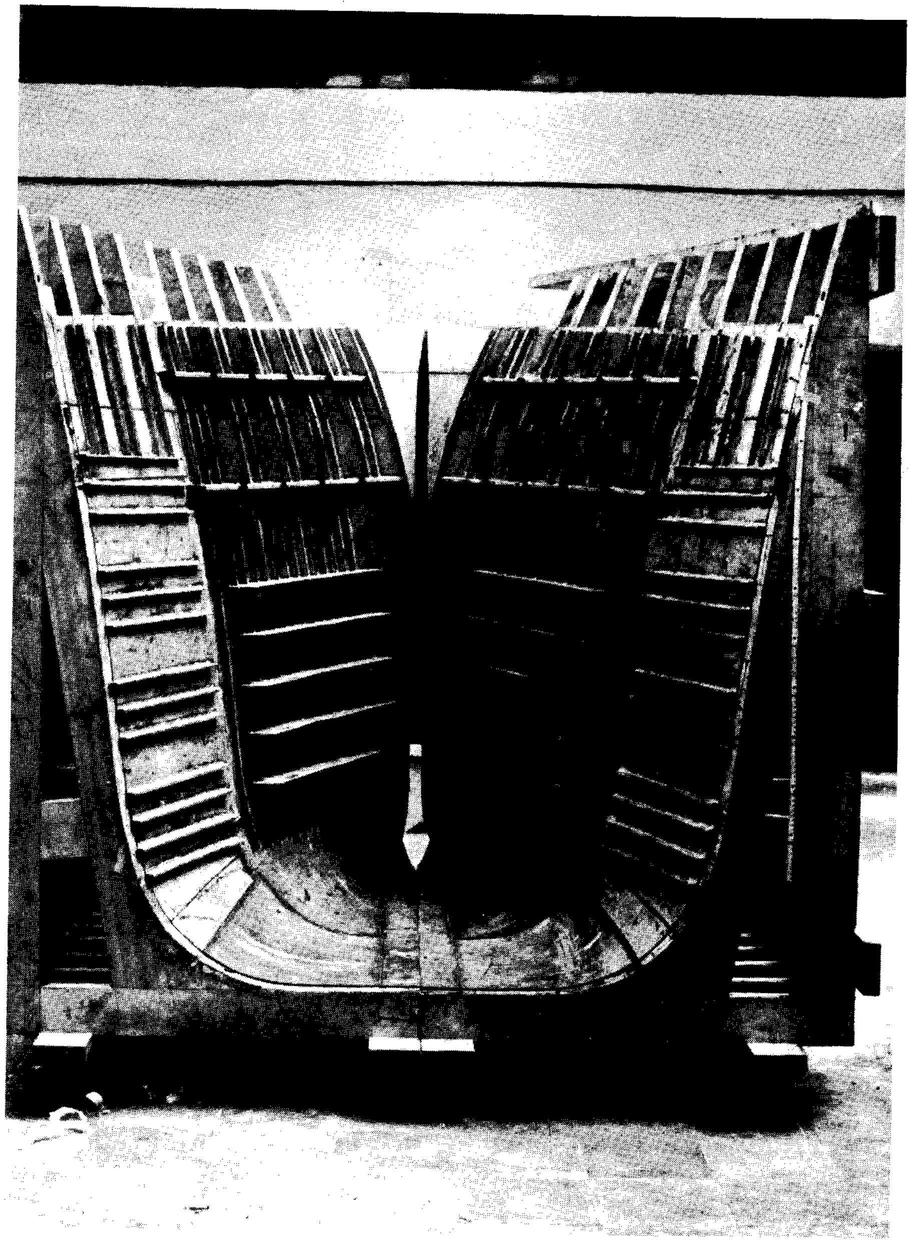
3. Bending test of a tin soldered H specimen



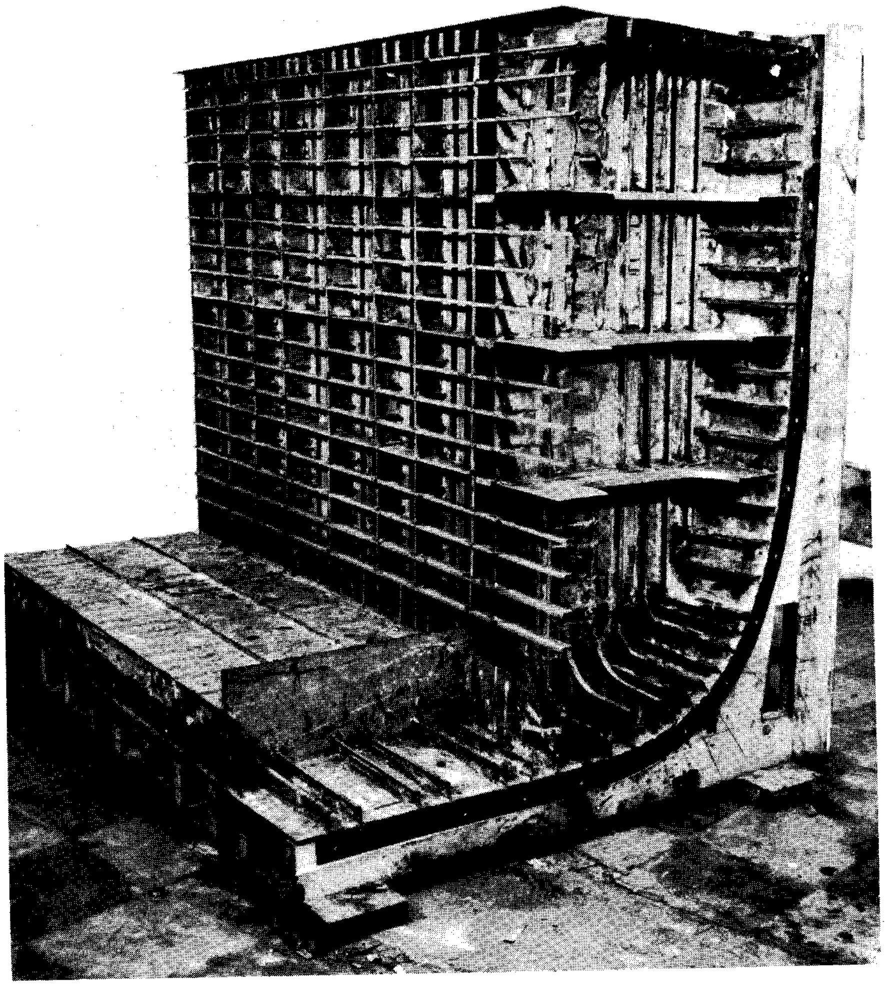
4. Bending test of a tin soldered cross shaped specimen



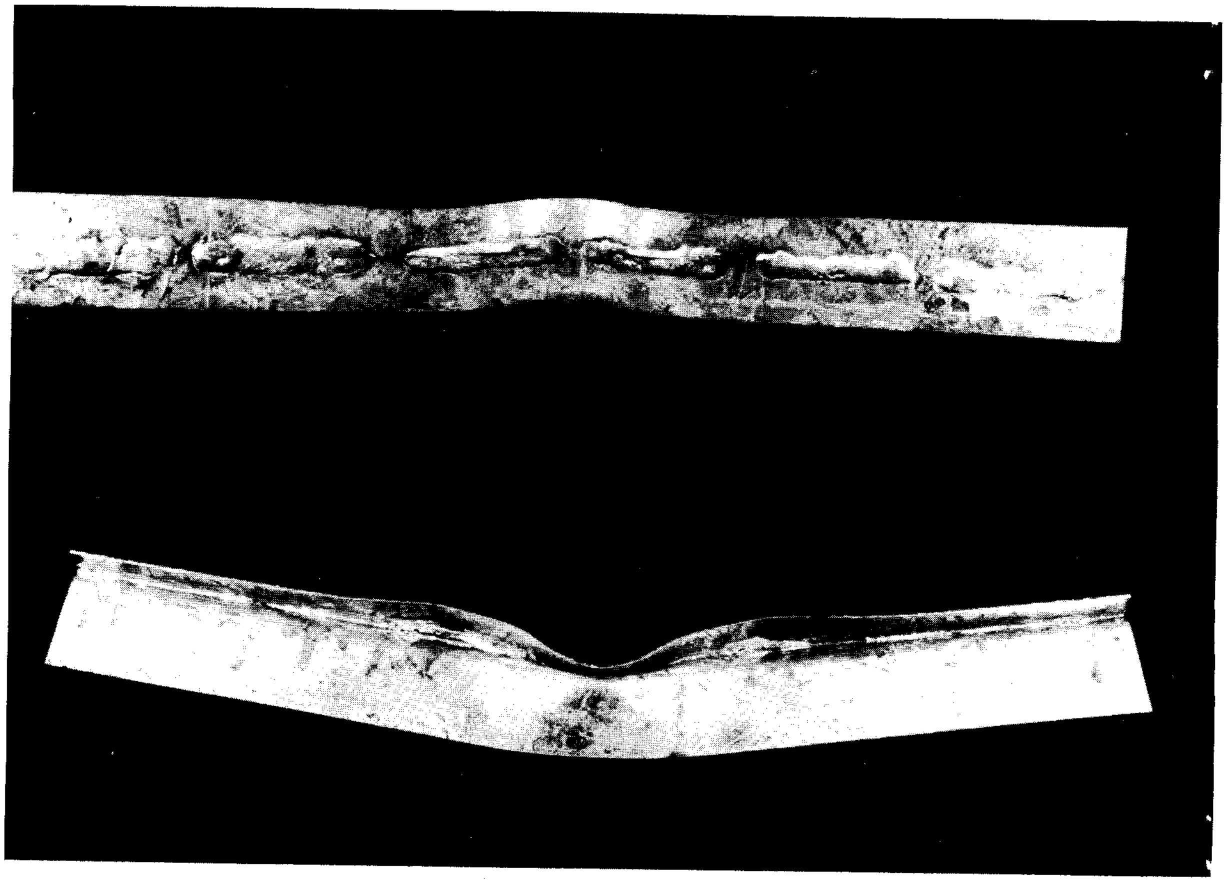
5. Wooden moulding frame



6. Wooden moulding frame

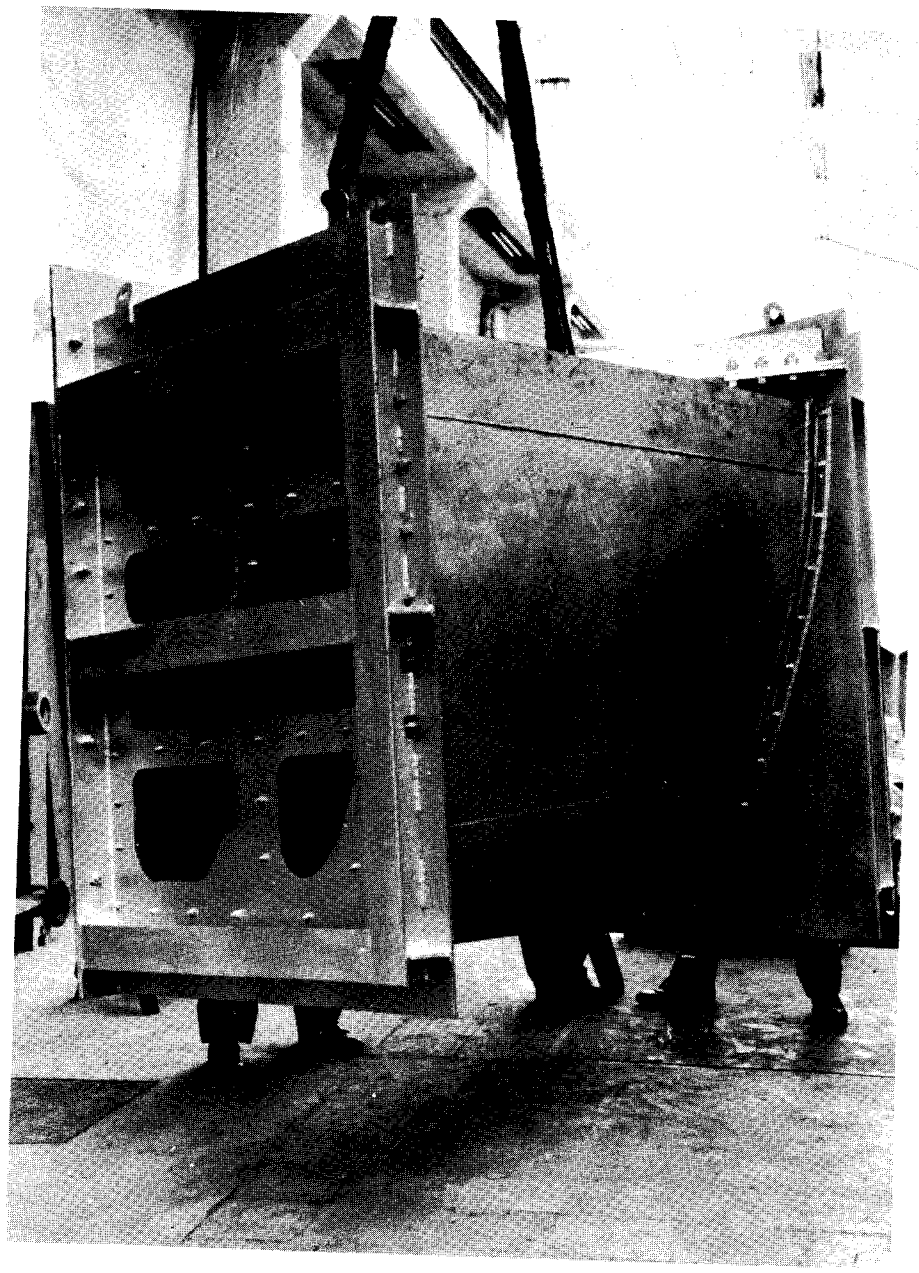


7. Wooden moulding frame

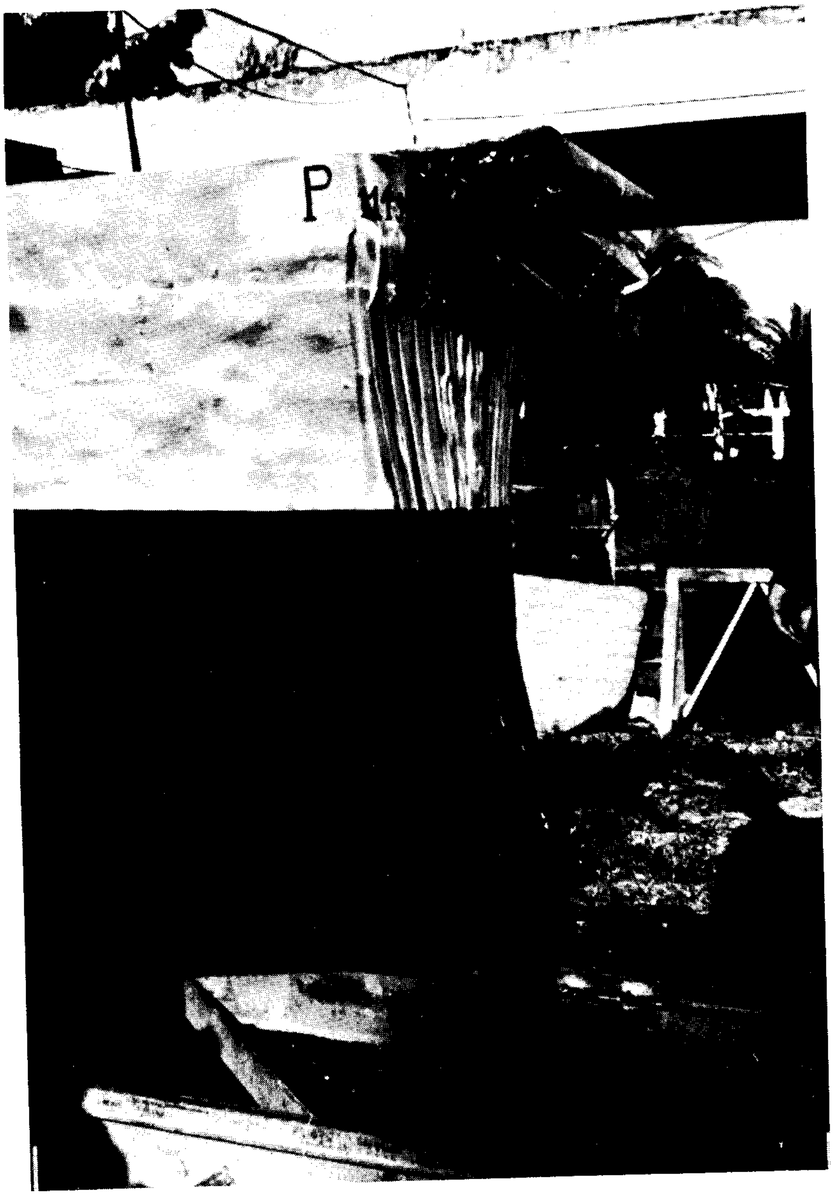


8. Bending test of a slot tin soldered specimen

9. Bow model for collision test No. 1

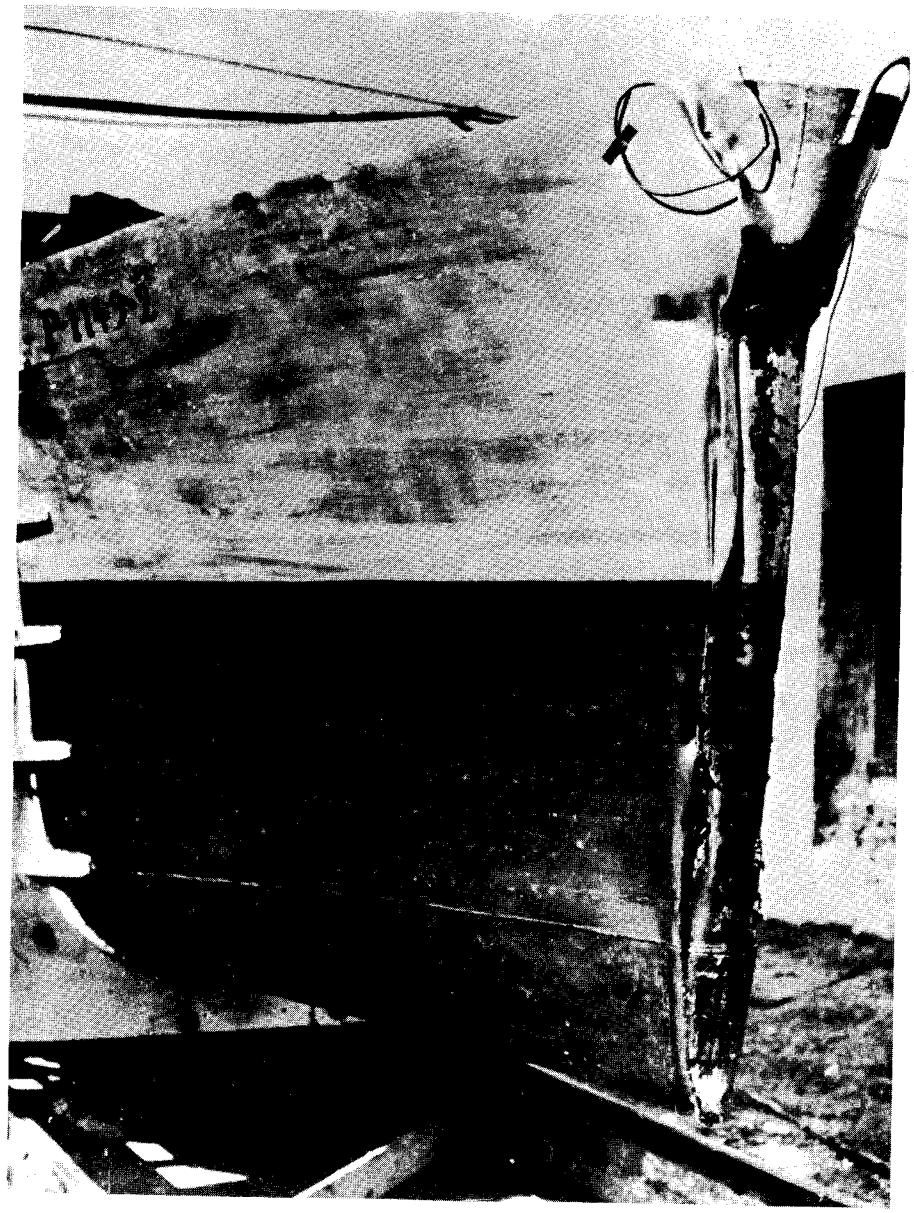


10. Ship side model for collision test No. 1

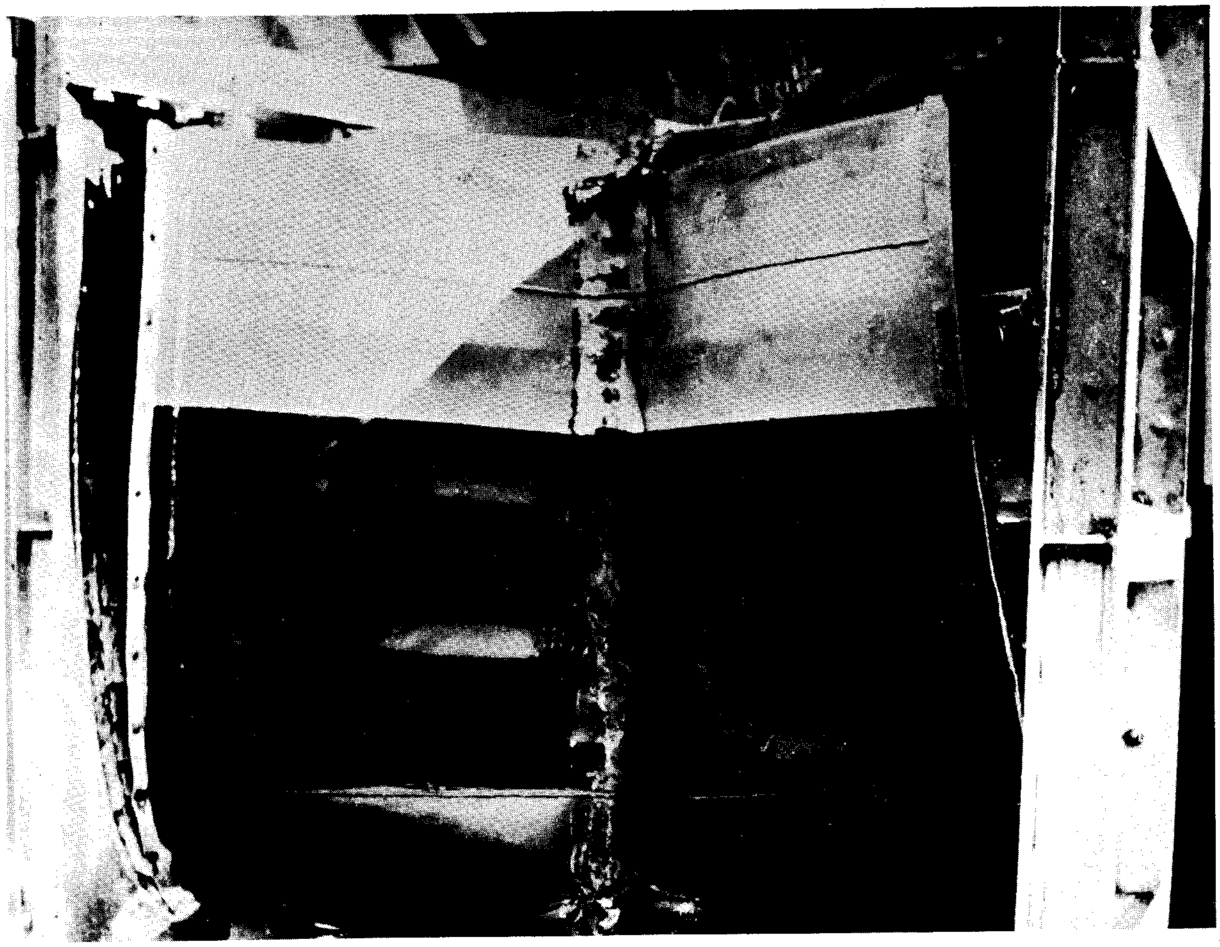


11. Bow model after collision test No. 1

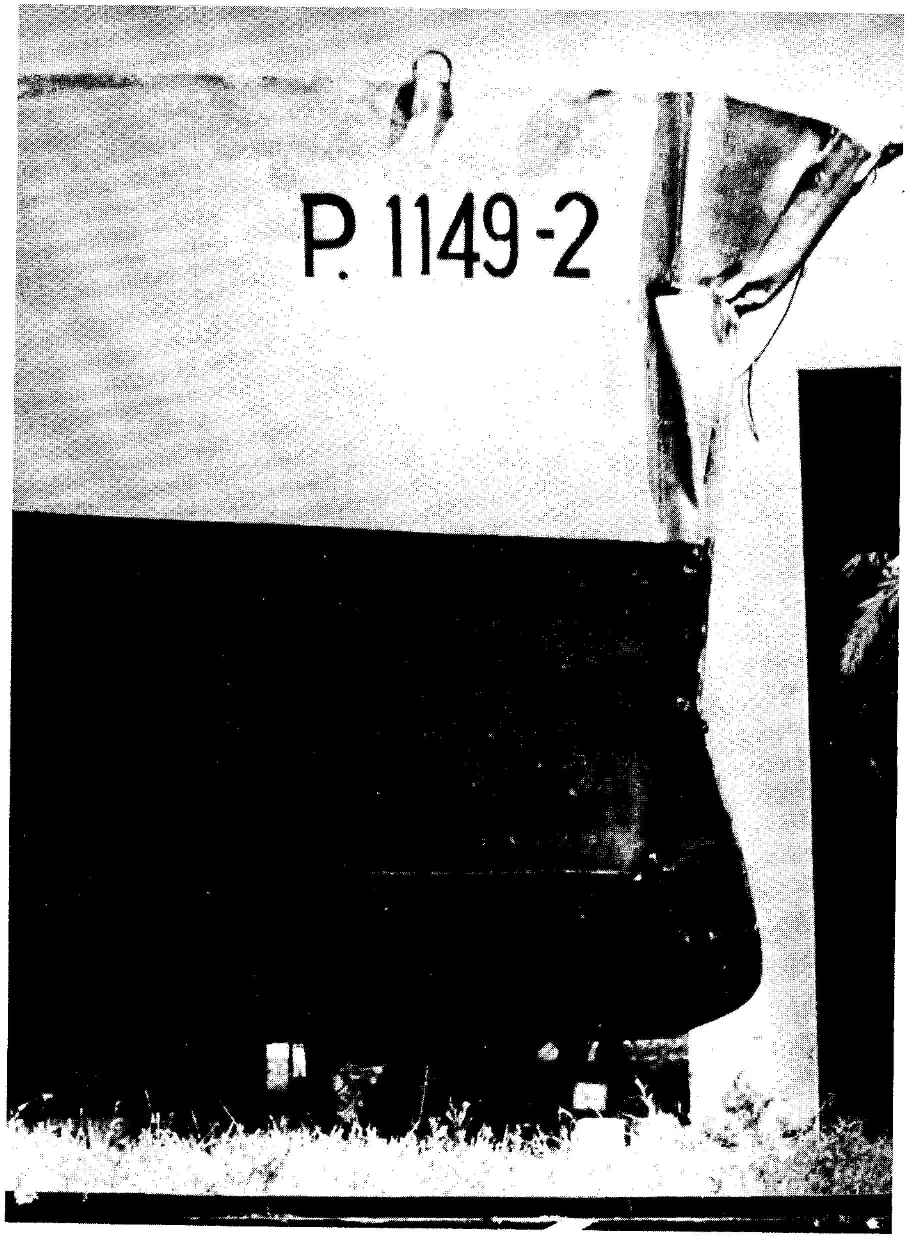
12. Ship side model after collision test No. 1



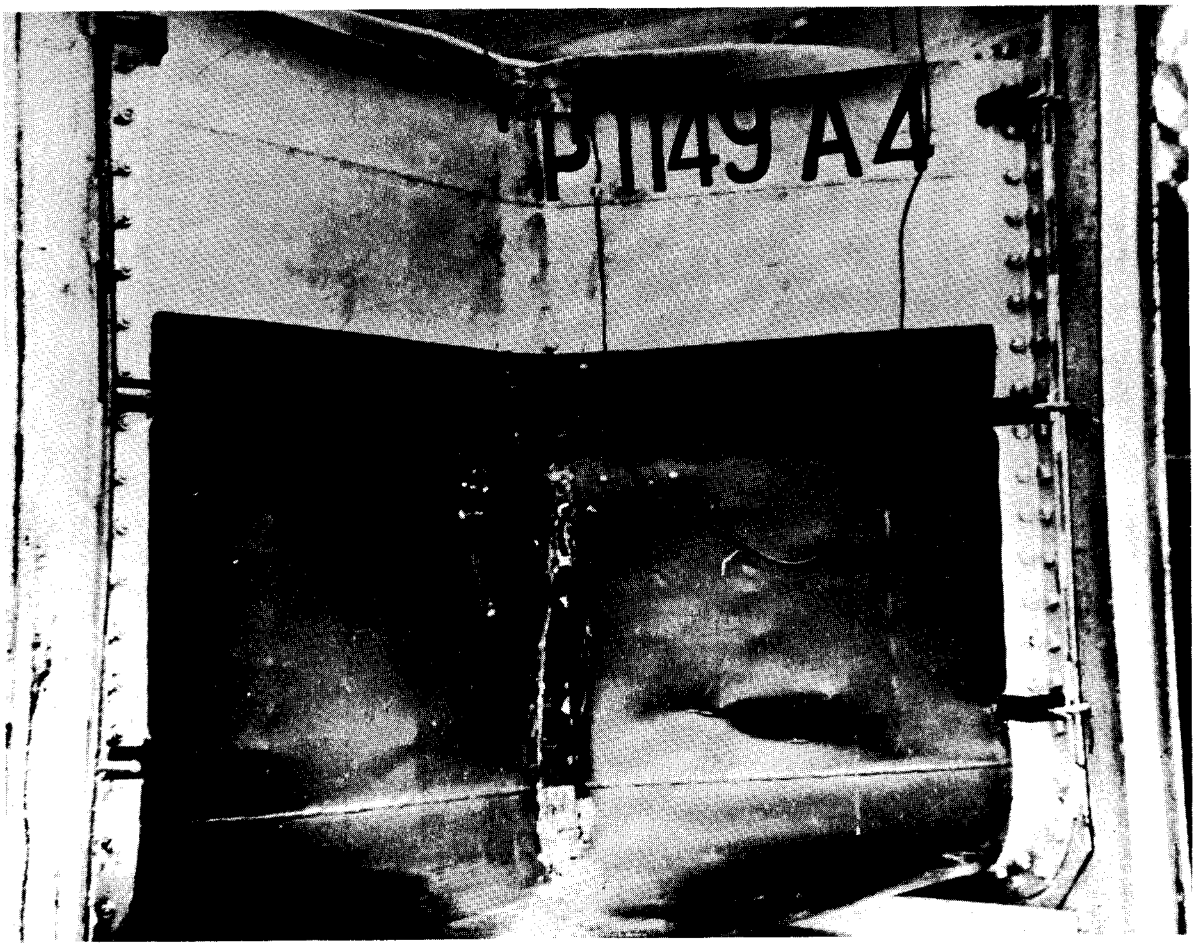
13. Bow model after collision test No. 2



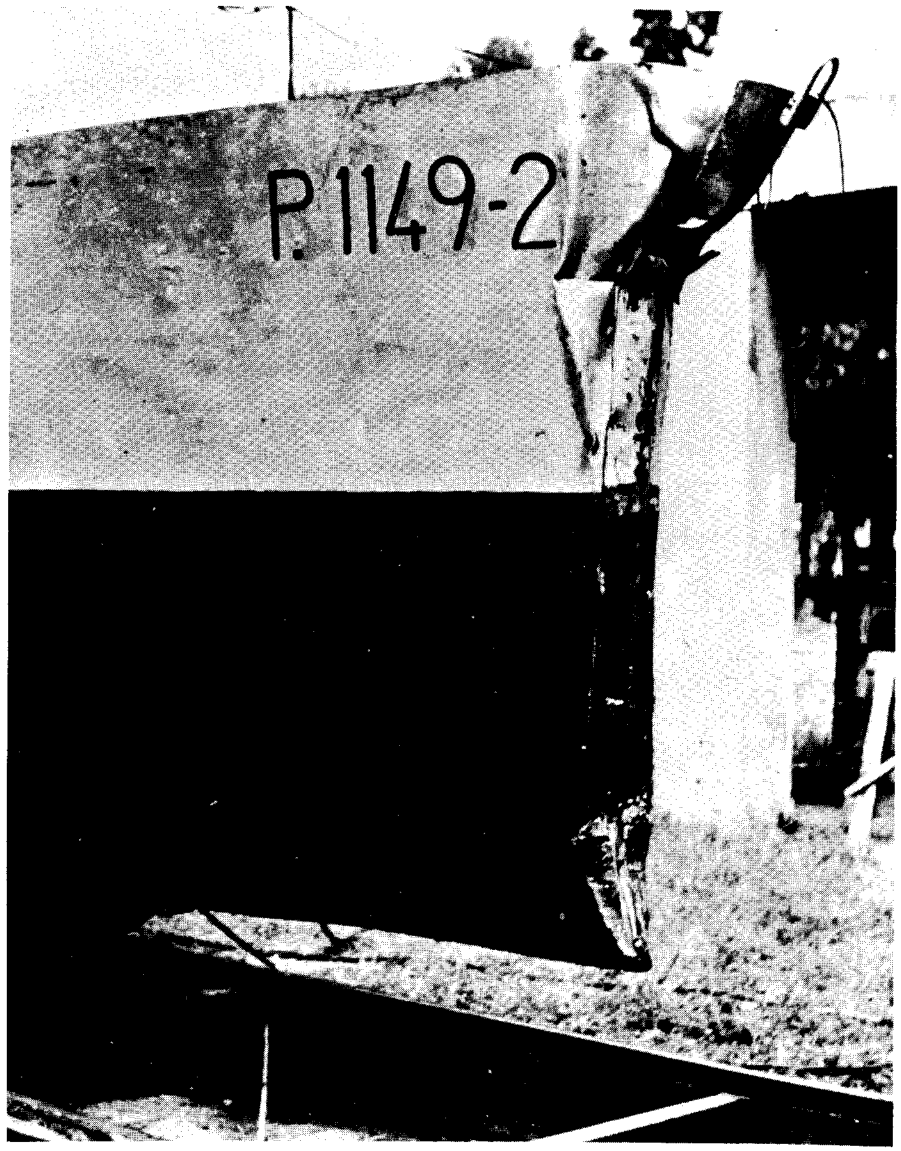
14. Ship side model after collision test No. 2



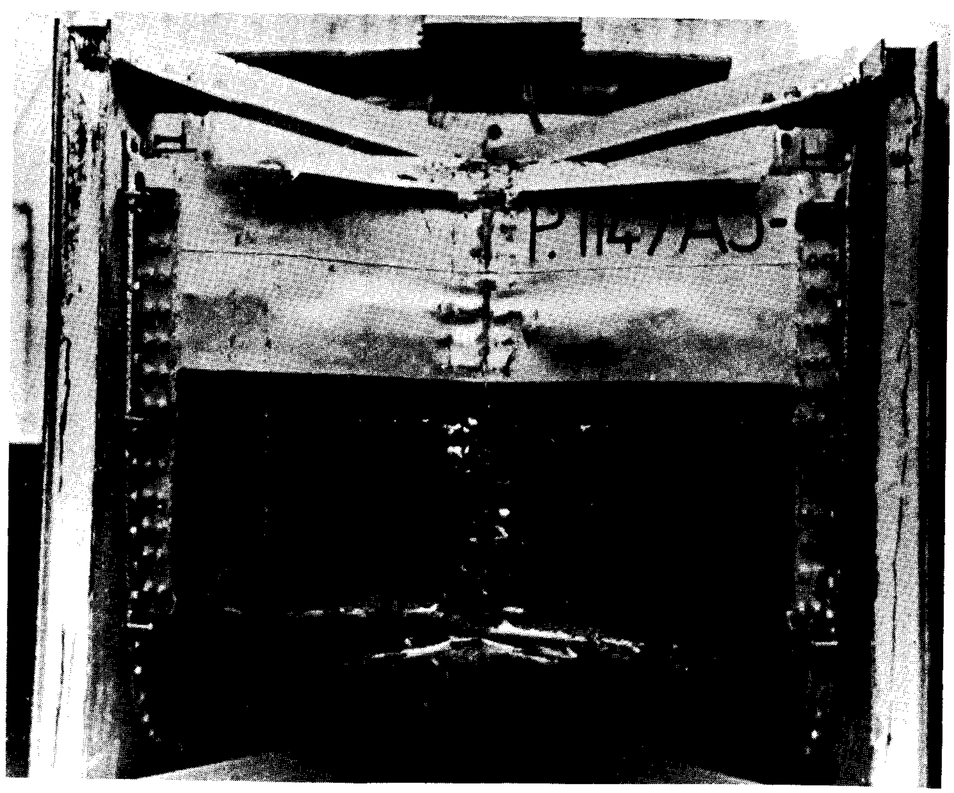
15. Bow model after collision test No. 3



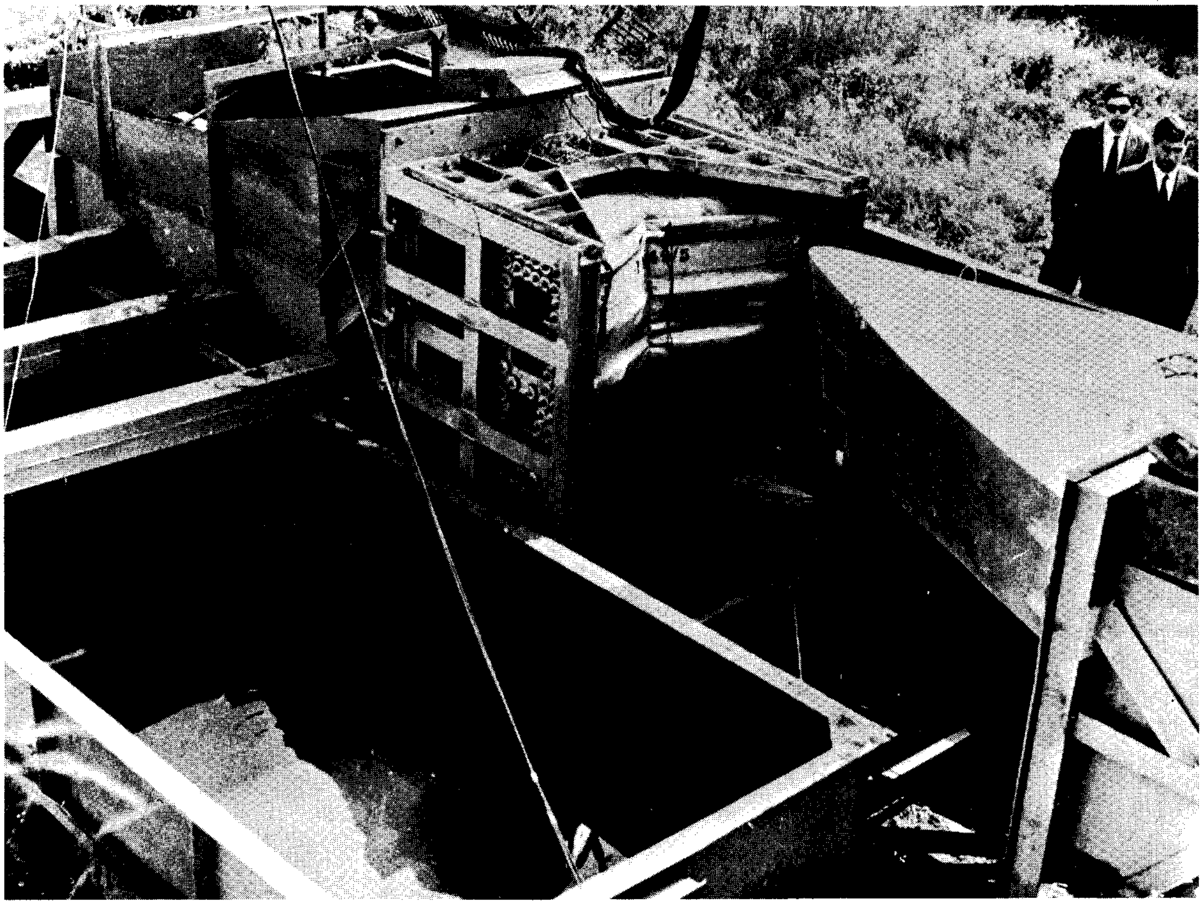
16. Ship side model after collision test No. 3



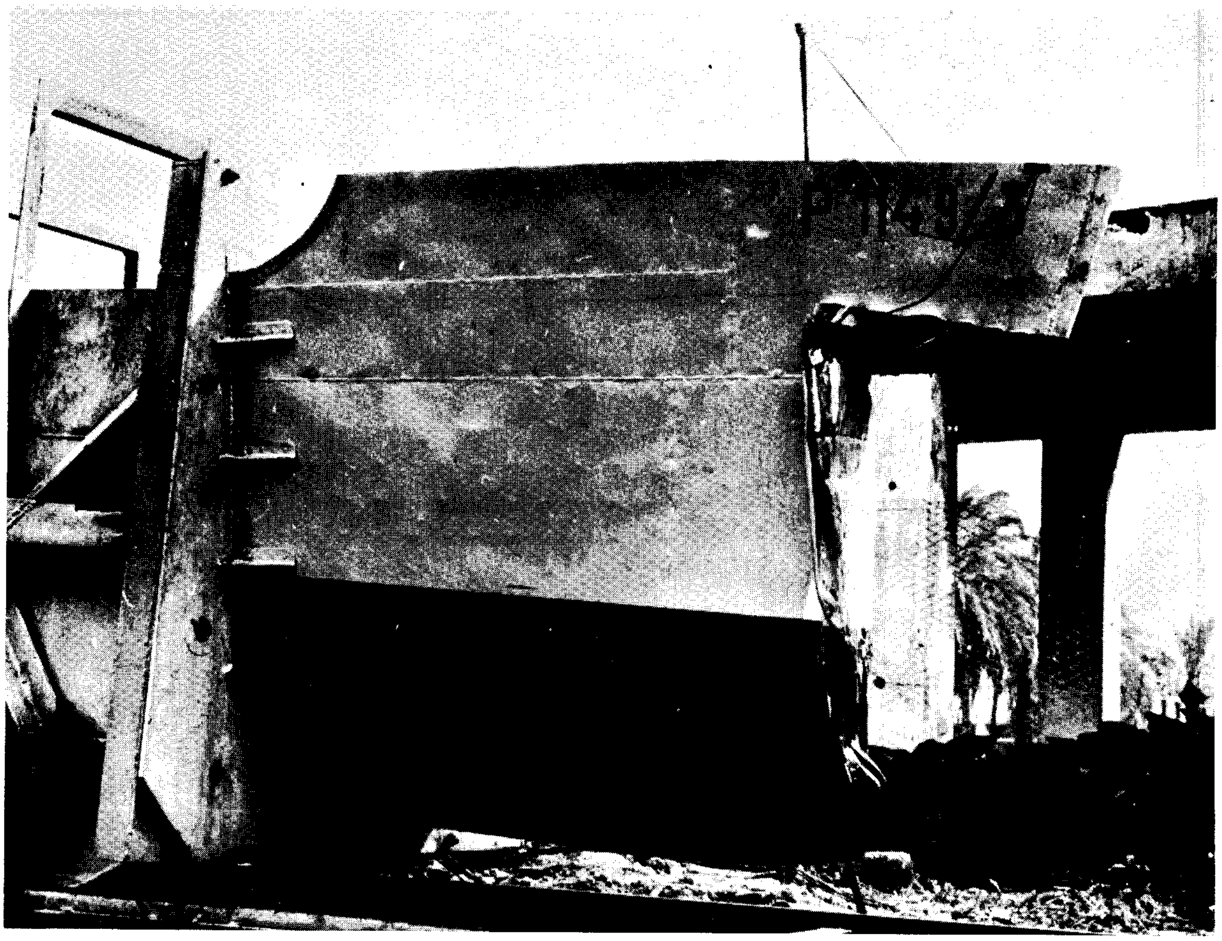
17. Bow model after collision test No. 4



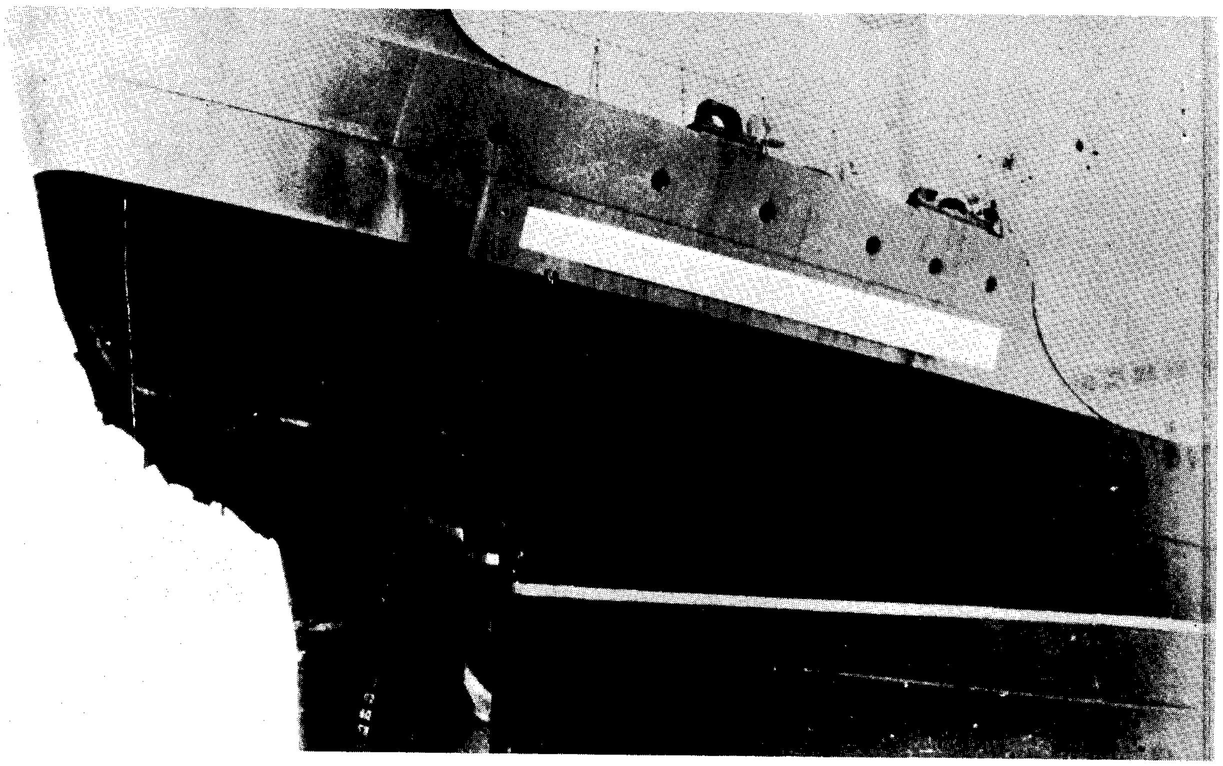
18. Ship side model after collision test No. 4



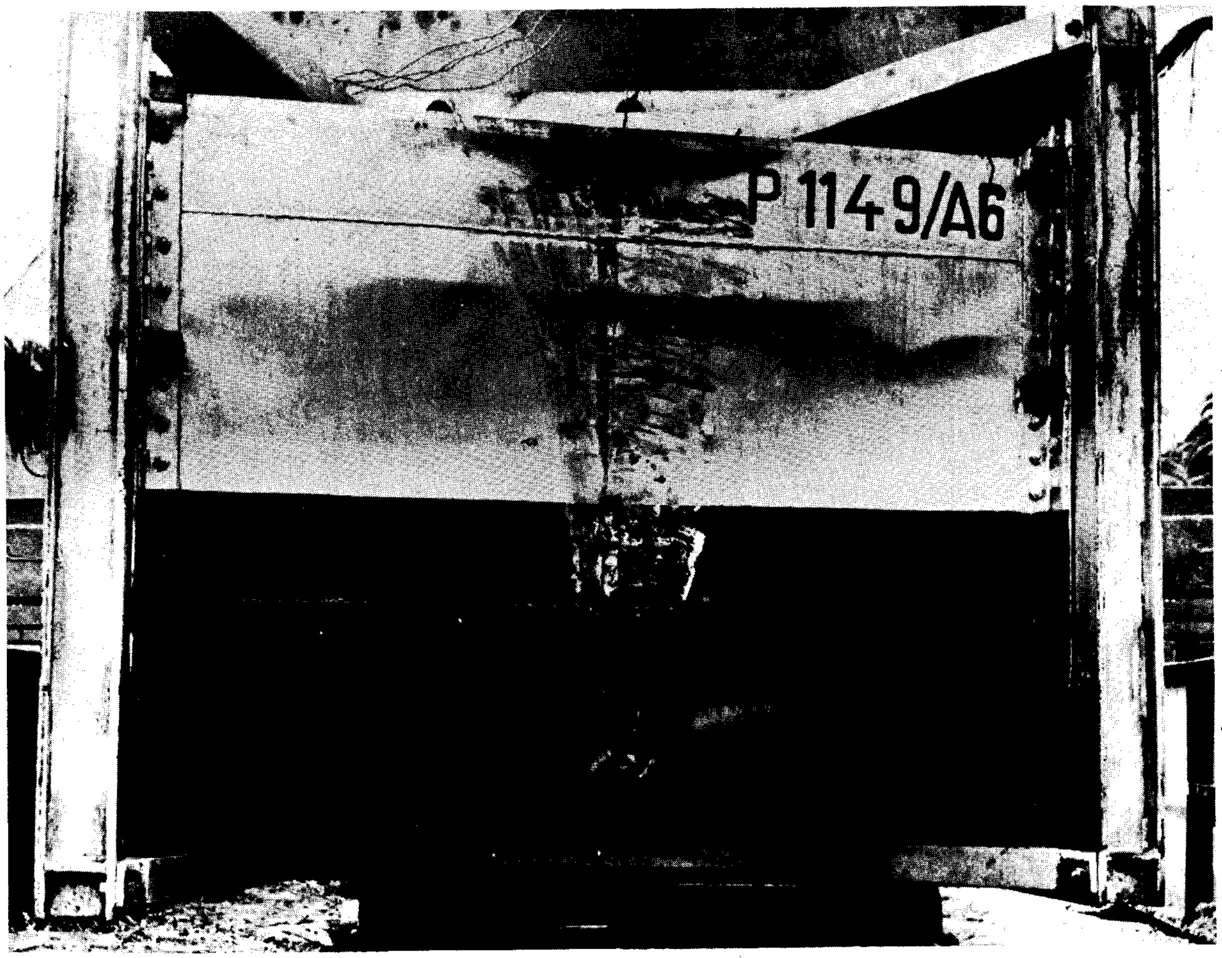
19. Bow and ship side models after collision test No. 5



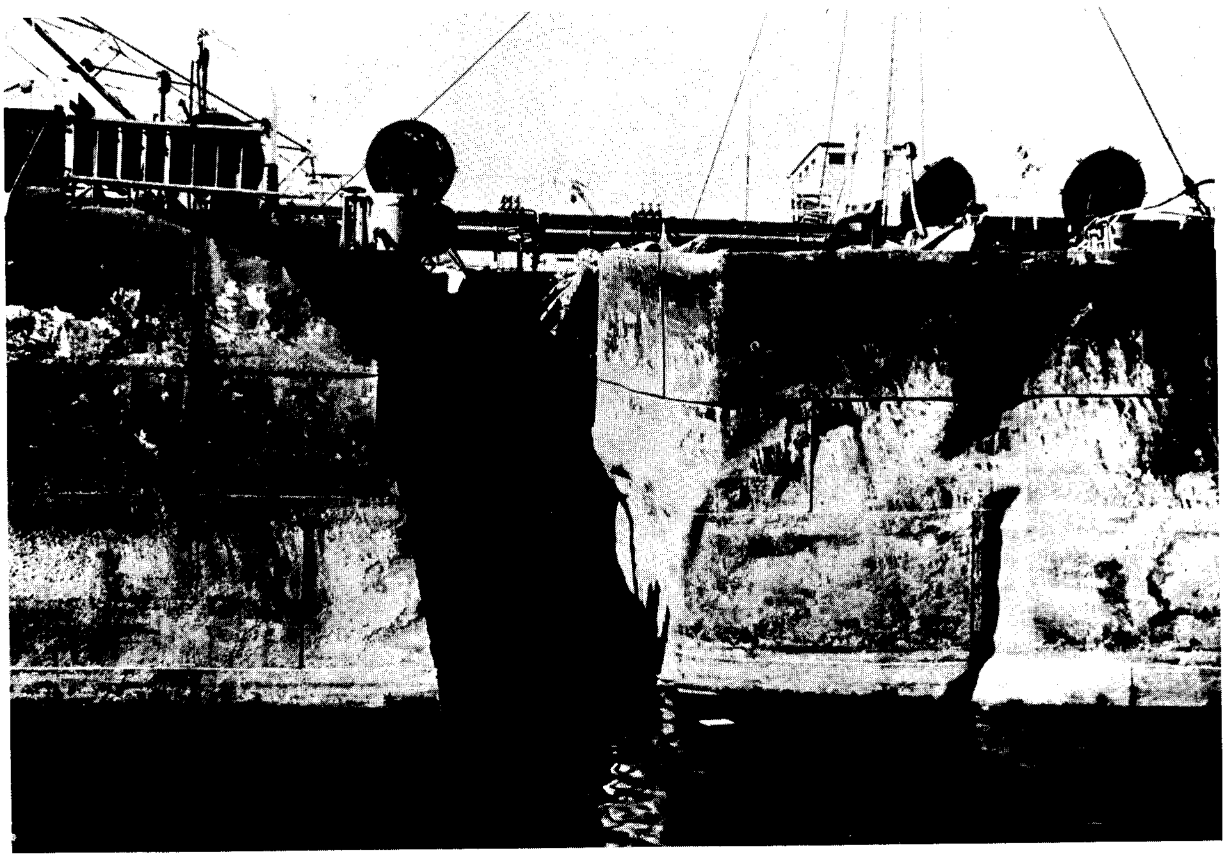
20. Bow model after collision test No. 6



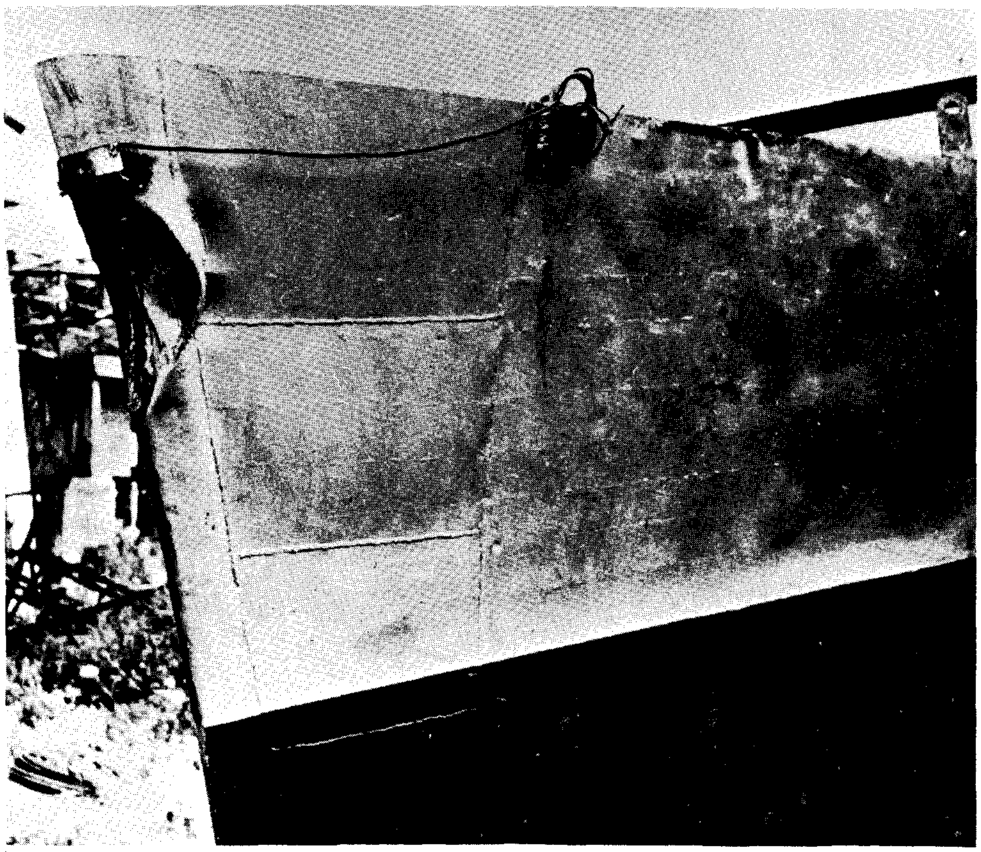
21. Bow of the colliding ship after real collision



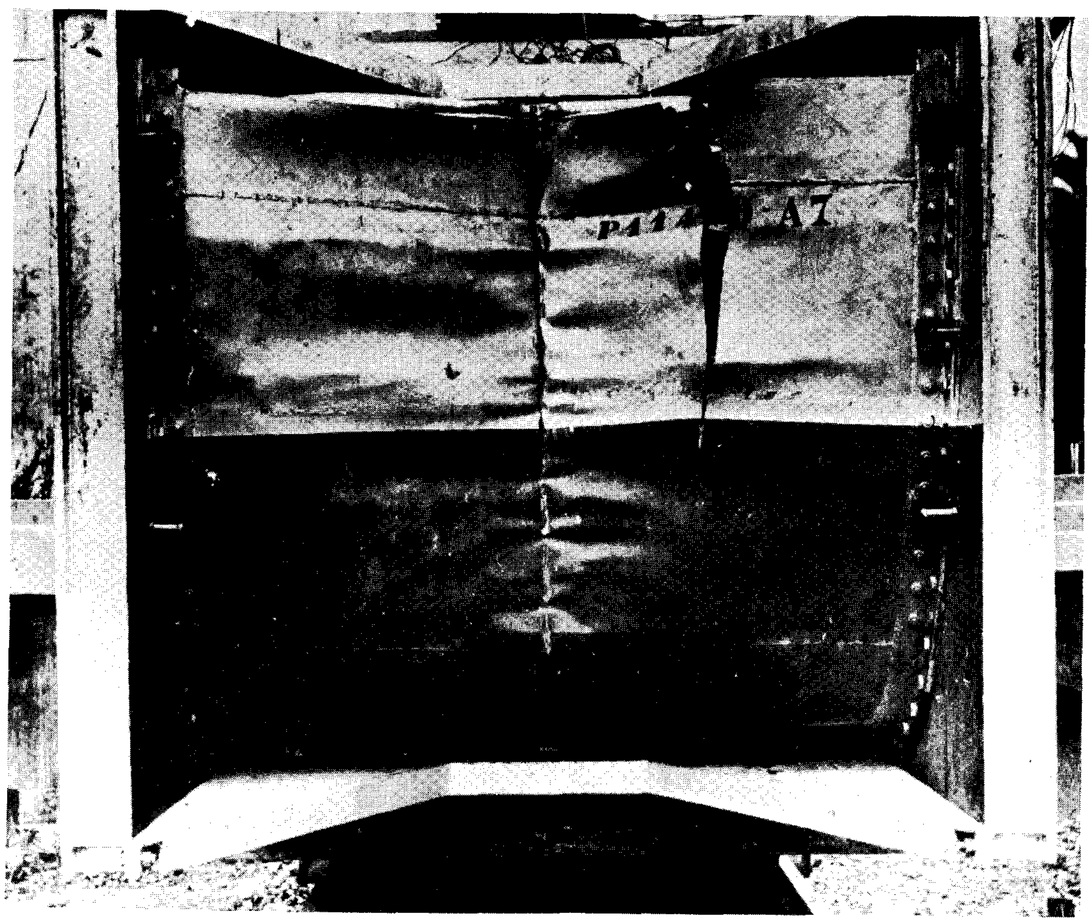
22. Ship side model after collision test No. 6



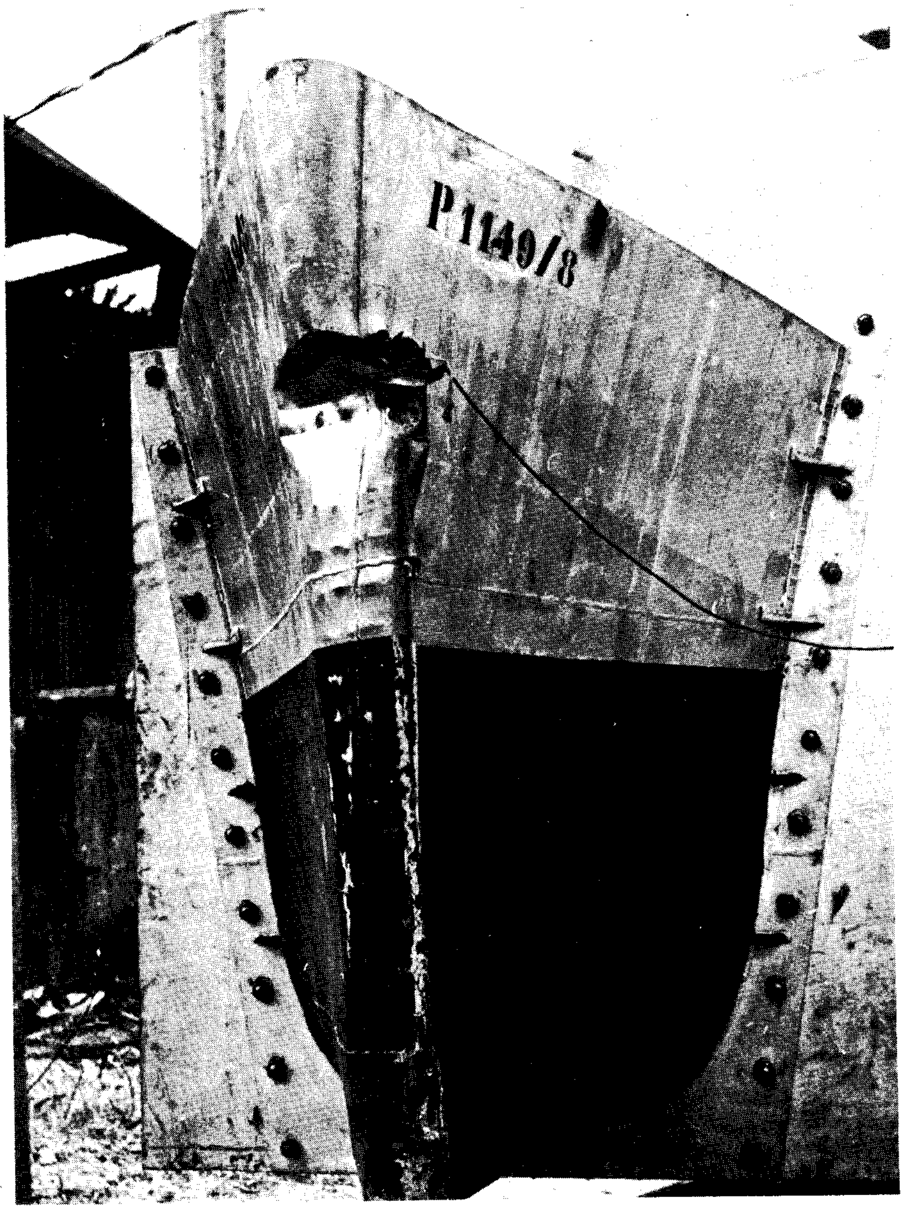
23. Side of the struck ship after real collision



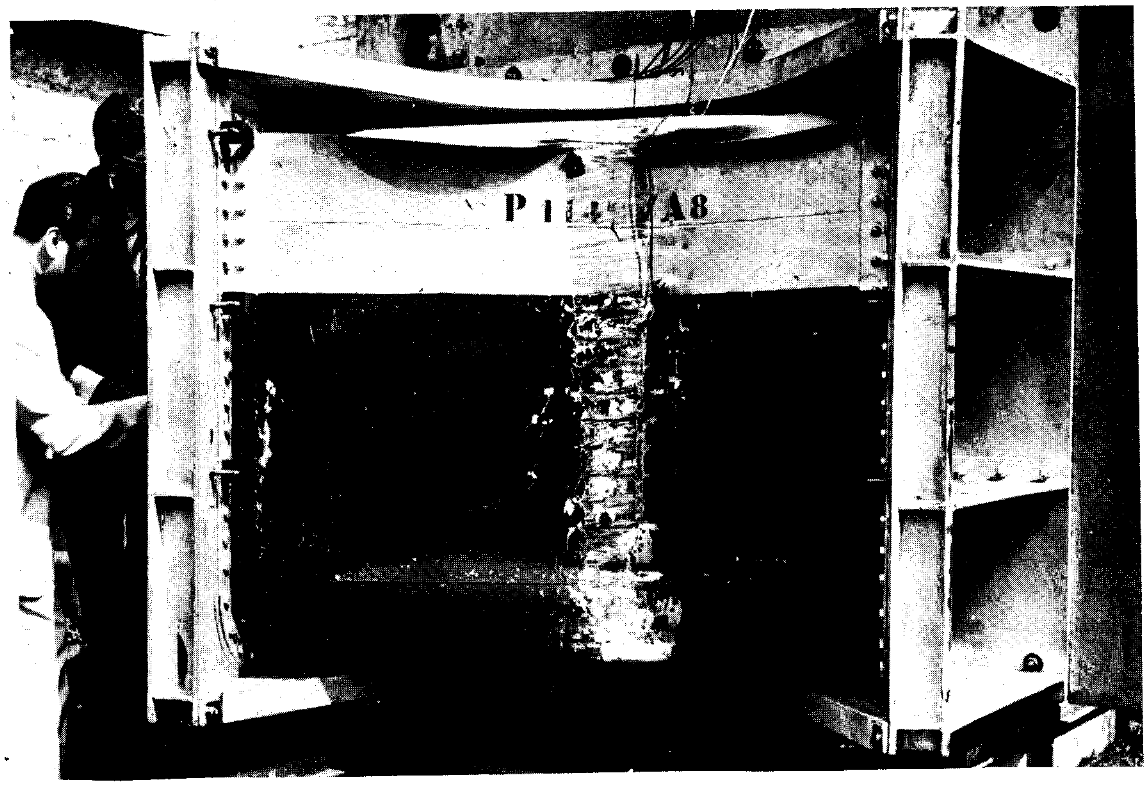
24. Bow model after collision test No. 7



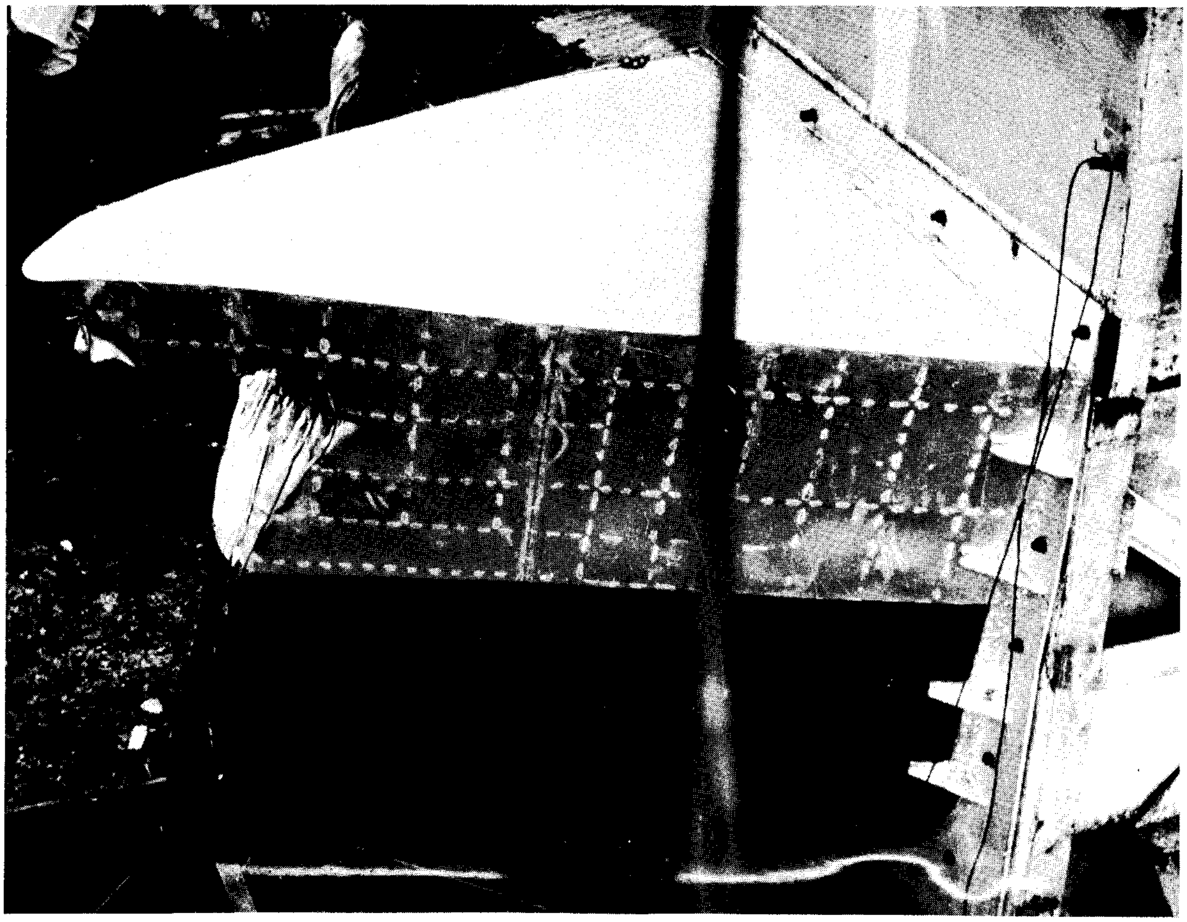
25. Ship side model after collision test No. 7



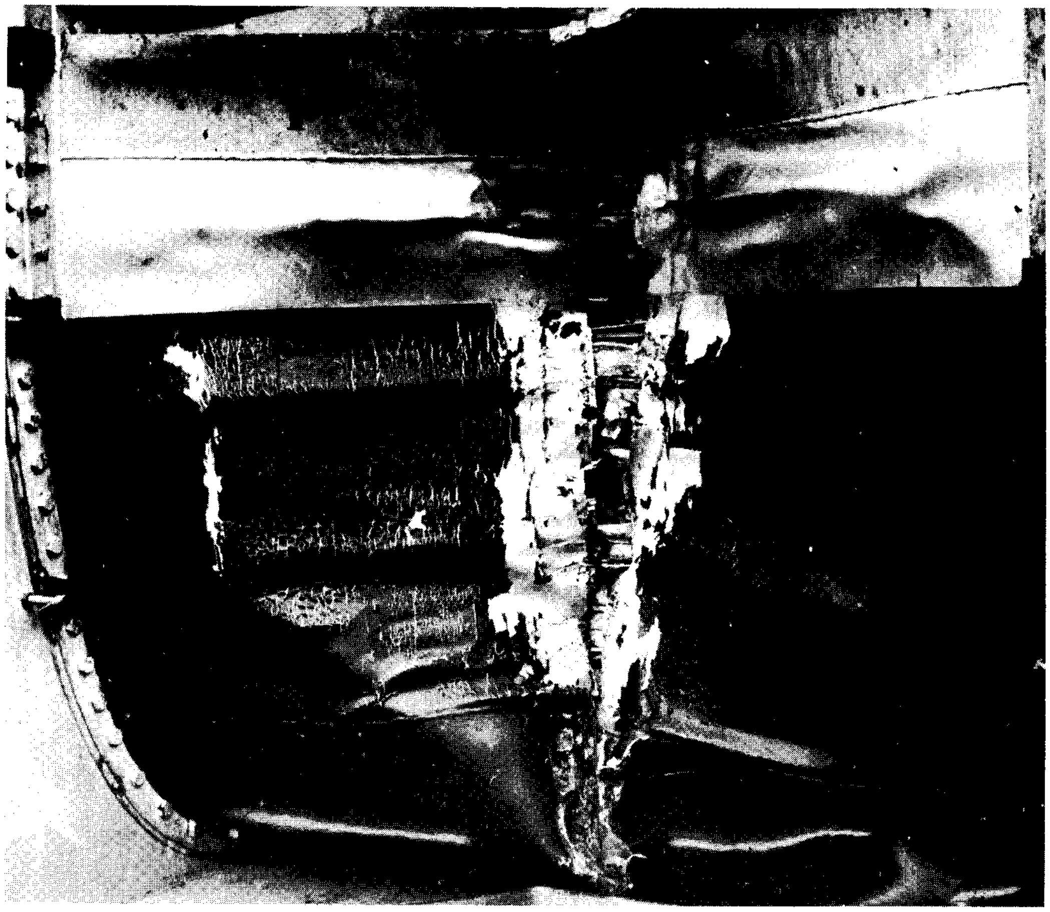
26. Bow model after collision test No. 8



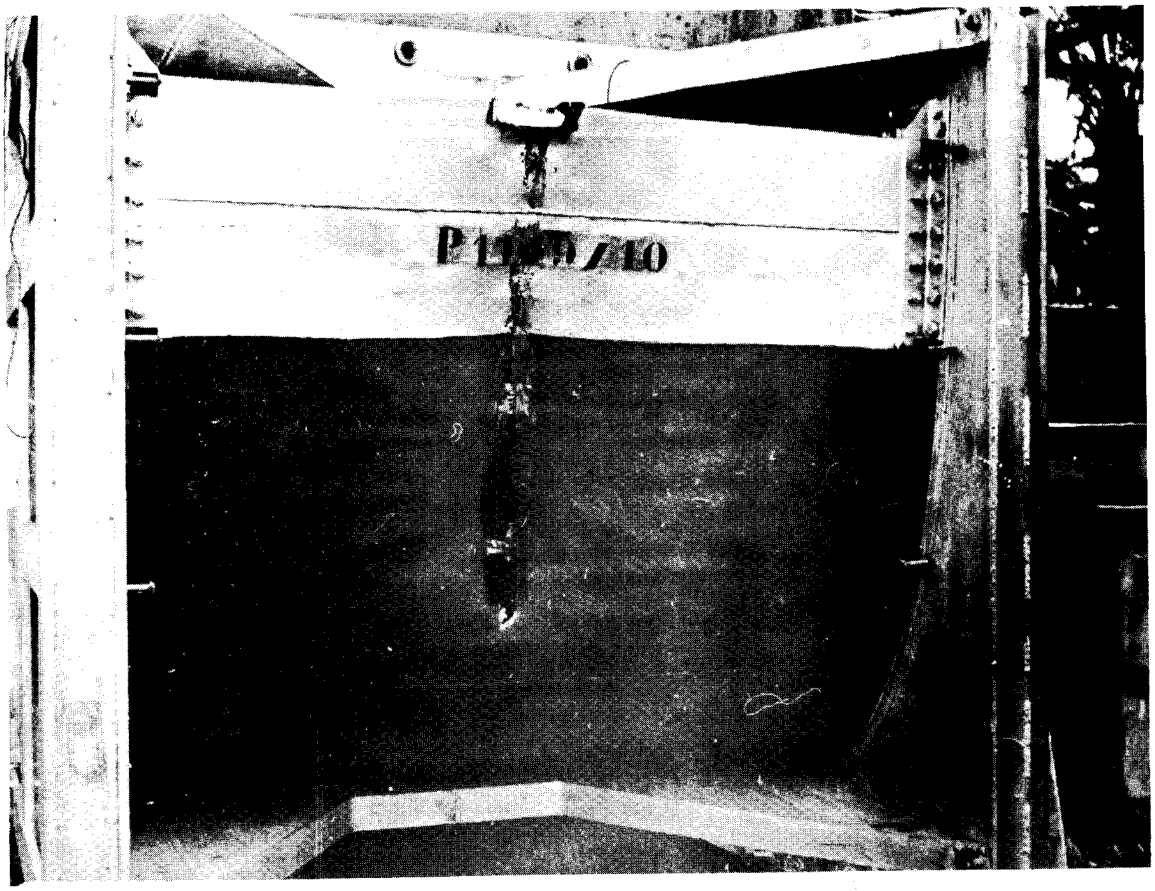
27. Ship side model after collision test No. 8



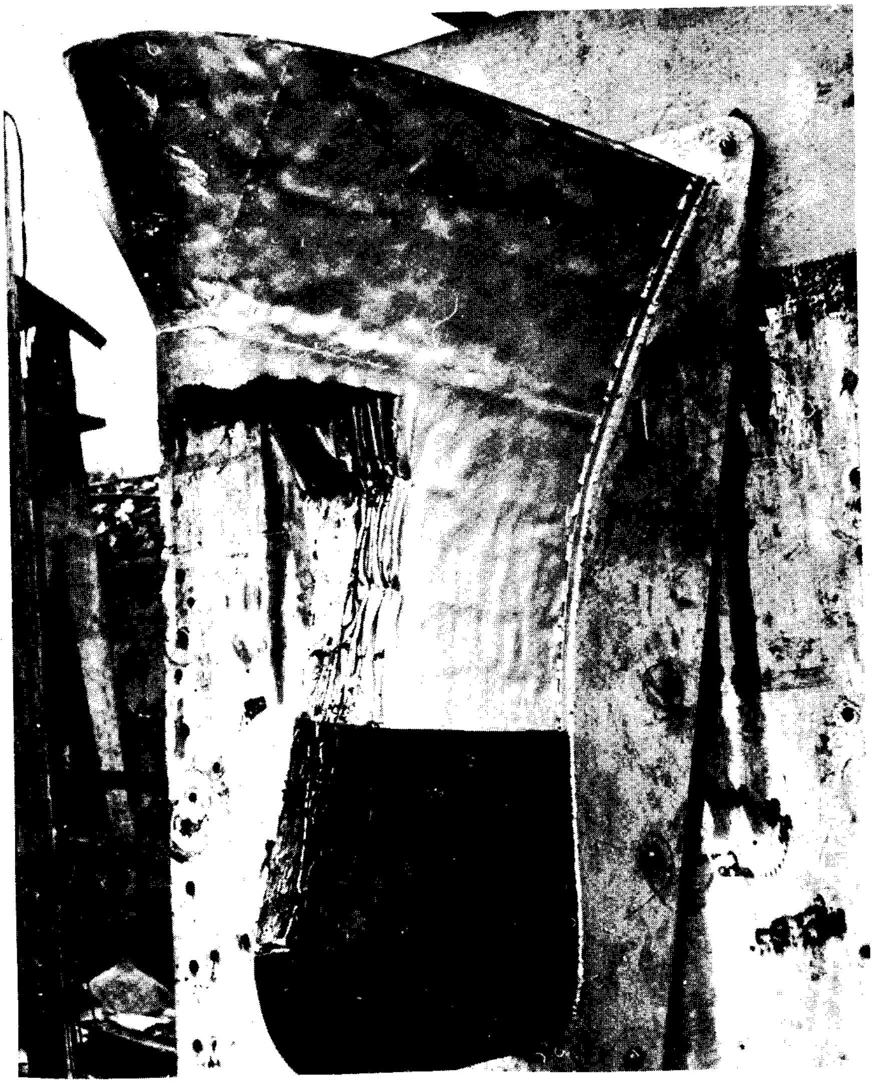
28. Bow model after collision test No. 9



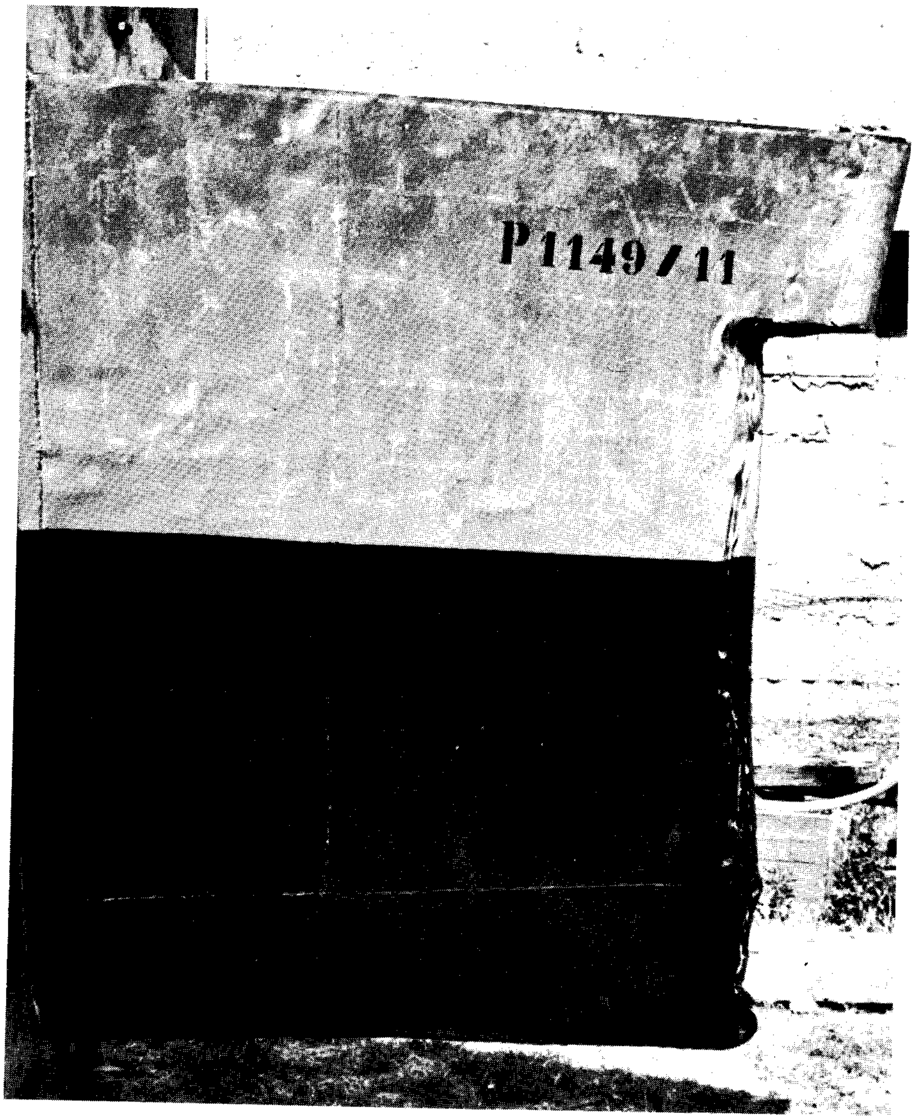
29. Ship side model after collision test No. 9



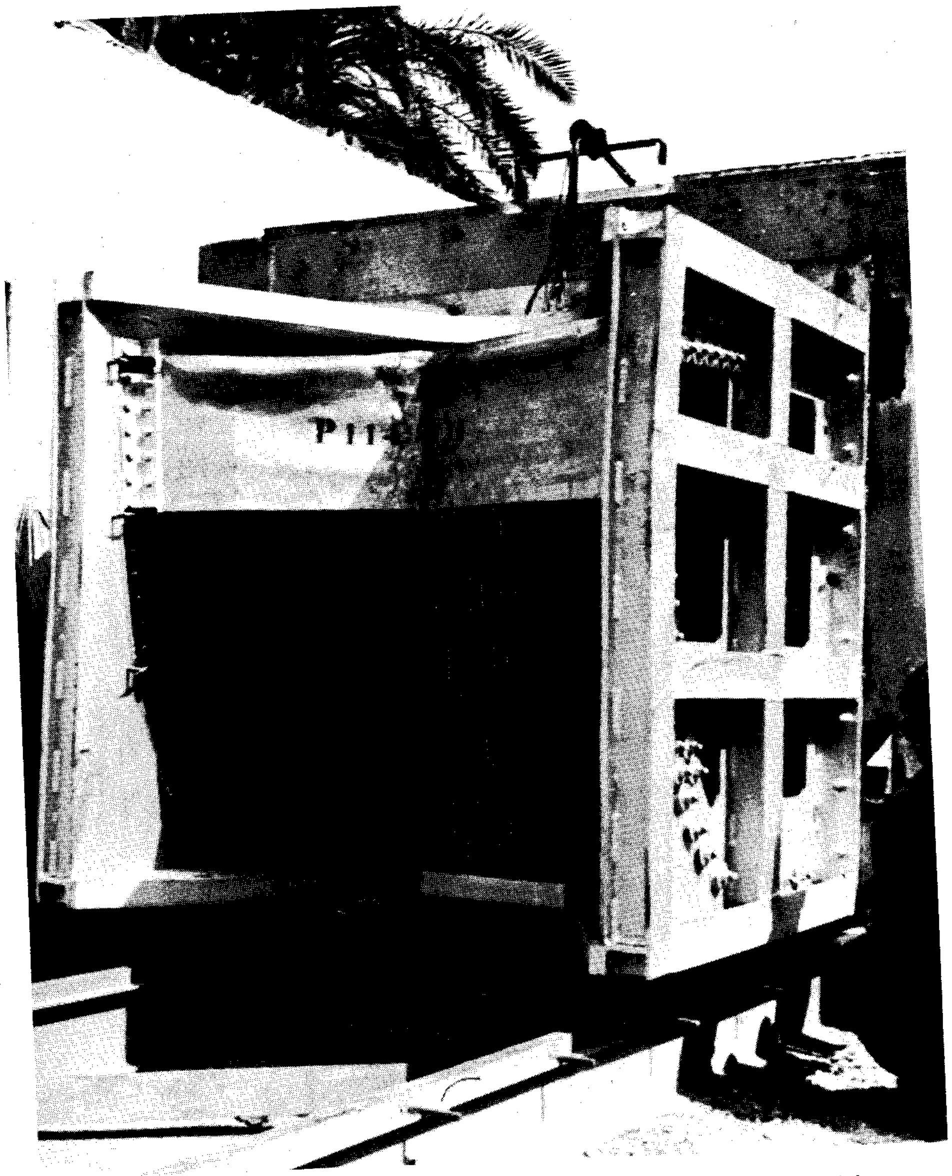
30. Ship side model after collision test No. 10



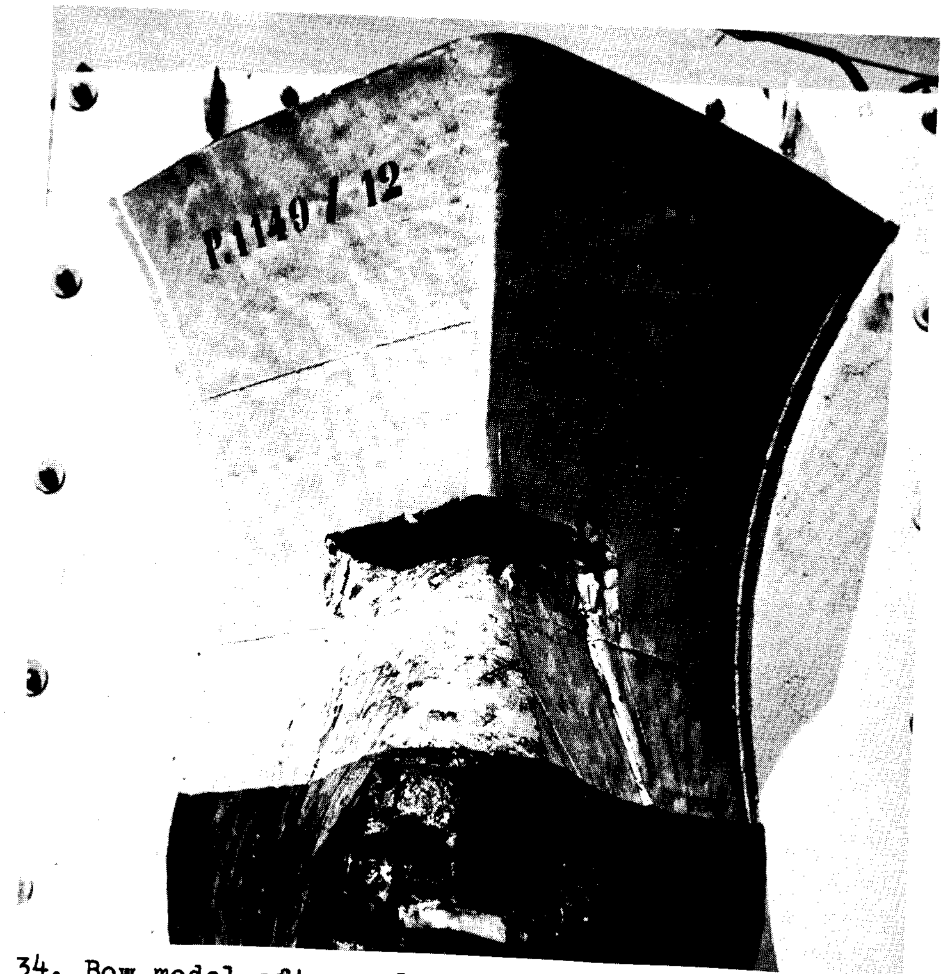
31. Bow model after collision test No. 10



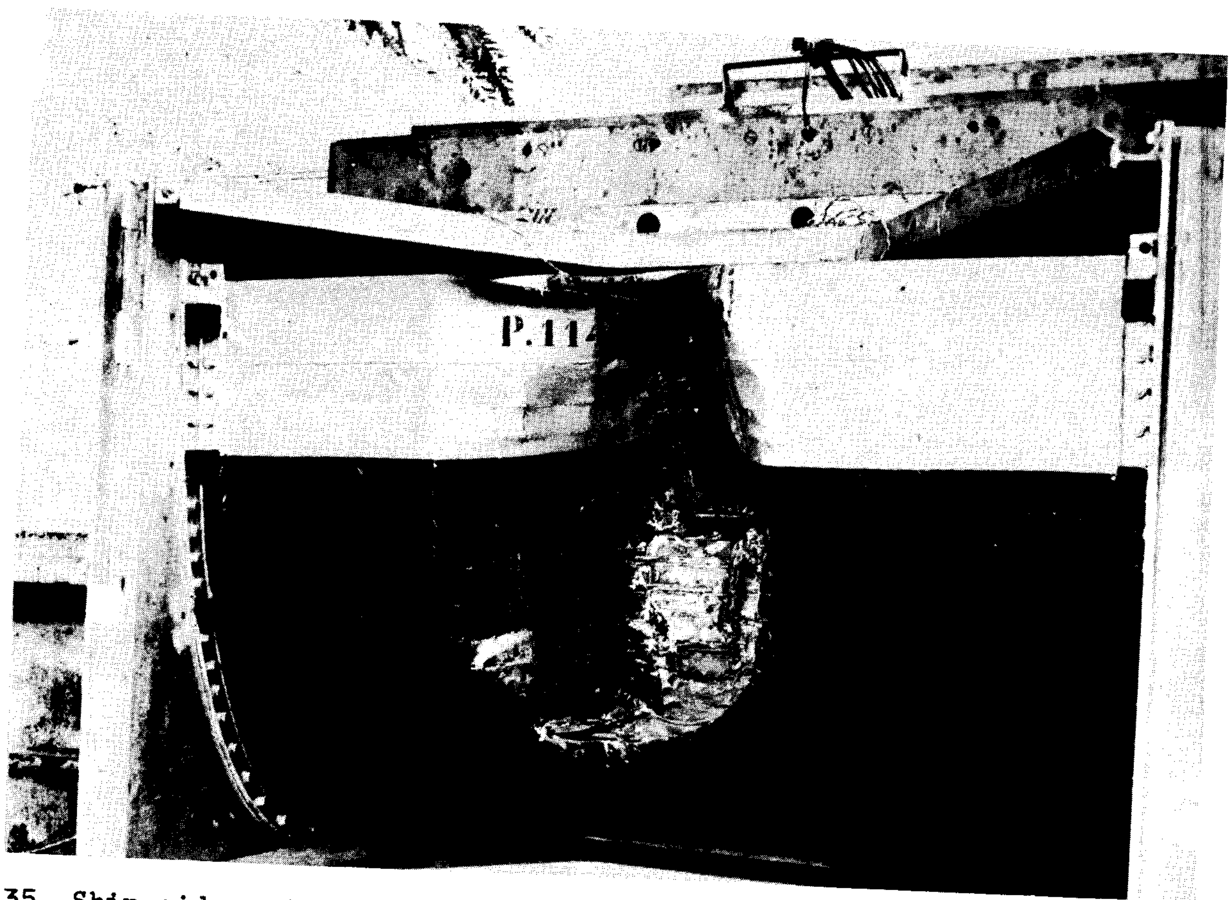
32. Bow model after collision test No. 11



33. Ship side model after collision test No. 11

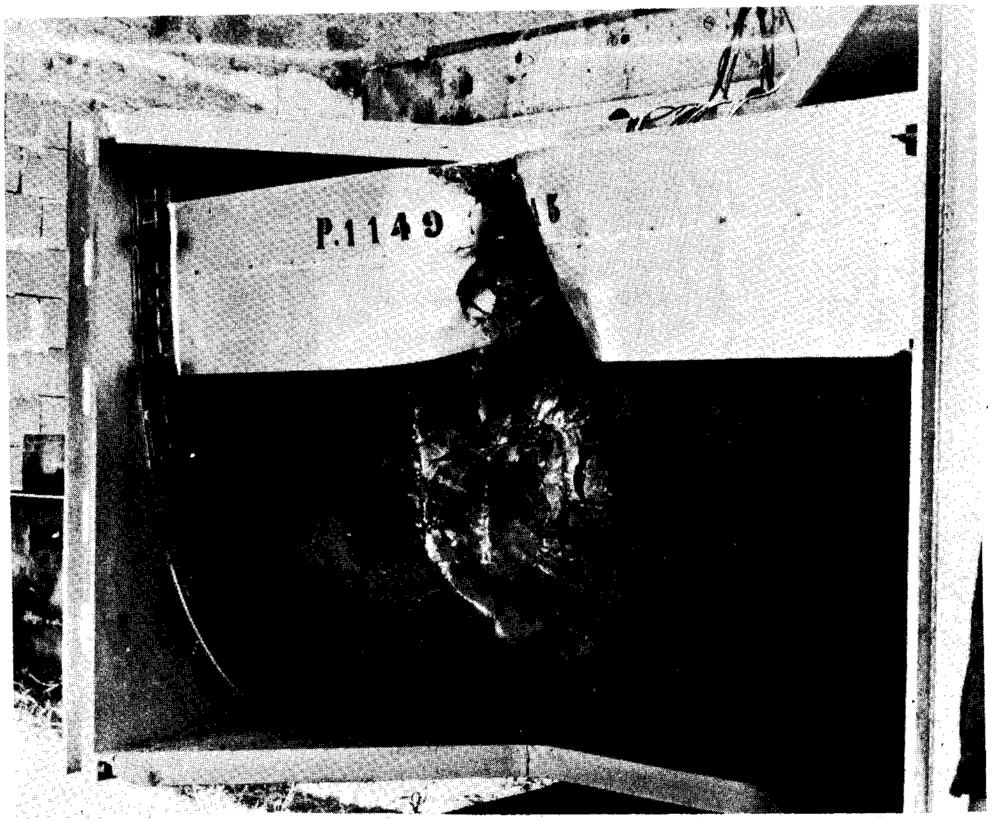


34. Bow model after collision test No. 12

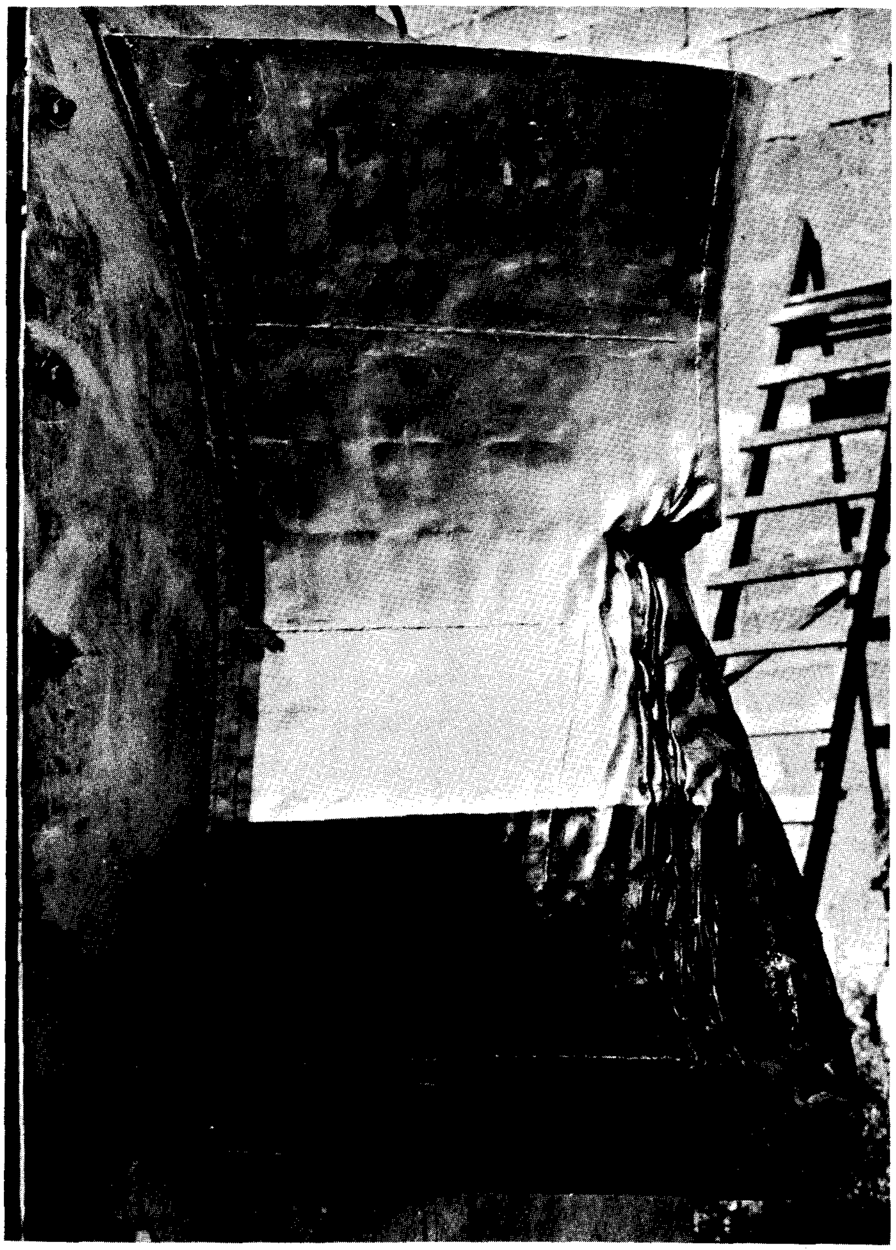


35. Ship side model after collision test No. 12

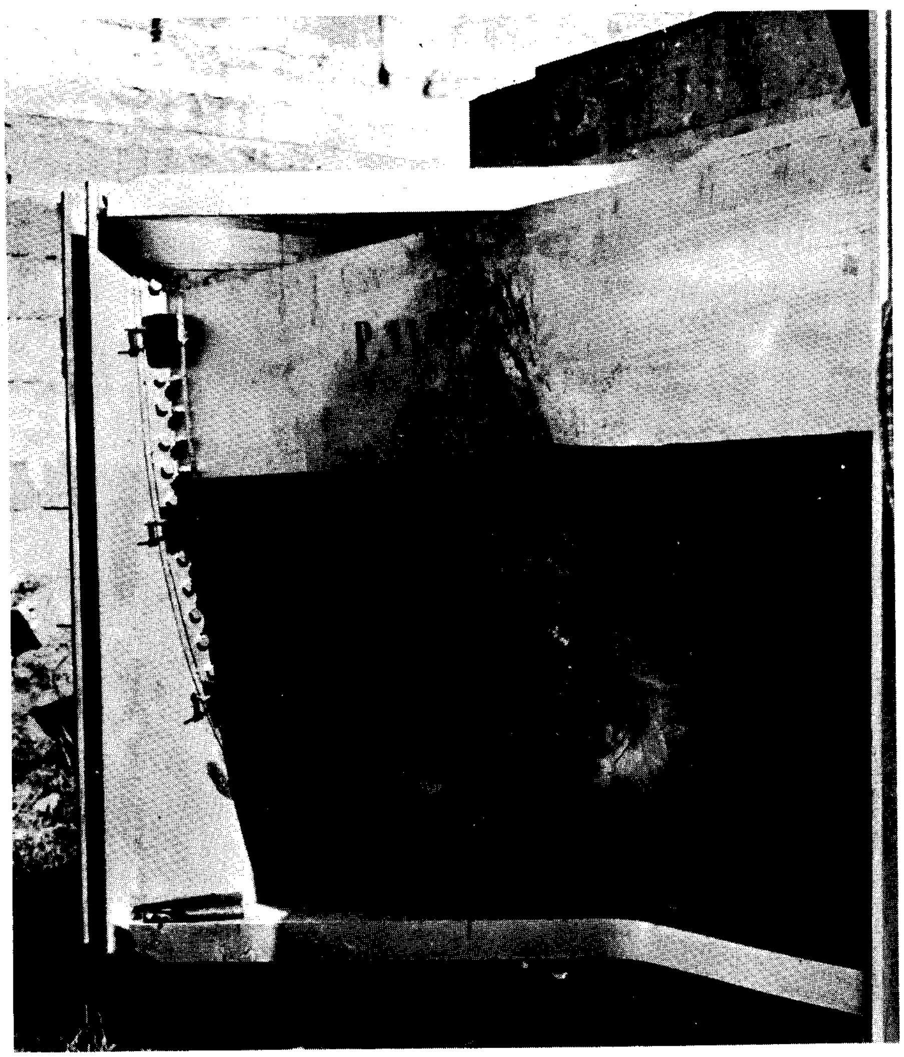
36. Bow model after collision test No. 13



37. Ship side model after collision test No. 13



38. Bow model after collision test No. 14



39. Ship side model after collision test No. 14

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Alfred Nobel

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