THE DESIGN AND TESTING OF LARGE BORE HOSE USED FOR THE OFFSHORE TRANSPORTATION OF HYDROCARBONS

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SUMMARY

For many years the physical design of offshore hose has relied on subjective opinion and experience only occasionally populated by factual design criteria. Similarly the testing of the same could only be described as a faint shadow of reality. It was against this background and an increasing history of hose failures that the work described by this thesis was initiated in the form of an I.H.D project put forward by Dunlop Limited.

The thesis describes the initial investigation into the problem and how this was translated into long, medium and short term objectives together with an action plan necessary to achieve the same.

A need for objective design information was identified in the very early stages of the project. This precipitated the setting up and conduction of field tests, offshore Nigeria 1973, designed to monitor the forces acting on such hoses at a Single Buoy Mooring installation. In order to extend the scope of the above tests a need was highlighted for the establishment of model testing facilities aimed at studying the submarine hose system design for such terminals. Such facilities were designed, installed and resulted in the establishment of much desired information.

On the basis of the above work, a full scale test rig was designed, in order to simulate load generation on hose under laboratory conditions. The rig as designed was installed during 1976/7, following which initial hose testing has been conducted.

On the basis of results to date, all indicaters point to a realistic testing situation (rig and service failures relate), which should enable future hose designs to be based on scientifical fact rather than opinion.

INDEX TERMS:

OFFSHORE TERMINALS
HOSE
MODEL TESTING
TEST RIG

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I. SYNOPSIS AND CONCLUSIONS

Hoses have been used at single buoy mooring terminals since their inception in 1958 and during this time tanker sizes have grown enormously with the hoses servicing them growing in proportion. Changes in hose design have also occurred but often only as a result of all too painful experience. In the early days problems were perhaps to be expected, as a product designed essentially for use on land, was being asked to operate reliably in a totally different environment. Early experience was, however, quite encouraging and, other than the increase of hose size the basic technology of hose design was not forced to adapt to the new circumstances.

Eventually, the trend to larger sizes revealed the essential weaknesses of the designs and the criteria upon which these decisions were based. Early attempts to improve hose life and reliability were based upon an intuitive understanding of experience and, although often successful, the performance of the next generation of hose and its relationship with the design changes was not understood sufficiently to optimise the design process.

An initial investigation by the writer confirmed the already held suspicion of an increasing hose failure rate with respect to increasing bore size: Also the fact that poorly kept records, by oil companies and manufacturers alike, had in effect been masking the above trend for some considerable period. In addition it was also determined that:-

(a) There existed a total lack of design criteria data upon which to reasonably specify the performance and characteristics required of the product.

(b) Given the assumption that the latter could be made available, few design methods, analysis techniques etc were available within the industry by which to utilise the information gained.

(c) Totally inadequate test methods, procedures or indeed equipment existed by which to evaluate revised or even existing product design prior to service installation.

A strategy was therefore put forward by the writer in order to supply the needs expressed in relation to the aforegoing statements. The basic elements of this strategy may best be described by Figure (20), however the programme basically involved:-

(1) Establishment of design information.

(2) Analysis of such in relation to product design.

(3) "Realistic" testing of resulting designs.

Because of the organisation of the company in which the project was conducted, the writer achieved total responsibility for areas (1) and (3) and a shared responsibility in area (2). It was proposed that the existing development potential should be expanded in order to forefil much of the improvement needed in relation to direct hose design technology.

FIELD TESTING

After careful study of the problems involved it was considered that in order to gain a knowledge of design forces etc., which were reliable and attainable in a short period of time, field measurements would be required. A comprehensive field study programme was therefore set-up in order to monitor the loads acting on hoses at a typical single buoy mooring terminal. The location eventually chosen being offshore Nigeria.

Because work of the nature involved had previously not been conducted in relation to such a situation it was necessary to formulate study methodology as well as equipment design and construction. Due to the adverse environmental conditions under which the tests would be conducted great emphasis was placed on system integrity and reliability. The study was eventually conducted during the last half of 1973. (NB: Since the completion of this study a similar study has been conducted in Japan by a competitor - results unpublished, and further, such studies are being discussed by major oil companies).

As a result of the said field work reliable design data was at last made available. The results of the study may be summarised as:-

(a) The information required in terms of the identification of the principle loading modes, their respective magnitudes and application frequencies were determined.

(b) A general hypothesis that most failures result from an inability to combat fatigue rather than an exceeding of 'design' loads.

(c) Some correlation between the various forces measured and prevailing sea conditions, particularly a dependence in relation to short period, (choppy), waves.

(NB: One immediately observable result of the said study was the design of a new type of "Buoy Connection Hose" which has been installed and proved to give longer life expectancy by a factor of approximately 3, ie: 6-18 months).

MODEL TESTING

It was recognised at the on-set of the project that field testing could only forefil a small, allbeit extremely vital, role in relation to establishing design criteria. Resources and expense were naturally heavy factors in this decision although practicalities of the various system designs employed also were extremely important considerations. Whilst floating hose systems may be generally considered similar, (apart from weather conditions) for all terminal sites, the differences and scope of submarine hose systems was immense. Apart from the basic system designs commonly employed, ie: Lazy 'S', Steep 'S', Chinese Lantern etc., many variations exist in respect to each of these resulting in such complex interaction that no two systems may be regarded as identical. It was in this area therefore that it was felt that model testing could play an extremely important role.
Whilst determining the type of facility required it was recognised that many forms of model testing basins existed both within the UK and World-wide. However it was discovered that although such facilities had been latterly used to study SBM terminals, such studies had been generally restricted to the buoy, tanker, mooring and anchoring systems. On the few occasions that such tests included the submarine hose system, the results had been uncompromisingly bad. The latter generally being due to the high modelling scales employed and a lack of knowledge/appreciation with respect to hose characteristics.
Armed with the above information a proposal was put forward for the installation of a purposely designed unit aimed at filling this gap in existing facilities. The proposal was accepted and the necessary equipment designed and installed during 1974. This equipment was

employed with considerable success with respect to providing information to the company in terms of design data and in terms of design studies for potential customers. Due to the success and resultant high utilisation of the facility installed a second and more sophisticated unit was designed and installed during 1976.

Generally the benefits derived from the model testing conducted may be summarised as:-

(a) An indication that no single system or system variant may be deemed as the 'ideal solution'. Each site or terminal has its own particular anomalies which require a tailor designed system.

(b) The need to clearly define realistic system parameters (eg: water depths, sea conditions etc) before commencing to design the submarine hose configuration.

(c) Identification of areas where small changes can influence major changes to the system, (eg: manifold angles, hose lengths etc.)

(d) The ability to optimise the design of system for stated conditions, then measure the loads inherent in that system design under varying conditions - Survival, Maximum and Normal Operating.

(e) The provision of the above data for hose design and testing purposes.

The success of the model testing facility can be judged by the number of substantial contracts won by the Division in which model testing formed an integral part of the package.

DYNAMIC TEST RIG

Of fundamental importance to the project, (its aim as expressed by the Company), was the design of a simulation testing facility. Previously the only method of evaluating revised design was to put prototypes into service and await results, with the attendant consequencies of such actions.

The subject areas previously described, whilst complete in their own rights, were designed so as to provide data for the design and programming of such a rig. With the data thus provided a design study was undertaken by the writer resulting in a number of design options. From these a single option was selected (nominally on an information : cost ratio basis) for a further detailed design study.

On completion of the said study a technical and financial presentation (Sanction application) was made to Dunlop, whereby a request was made for funds in order to build such a test rig. The application was granted, and the rig successfully constructed and installed during 1976/77.

A hose test programme has been initiated on the test rig (initial programme period covers 12-18 months), which has been found to perform satisfactorily. On the basis of early test results, the failures produced by rig testing concur in general type with those occurring under service conditions. The hypothesis that the rig is realistically simulating service loading conditions might therefore be made although it is recognised that further proof is required in order to confirm the initial results.

Since the subject of the project has been real problems in a practical environment, frequent reference is made throughout this thesis to commercial and financial factors, which have been treated with equal importance to technical consideration.

FURTHER WORK

In many ways the methodology of this project and thesis has been to create the foundations and framework upon which to build the future development of offshore hose. It is recognised that much of the work contained is either incomplete or subject to improvement or extension. It is considered that additional work is required in some or all of the following areas in order to supplement the work presented.

(a) It is essential that further field work is conducted in order to confirm the results of earlier work and that information gained from model testing. In addition greater emphasis should be placed on attempting to correlate such loads with prevailing environmental conditions.

(NB: Discussions are currently taking place, August '77, between Dunlop Limited and SBM Inc - Monaco, in order to conduct such a joint study in the North Sea).

(b) Model testing should be continued and expanded, hopefully until such a state is reached that all submarine hose system designs are subjected to model testing prior to an installation taking place. Tests should also be conducted to establish the effect upon such systems of revised hose designs and characteristics.

(c) Based upon the results of the initial test programme further testing will be conducted on the test rig in order to meet the objectives as laid down in this thesis. Simultaneously research is required into hose orientated "NDT" techniques which may be used in relation to both rig testing and site inspection.

(e) The mathematical models developed require additional sophistication in order that they might better describe reality. In many cases this will require a systematic review of the initial simplistic assumptions which were made.

(f) In addition to the above, much work is required into those areas of the hose design which might be termed rubber technology. This area has purposely been neglected by the work presented in this thesis due to the vast area already considered.

It must be stated that this thesis; rather than an authorative work on the subject of hose design and related technology, if not as a beacon to illuminate the true way, then as a candle to show where others have already been.

II OFFSHORE MOORING TERMINALS

II.1 INTRODUCTION

In the history of nations, no single event in the last 100 years, not even the number of global wars, has created so much panic, has shaken the economic stability of such a large number of affluent as well as developing and poor nations, as the sudden rise in oil prices. This has created great inbalance in the payment situation of many countries and brought about considerable strains on development plans.

Few would argue that oil energy is a key element in the well-being of people all over the world. There is much to bet that this preponderance will probably last until the end of our century.

A great number of related activities have sprung up around oil production and transportation. Among the most important is the off-shore terminal construction industry, which came as an answer to the problems of handling oil transfer and storage at sea.

"Terminalling plays an extremely important function in the immense ocean transportation system that is vital to the global distribution of petroleum. Reliability is a must".

The following paragraphs survey some of the events that precipitated the single point mooring system (S.B.M). Also are briefly described the factors that are determinant in the SBM's favour and make it prevail over other systems.

Millions of years ago, the earth laid down thick deposits of organic material that under heat and pressure became coal, oil and gas. Since being discovered, these fossil fuel deposits have been consumed at ever increasing rates, mainly because of their cheap development cost.

Most of the first discoveries of giant oil fields were made in remote areas where natural harbours were nonexistent. This created the need for artificial berths to moor the tankers during their loading operations.

With the growth of tanker size and the high cost of building artificial harbours, the way was paved to make use of offshore mooring terminals (bringing the moorings to the ship). Many types were tried, but in view of its flexibility, reliability and economy, the single point mooring (SPM) prevailed.

II.2 OIL - ITS PRODUCTION AND CONSUMPTION

The history of energy is closely linked to the history of industry and economic growth.

During 1974, the world consumption of all types of primary energy was the equivalent of over 115 million barrels of crude oil per day. more than three times the 1950 requirements.

In the same period, the consumption of petroleum products had jumped from 9.6m b/d to 45m b/d. (See Fig. 1).

Oil has been discovered almost everywhere: North America, North Sea, North and Central Africa and Indonesia. However, the most productive fields were found in the Middle East.

Table 1 shows the total production and consumption in 1970 for the different parts of the world. As can be seen, Western Europe was in need of about 11.4m b/d in excess to its production, the U.S.A. 5.1m b/d and Japan 4.9m b/d.

This disparity between production and consumption in the different countries, which has now been going on for many years, is creating a worldwide petroleum flow.

This flow was around 21.4m b/d in 1970. It reached 27m b/d in 1973.

Fig. 2 illustrates the movement of oil in 1970. Obviously, most of the oil was exported from the Middle East to the West and to the Far East.

In order to transport such big amounts of oil, over the years large fleets of tankers were built and world tanker routes were established.

Fig. 3 shows these world tanker routes and the oil loading and unloading ports.

II.3 TANKER GROWTH

Until 1956, the largest tanker did not exceed 56,000 dwt.

That was the year when the Suez Canal closed. This event triggered an ocean bulk transportation revolution : new routes had to be established around the Cape to carry the evergrowing flow of oil from the Middle East to Western Europe.

Longer tanker routes and a rising demand on the transport market pushed the charter rates up, encouraging the newly prosperous owners to re-invest in their fleet. Tankers increased in numbers as well as in tonnage.

At the end of 1968, the world fleet totalled 119,348,000 dwt for vessels in the 2000 dwt class and over, while figures in June 1970 for 6000 dwt class and over were 154,509,000 dwt. This represents an increase of more than 30 million dwt in an eighteen-month period. Since tankers could no longer navigate through the narrow waters of the Suez Canal, new standards appeared. Draft and ship length no longer had to be restricted. Shipyards were flooded with orders for larger ships. The era of the 1,000,000 dwt megatanker was not far away.

The following list of world record holders illustrates the rapid growth of tankers over the past two decades.

NAME	DWT	LAUNCHED
Sinclair Petrolure	56,089	1956
Universe Leader	85,515	1957
Universe Apollo	104,520	1959
Nisho Maru	130,250	1962
Tokyo Maru	157,290	1966
Idemitsu Maru	206,000	1966
Universe Ireland	326,000	1968
Nisseki Maru	372,700	1971
Globtik Tokyo	477,000	1973

II.4 TYPES OF OFFSHORE TERMINAL

It has been proven that large tankers have an economic viability superior to that of their smaller counterparts. However, proliferation of large carriers was rendering many of the world's traditional ports obsolete. It was also causing public concern because of hazards to port traffic and pollution risks. The situation became even more critical as the consumption of oil in heavily populated centers increased while oil was found always further afield. It was imperative to find other facilities for handling oil.

Many solutions were studied and sometimes experimented with different degrees of success. Among them:

- The artificial harbour protected by concrete jetty and breakwater. This was found to be a very expensive solution, especially in shallow waters since large amounts of sediment had to be dredged (Fig. 4).
- The artificial offshore island made up of steel construction or precast concrete pilings supporting precast or cast-in-place concrete deck structures. (Fig. 5).
- The multiple-buoys-mooring system, consisting of several mooring buoys anchored around a berth.(Fig. 6).
- The tower mooring system, consisting of a steel structure fixed to the bottom by piles. A turntable is fitted on top of this structure, from which a mooring rope is connected to the ship. (Fig. 7).
- Single Point Mooring System of either the catenary-chaintype SBM or the Single Anchor Leg Mooring (SALM). Both allow the ship to rotate freely and take the position of least resistance to the combined actions of waves, currents and wind. (Fig. 8 and 9).

In order to make a valid comparison between these systems, a site has been simulated where any of these systems may be installed. Site, sea and weather conditions are assumed and the mooring forces for each system estimated. This criterion is the major factor determining the size of different terminal components and, consequently, the weight and invested capital.

Fig. 10 illustrates the mooring forces on each system when all factors are fixed. Note that mooring forces on the SBM are comparatively the lowest: this is due to the system's greater flexibility and the damping effect of the chains.

In order to understand the popularity of the SPM system it is necessary to compare the operating costs of such terminals as shown by Tables (2) and (3).

It is obvious that up to a 15,000 t/hr throughput the SBM terminal is more economical and more flexible.

The multi-buoy mooring system finds its application only in protected waters and for relatively small tankers.

The fixed conventional berth seems justified only when the throughput is very high: above 20,000 tons/hr and for huge-size tankers. Furthermore, the waters should be well protected. Otherwise,a breakwater has to be constructed at high cost.

The tower terminal does not arouse much interest. Apart from its costly operation, it presents collision risks. However, it is believed that it can find an application in shallow, protected waters.

II.5 THE SINGLE POINT MOORING SYSTEM

In view of its relatively low operational cost, reliability and flexibility, the single point mooring has been widely used. More than 100 SPMs are now in operation in all parts of the world: Most of them are of the type anchored to the seabed with long catenaries or chains. (SBM Fig. 11).

What does an SBM look like? Basically, it is a circular buoy with a diameter varying between 10 and 17m, anchored to the seabed by means of four, six or eight chain legs and fixed to the bottom either by conventional anchors, driven piles or drilled-in piles.
On top of the buoy, a turntable is mounted on a roller-bearing allowing a 360 degree rotation. This turntable is fitted with pipings, valves, mooring bit, floating hose connections, navigational aids and, in most cases, lifting equipment.
The center of the buoy body houses the central swivel essential for fluid transfer between fixed and rotating parts of the buoy.
Usually, the bottom connection to the pipeline manifold is made by one or more hose strings. Floats or buoyancy tanks are fitted to the underwater hose strings to obtain a smooth curve between the pipeline manifold and the buoy.
The tanker is moored to the turntable mooring lugs by thick nylon ropes. Oil transfer is by way of one or more floating hoses.
Another type in the SPM system is the Single Anchor Leg Mooring "SALM", in which the floating buoy is anchored to the bottom through one single anchor leg to a base type anchor point.
The oil is transferred from the manifold through a hose, then through a pipe connected to a riser. From there it flows through a fluid swivel and floating hoses to a tanker. (Fig. 12).
The anchor leg chain is connected to the mooring buoy and the fluid swivel by means of a universal joint.

II.6 OFFSHORE HOSE

As shown in the preceding diagrams SPM's make considerable use of large bore marine hose, "THE FLEXIBLE LINK". These are used, in submarine form, to convey cargo from the underwater sealine manifold to the buoy; and, in floating form, from the buoy to the ship's manifold.

The hose is generally constructed from a 'smooth bore' synthetic oil resistant rubber tube or liner, reinforced with multiple plies of helically wound high tensile textile cord or wire tyre cord. Above such primary reinforcement are usually two 'heavy' helical wires, alternatively discrete hoops or rings, which are embedded in a rubber matrix. A final cover of abrasion/weathering resistant synthetic rubber, alternatively polyurethane completes the construction, Figure (13).

In the case of floating hose, Figure (14) these generally are of the integral floated type, whereby a buoyancy medium is built as an integral element of the hose carcase. This is generally provided by closed cell expanded rubber sponge between the reinforcement sector and final hose cover.

Steel fittings or nipples are built into the hose at either end which are completed by carbon steel flanges, (slip on or weld neck) drilled to ASA 150, 300 or similar specification. These are used to couple individual hose lengths together to form the required hose string length.

Generally such hoses are 'designed' on a pressure rating criteria of 150-225 psi with appropriate safety factors relating to burst, typically 5:1.

Many variants of the preceding is of course possible, particularly for hoses performing specialist functions:-

Hoses attaching to the buoy, seabed manifold or similar semi-rigid structures often have additional stiffness reinforcement at the connection end tapering gradually over the hose length.

Other hoses provide location bands etc for the attachment of discrete buoyancy chambers or floats.

It may readily be seen that hose plays an extremely important role in the successful operation of the SPM concept.

II.7 THE PROBLEM

Oil pollution at sea has been a problem of international concern for many years. Its deleterious effects on the environment and on wildlife, fisheries, etc., are matters of major concern. In any operation where spillage might occur, every reasonable precaution must be taken to avoid such an occurrence.

The Torrey Canyon accident in March 1967 and the collision of the Pacific Glory in October 1970, which gave rise to localized disaster conditions, have highlighted in the public eye the potential hazards created by the increase in tanker traffic, and associated activity.

In spite of the many advantages of the SPM's they are viewed with some suspicion, particularly by envirormentalists. To some extent the worry is justified if viewed against the experience of particular loading sites.

The SPM system comprises all equipment from the end of the sea line to the tanker manifold. Excluding the buoy and the mooring equipment, the rest of the equipment consists of piping for the conveyance of the oil. Over 90% of this piping is flexible rubber hose.

This has been - and still is - regarded by many as the weak link of the system.

This is a wrong concept. What many observers have not appreciated is that the flexible-hose line could be described as the overworked section of the system. In the normal course of working within a dynamic environment it is the flexible line that has to compensate for movements in tanker position relative to the buoy and is still subject to its own reaction to wind and sea movement.

Frequently, these forces do not act in concert and the flexible lines are subject to severe axial, bending, or torsional forces as a consequence.

The criticism has often been applied that hose manufacturers have merely modified conventional submarine dock hoses to work in the quite different conditions that pertain to SPM's.

This was certainly true in the early SPM sites, though they were successful. Up to 1964 when the largest hose sizes in use were 12in bore, the troubles associated with flexible hose lines were minimal and were confined, mainly, to critical positions, such as the first hose off the buoy and ships' rail hoses.

With the advent of 16in bore hoses more problems were encountered, but again were mainly associated with the first hose connected to the buoy.

The introduction of 20in bore hoses gave some operators handling problems, particularly as 16in rail hoses were used in conjunction with 20in main lines. Nevertheless, most problems were still centered around the first hose off the buoy and the rail hoses.

Most of the hose manufacturers, at this time, were offering specially reinforced hoses for these critical positions. But the whole system was - and still is - in a continuous state of development. Buoys were being sited in more exposed locations. Throughputs were continually being increased and tanker turnaround was becoming more frequent.

Experience was beginning to point to the fact that sites needed to be considered in much greater depth than had originally been thought necessary, depending upon the peculiar conditions that applied in that specific location.

A particular design of hose that was being successfully used at one installation could prove quite unsuitable for another which experienced different wind, sea or operational conditions.

A variety of underbuoy-line configurations was considered at one time or another, to be the most suitable. The height of the buoy manifold relative to the sea surface was varied within wide limits and the angle of the pipe-flange connections to the first hose off the buoy has been varied from 45° to 90° to the horizontal, with varying degrees of success depending upon the conditions peculiar to the site.

Very often, when a modification made to a system has not been a success, the first indication of this has been exhibited by the failure of another of the so-called "weak links" which has eventually fatigued due to the overstressing which had been induced.

Due, in large measure, to lack of information on the actual forces to which hoses would be subjected in service, hose design has been biased towards providing hoses with ever increasing bursting strength and the properties usually associated with reinforcing a hose in this direction, such as low elongation characteristics.

While strength - both radially and axially - is an obvious asset, it has to be viewed in the context of its effect on flexibility and fatigue life, both of which are not necessarily enhanced and could even be lessened by a mere increase in resistance to burst.

Surprisingly in view of the possible environmental problems, cost has been a major deterrent in the development of better, more-reliable hoses. It still is to some extent.

With the advent of 24in bore hoses, a more random pattern of failure has been observed.

This is not too surprising since the margin of safety - over and above the built-in minimum design requirements - tends to lessen with the increase in size. Of more importance is the fact that these products are still, essentially, handmade and can consist of upwards of 100 separate manufacturing operations. Quality control in this type of operation has depended too heavily in the past upon the individual skills of the operative. Final inspection methods are prescribed by the various relevant specifications are not sufficiently searching to separate the good hose from the not so good.

Nevertheless, the need for design improvement is as imperative as ever. Apart from the higher performance requirements that are being demanded there must also be an insurance factor to reduce the risk of pollution, even in the case of hose failure.

Geologists hold that the quantity of ultimately recoverable oil in the world could reach 300 billion tons. Forty billion tons have so far been produced and proved reserves amount to 90 billion tons. This leaves some 179 billion tons to be discovered mainly in the oceans and under the Arctic cap.

It is well known that since October 1973, oil prices have quadrupled and that for many years, consumption has been increasing by almost 4 percent annually. Both prices and consumption are bound to escalate further, creating a strong incentive to explore and produce oil from deeper waters on the one hand and from under the ice of the Arctic, on the other.

We are therefore entering a new era in oil exploration and supporting industries will have to adapt themselves to the new demands of the market. Our own industry is bound to follow and will have to develop equipment to handle oil extracted from ever greater depths and under even more severe environmental conditions than those encountered todate.

	PRODUCTION	CONSUMPTION	DIFFERENCE
EUROPE	375,600 b/d	11,810,000 b/d	-11,434,400 b/d
MIDDLE EAST	13,718,100 b/d	1,002,000 b/d	+12,716,100 b/d
AFRICA	6,359,200 b/d	918,000 b/d	+ 5,441,200 b/d
ASIA - PACIFIC	1,440,400 b/d	6,347,000 b/d	- 4,906,600 b/d
USA - CANADA	10,749,700 b/d	15,850,000 b/d	- 5,100,300 b/d
LATIN AMERICA	5,171,600 b/d	2,786,000 b/d	+ 2,385,600 b/d
EASTERN EUROPE & CHINA	7,549,400 b/d	6,637,000 b/d	+ 912,400 b/d
TOTAL	45,364,000 b/d	45,350,000 b/d	+21,455,300 b/d
			-21,441,300 b/d
			+ 14,000 b/d

TABLE I TOTAL OIL PRODUCTION AND CONSUMPTION IN 1970

	SBM	TOWER	MBM	FIXED BERTH
INVESTED CAPITAL*	$ 8,500,000	$ 10,500,000	$ 5,500,000	$ 25,000,000
THEORETICAL OCCUPANCY	347 days	340 days	306 days	300 days
FILLING TIME	13.5 hrs	13.5 hrs	13.5 hrs	13.5 hrs
TIME AT TERMINAL	15.5 hrs	16.5 hrs	18.5 hrs	16 hrs
TIME BETWEEN TWO SHIPS	17.5 hrs	18.5 hrs	20.5 hrs	18 hrs
NUMBER OF SHIPS OCCUPANCY 50%	238 ships	220 ships	179 ships	200 ships
QUANTITY OF OIL MILLIONS TONS/YR	23.8	22	17.9	20
OPERATIONAL COSTS				
DEPRECIATION	$ 850,000	$ 875,000	$ 458,000	$ 1,250,000
MAINTENANCE	$ 300,000	$ 250,000	$ 200,000	$ 100,000
ASSISTANCE TUG & SERVICE BOAT	$ 50,000	$ 300,000	$ 300,000	$ 300,000
PERSONNEL	$ 100,000	$ 100,000	$ 80,000	$ 50,000
TOTAL	$ 1,300,000	$ 1,525,000	$ 1,038,000	$ 1,700,000
OPERATIONAL COST $/TON	0.054	0.069	0.058	0.085

* Invested capital refers to the supply and installation of mooring equipment and a sealine 3 kilometers long.

TABLE II : OPERATIONAL COSTS FOR DIFFERENT TERMINALS ON THE BASIS OF A 100,000 D.T TANKER AND A THROUGHPUT OF 7500 TONS/HR

Table 3 shows a comparison between operational costs for different tanker sizes ranging from 50,000 dwt to 300,000 dwt and throughput from 5000 tons/hr to 20,000 tons/hr.

Table 3

Comparison of Operational costs for different types of terminals and different sizes of tankers.

TANKER SIZE DWT	THROUGHPUT TONS/HR	SBM Cents/Ton	TOWER Cents/ton	MBM Cents/ton	FIXED BERTH Cents/ton
50,000	5,000	6.8	9.2	6.7	13
100,000	7,500	5.4	6.9	5.8	8.5
150,000	10,000	4.2	5.5	7.7*	6.7
200,000	15,000	3.5	7.6**	-	5.7
300,000	20,000	4.6***	-	-	4.5

* For two berths each for 5,000 t/hr

** For two towers each for 7,500 t/hr

*** For two SBMs each for 10,000 t/hr

TABLE III : COMPARISON OF OPERATIONAL COSTS FOR DIFFERENT TYPES OF TERMINALS AND DIFFERENT SIZES OF TANKERS

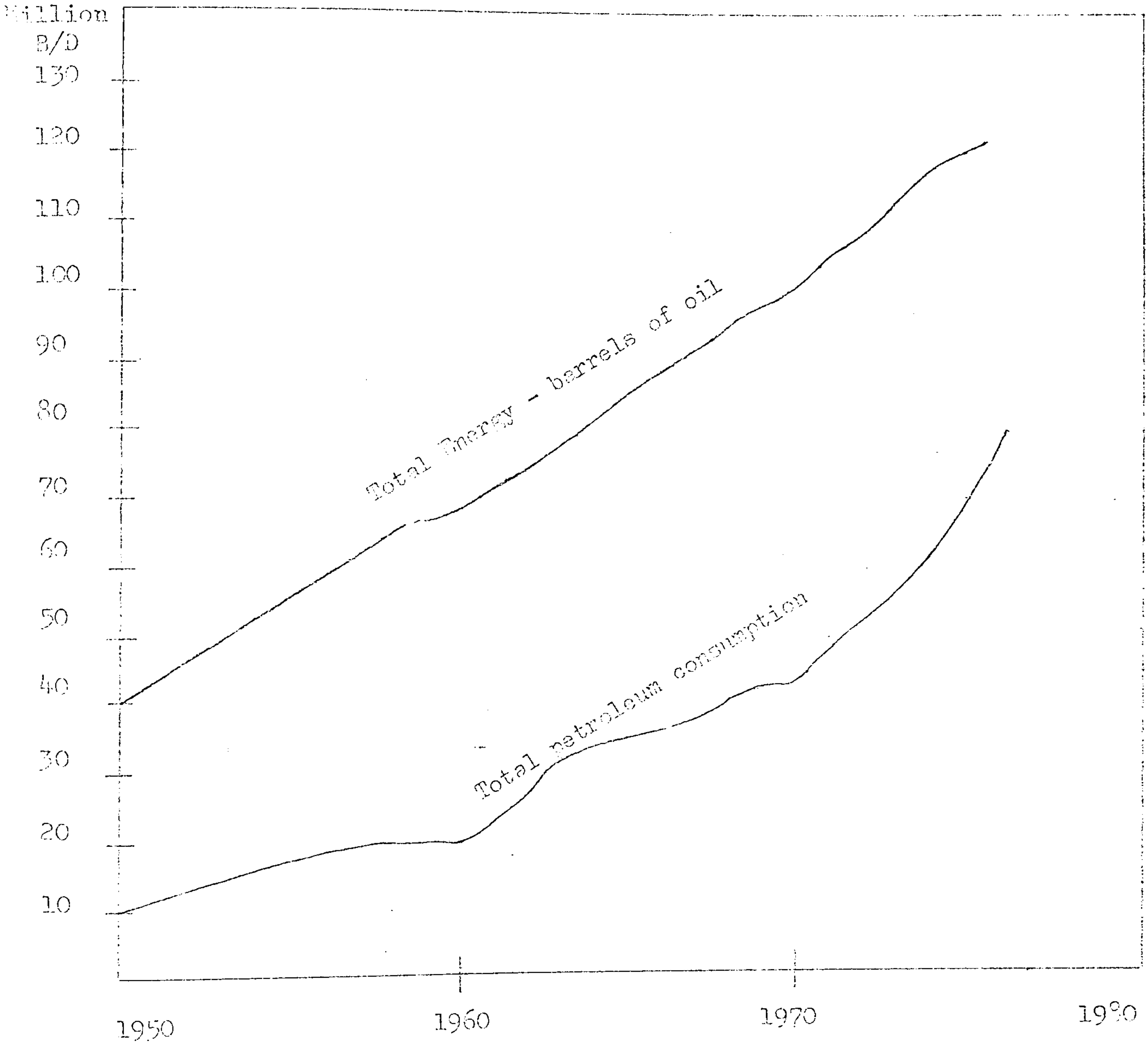

FIGURE I TOTAL ENERGY AND OIL CONSUMPTION FROM 1950-1974

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III DEFINITION OF THE PROBLEM (INITIAL INVESTIGATION)

III.1 INTRODUCTION

This project was initiated by Dunlop Limited, Oil and Marine Division because they considered a gap existed in hose design philosophy and that this was indicative of the current problems being encountered by themselves and the offshore hose industry in general.

Taking an overview of the situation the problems experienced could perhaps be most graphically described by Figures (15) and (16) - hose failure and resultant pollution. Typical costs of such a situation are examined by Table (4). Fortunately most current SPM installations were in remote areas of the world where oil spillage, albeit expensive to clear up, did not carry the political consequences, (compensation etc), as such an occurrence around civilisation centres, ie., Europe, USA etc. However the move to install SBM's in such "dangerous waters", eg., Anglesey, prompted greater concern in this respect.

Initially the problem in the context of this project was not readily definable in concrete terms. The following broad areas, put forward by Dunlop Limited, formed the basis of the problem definition.

a) A suspected increasing failure rate in respect of increasing bore size, (ie nominally 20in and 24in bore sizes).

b) Insufficient input design data upon which to base design modification or future designs.

c) Given that the above information could be made available, few design calculations or analysis techniques were historically available to the industry in which to make use of the available data.

d) Inadequate test methods, procedures, or indeed equipment existed within the industry in order to successfully evaluate revised hose design; or even existing designs for that matter. (Practice within the industry had previously been to place prototypes directly into service and then await results!)

The initial work within the project framework was thus to define the problem in rather more precise scientific terms.

III.2 HOSE FAILURE STATISTICS

The objective of this initial study was to determine the type and magnitude of large bore SBM hose failures.

As a result of many years of trading, considerable information in the form of customer feedback, failure reports, site visits etc was already available. This was supplemented by discussions with a number of terminal operators and the resultant information correlated.

A summary of the general failure statistics is shown in Table (5). Figure (17) is a map showing the graphical breakdown of the failure statistics tabulated.

The failure data described is for all hose failures during the past several years, some reported failures may be for hose models no longer in production, however to show general trends no distinction is made for hose models.

General conclusions which may be made from the data are as follows:-

In most locations underbuoy hoses have a longer life than surface hoses, the average underbuoy replacement rate being generally in excess of two years.

The first off the buoy hoses averaged ten months between replacements. It should be noted however that some of the replacement frequencies represent scheduled replacements and not necessarily hose failures. The average service lives of failed hoses were in fact slightly over seven months. First off the buoy hoses comprise 15% of the total number of failures reported while they only account for 3% of hoses in the floating hose string. Main line floating hoses were replaced at an average of every twenty-one months with the range being 6-48 months. Tail/rail hoses had similar figures of 12 months and 2-48 months respectively. (NB: Area (5) Table (5) reporting longer life cycles was not subject to severe environmental conditions).

III.3 FAILURE CLASSIFICATION

Hose failures were classified into the following failure types:

a) Hose fracture or breakage;

b) Nipple leak;

c) Kinking;

d) nipple pullout;

e) liner collapse;

f) abrasion of covers;

Hose fracture or breakage can be the result of almost any loading flectural fatigue, excessive internal pressure, bending tensile loads or impact by a boat can all cause this type of failure. Quite often nipple leakage, described in a later paragraph precedes the fractural breakage especially where the flectural fatigue is suspected to be a cause.

Nipple leakage has occurred at every position in the hose string, the nipple liner area is one where dissimilar material bonding causes the weakest adhesion. When nipple liner adhesion is weak, oil will work its way along the nipple until it finds its way out. Instances of this have been reported immediately after a receipt of hose and others after having been in service for a period of time. The second types of nipple leak occur when there is a tearing of the liner just down-stream of the nipple end, oil penetrates through the liner and oozes along the reinforcing or breaker plies. Evidence of this has been found when leaking hoses were sectioned and oil found in the interior of the hose all along its length. Other evidence of this condition is a swelling or bulging of the hose liner and could be easily found by visible inspection.

Kinking can occur in any hose which is bent beyond its elastic limit, that is beyond its ability to recover. Kinking failures occur most often on first off the buoy hoses, over the rail hoses and submarine hoses. First off the buoy kinking occurs usually just down-stream from the extra reinforcement peculiar to that type of hose, it should be noted that kinking often occurs during handling and installation. The danger caused by kinking is sometimes evident by hose fractures after some service life.

Nipple pullout occurs either through a failure of the binding wire, wire breakage or incorrect tension or excessive bending and tensile forces being applied to the hose.

Liner collapse is a separation of the liner from the rest of the hose. This can be caused by poor adhesion, oil penetration or excessive vacuum for long periods of time or a combination of these conditions.

Abrasion can occur in submarine hoses rubbing on the sea floor, submarine or floating hoses rubbing against each other or some foreign object rubbing against the hose. Results of this condition are the wearing away of the outer cover exposing the interior of the hose, this results in a weakening of the hose and eventual failure.

In order to assess the failure picture on a hose type basis, the failures known were split into hose type groups and assessed as to the mode and possible cause of failure.

Table (6) is a summary of failures reported on main line hose; in this table only the large size hoses are included, ie 20in and 24in. It may be noted that failure types not attributed to handling are primarily cracks or fixtures and seem to occur six to twenty-one months after installation. The three areas of shortest life circle, namely two, nine and ten, all have steady winds and climatic conditions inducive to a short choppy sea causing constant motion in the hose strings. Other conditions which take their toll on main line hoses are handling of the hose and boat handling around the buoy.

First off the buoy hose failures, Table (7) exhibited a significant proportion of the failures occurring at the buoy nipple. This is not surprising of course as this hose nominally forms the "elastic link" between two systems of vastly different motion characteristics. As stated previously the failure incidence is a significantly greater proportion than the related hose population.

Rail and tail hose failures are shown by Table (8). The average service life of rail hoses is 7.4 months within the range of 1-25 months. As would be expected most failures occurred at approximately midbay of the hose, ie., the section bent over and in contact with the ship's rail.

It would probably be beneficial at this stage to review the requirements of hose design, and the basic hose construction itself.

Offshore loading and discharging hoses are subject to many external forces. Service, environmental and handling are the general classifications assigned to these forces.

Service forces are those experienced during the time a hose is being used. Internal pressure, internal pressure surge, vacuum and flow velocity are the four primary service loadings.

Internal pressure is used as a criteria for the rating of hoses, however pressure appears to be the least harmful.

Internal pressure surge is a condition which occurs when the receiving station experiences a sudden valve closing or line restriction causing the flow rate to drop and the pressure to rise. Surges of this nature in excess of 500 psi have been reported. Some operators have reported hose failures occurring during surge. It has been reported that some sites have installed or are in the process of installing surge tanks for pressure safety devices to alleviate the problem.

Vacuum can occur under three conditions, the first is a natural vacuum which occurs when a loading facility shuts down its pumps and the velocity head of the string continues to move causing a void condition or closes its valves when gravity loading. Actual values have not been measured but since they are of short duration their effects are minimal. The second type of vacuum is purposely applied to drain the ship end of the hose so oil spillage will be lessened or eliminated when disconnection of the flange occurs. This vacuum condition usually lasts from two to five minutes and normally effects the floating string up to the buoy. The third type of vacuum condition is brought about after the fluid is pumped down and the hose is capped. As the oil/air left in the hose cools down, the fluid contracts and since the passage is sealed off a vacuum occurs. This vacuum varies depending on the vapour pressure and volatile content of the fluid enclosed but has been reported by one operator to be as high as 20" - 25" of mercury. The volume effected can be 75 cu.ft or more (1,300 ft of 24" diameter hose cooling down from 120°F to 85°F).

Environmental loadings are those caused by wind, wave and current conditions and these are some of the most variable. Conditions vary from calm to rough and to areas where general calm conditions occur but very violent short duration storms occur frequently.

Wind, wave and current act in conjunction to produce cyclic forces which set up fatigue loading in the hose. These forces are most evident on the first off the buoy hose, first underbuoy and first off the PLEM. All of these hoses represent a significant change from a very rigid system to a more pliant one.

Waves need to be further classified. There are two types, short period "choppy" waves and long period "rolling" ones. The short choppy waves produce unsupported bending, hence are more detrimental. A long rolling sea will cause the hose to flex gently and smoothly usually providing continuous support.

Improper installation can impose far greater loads on hoses than can be imposed by the most severe environmental conditions. Excessive tensile bending and Kinking are the most common occurrences. Stretching of hoses due to excessive tensile loads can effect both the hoses ability to resisting external pressure and internal pressure. Bending and kinking can occur at any time if proper safeguards are not taken. Lifting from storage by forklift or improper strap arrangement, incorrect ballasting during submarine hose installation, improper lifting and handling equipment and improper storage can severely weaken hoses. Operational handling loads occur primarily on the rail and tail hose sections as the hoses are lifted to manifold level. These loading conditions are very difficult to control or predict as there are so many variations in manifold lifting capabilities and manifold placement, while excess tensile loads are rare, bending and kinking are major replacement criteria. (Handling criteria are however not generally within the control of the hose manufacturer, apart from recommendatory action, and thus such factors are outside the scope of this study).

III.4 SERVICE LOAD DATA

In order to design structural elements or components (a hose may be classed as such), a knowledge of the magnitude and statistical occurrence of service forces is essential.

Because in the past few people, particularly hose manufacturers, had treated hoses as structural members, a knowledge of service loads was not sought. Alternatively the subject was considered to be of such complex magnitude that no work had apparently been conducted. Because of the relatively small number of design personnel within the total hose industry it was readily established that service load data was virtually non existent. Equally non-existent was the manufacturers knowledge of the capabilities of his own design, eg., tensile strength, stiffness characteristics etc.

An initial survey relating to load magnitude as a value judgement by various personnel; operators, designers, consultants etc., proved to be perplexingly inconclusive. Such assessments differed not only by degree, (this could have been explained by varying operating conditions world-wide), but by order of magnitude.

Various research bodies, e.g. Netherlands Ship Model Basin, Exxon Research Engineering etc., claimed that they had or could calculate such figures from such input as wave spectra energy diagrams. However no results were forthcoming and certain other bodies threw much doubt on others abilities to meaningly conduct such work. Even had the existence of such work been proven certainly no party had conducted any onsite load measurement in order to validate such figures.

A need for the collection of such data was therefore obviously apparent.

III.5 HOSE DESIGN PARAMETERS

Generally the only characteristic required of hose by mandatory specification, (British Standards, B.M.F. etc) is that of the ability to resist internal pressure with "suitable" safety factor. (ie: usually 225 lbf/in^2 working, 5:1 safety factor, resulting in a 1225 lbf/in^2 burst pressure). Consequently hose has historically been designed on this burst pressure requirement.

Burst pressure - In order to resist internal pressure plies of fabric or wire cord are applied at some angle to the hose axis. Historically the formula used was:-

Burst pressure =

$$\frac{2 \times \text{sine(helix angle)} \times \text{strength of cord(per end)} \times \text{No.ends} \times \text{No.plies}}{\text{Mean application dia.} \times \text{helix pitch}}$$

It may be noted that only these primary elements are considered as contributing hoop strength.

Tensile strength - As a result of the above compilation a similar relationship is often quoted for tensile strength. Generally this feature is a resultant of the burst pressure design and not a design consideration itself:-

Tensile strength =

No. Ends x Strength of cord (per end) x Cosine (Helix angle)

Both of the above formulli yield only an extremely crude approximation to the properties in question. The latter is particularly suspect for a great number of reasons which will be discussed later. In order for either formula to apply it may readily be seen that a major assumption applies in that the application angle of the reinforcement is assumed to remain constant. This generally has been accepted within the industry based upon a theory termed the "braiding angle", (8).

Other design parameters frequently used relate to the heavy helical wires embedded in a rubber matrix above the main reinforcement layer:-

1. To give the hose resistance against external pressure, whether this be caused by submerging the hose or subjecting the hose to internal vacuum. The formula used was derived from an empirical formulation devised by Prof. R T Stewart in 1906 for the similar collapse of steel tube, (9).

2. To give the hose resistance to direct crushing forces as might apply when bent over a ship's rail etc, (10).

Nominally these were the only design formulations generally used in determining basic hose constructions. It may readily be seen that all are extremely suspect in operation and to the most part irrelevant. No attempt had been made to assess any other nominal structural property, stiffness, modulus etc., let alone design for such a property or series of properties.

III.6 TEST REQUIREMENTS

Buoy mooring forum hose standards have only two structural engineering requirements, a five to one safety factor based on hose working pressure rating and a vacuum test, nominally the ability to withstand 30" Hg without liner collapse. Manufacturers design hoses to meet these pressure test requirements, however it appears that these might be in direct conflict with fatigue life. At present no test or requirement exists which can evaluate hose life under combined loading or requirement in hose engineering design effort to increase combined loading fatigue life. In order to place the effectiveness of the present tests into perspective the following example is quoted;

"During the past four years Dunlop have manufactured approximately 3,000 large bore hoses, not one has failed the above statutorial tests".

Apart from the quoted statutory tests other tests are occasionally conducted by manufacturers:-

Bend Tests: Such tests were conducted in order to "assess" hose flexibility. Generally they consisted of either fixing one end of the hose to a stanchion and dragging the other end around using a winch or using a beam and tirfor arrangement, both of which are shown by figure (19), the hose being supported on trollies along its length. The tests thus conducted were however of little immediate benefit, test methods/conditions were variable and thus affected the results, in addition measurement of test parameters ie: bending moment/radii of curvature (or deflection)/and internal pressure were frequently neglected. Generally the test was of qualitative value, the prime function being to bend the hose to it's "minimum bend radius" (the only measured parameter), and demonstrate that no (noticeable) permanent damage resulted. In the writer's view much benefit and knowledge could have been gained had these tests been properly conducted and results correlated.

Such obvious lack of constructive knowledge may be gleaned from manufacturers published "minimum bend radii" figures,

BORE SIZE	M.B.R.
16"	8'
20"	10'
24"	12'

ie: although such hoses are of different detail designs/materials the M.B.R. is determined by expressing the nominal hose radius (in inches) as the same figure in feet. Not a very scientific process!

Elongation Tests: "Measurement of the hose's temporary elongation under working pressure". Such a test is recommended as a service test for assessing the life status of a hose in a planned maintenance scheme. (eg: If % elongation = 2 x % elongation during factory test = hose retirement). The effectiveness of such a test is open to extreme suspicion, no valid evidence is available to support the hypothesis and infact much exists to discredit it. No account is made of any permanent change in length (either compression or elongation), and not infrequently hoses have been known to exhibit less temporary elongation immediately prior to failure than that measured during the factory test.

Again the test method is one inherited from the historical traditions of the small bore hose industry and rubber processing in general, both of which are not renowned for their technological approach!

III.7 PROBLEM SUMMARY

Based upon the initial survey/study of the problem areas projected by Dunlop the following conclusions were made based upon the data available.

a) Failure of hoses in the 20" to 24" bore range was unacceptably high. There being no apparent change in failure rate with either time and/or experience.

b) Such hoses show a tendency towards premature failure four to eighteen months in areas where nominal World-wide sea conditions exist. A nominal sea is defined as one where the significant wave over one third of the time is 2-3 metres.

c) The first off the buoy and over the rail hose appear to have service lives averaging one to twelve months. Rail hose failure data indicates that failure rate is relatively insensitive to environmental conditions but very sensitive to handling facilities.

d) Most first off the buoy hose failures occur within two diameters of the nipple. Observed failures were similar in different areas indicating the same forces responsible for the failures. The mechanism for failure appears to be flextural fatigue.

e) Most hoses in use are constructed in a similar manner and most probably using similar equipment.

f) In service loads information is seriously lacking with apparently no organisation (manufacturer or user) willing to take the initiative towards filling this gap.

g) Design technology is seriously lacking within the industry, with manufacturers steps towards product improvement seemingly directed toward material variations and construction techniques rather than being design or functionally orientated.

h) Physical properties data on hoses is incomplete and more often than not inconsistent, ie: weight, stiffness etc.

i) Statutory inspection tests (pressure and vacuum) are virtually meaningless and indeed existing design philosophy and specifications serve to limit or impede hose improvement.

j) No method exists to meaningfully evaluate hose design/design changes either on a static characteristic or more importantly a fatigue basis.

TABLE 4

TYPICAL COSTS OF FAILED HOSE 'CHANGE - OUT'

		£
HOSE REPLACEMENT VALUE	(eg. 35' x 24")	10,000
Oil Tanker demoorage	(250,000 d wt)	20,000 / day
Service Barge		1,000 / day
Pollution Clean - up		1,000 / Ton

TOTAL COST

£75,000 (Assuming weather hold-ups, minimum pollution)

£500,000 (Assuming 14 day weather delay, moderate - severe pollution).

TABLE 5

HOSE FAILURE DATA SUMMARY

ITEM / OPERATOR	1	2	3	4	5	6	7	8
I. Site Information								
A. Buoy Type	CALM	CALM	CALM	CALM	CALM	CALM	CALM	CALM
B. Water Depth (ft)	250	50-100	50-100	50-100	162	50-100	50-100	100-150/>150 (4)
C. Current (k) - Avg/Max	.5/.5	2-3/4	1-2/1.5	1-2/3	.5/.5	1-2/na	73/3.4	.5-1/1.5-2
D. Wind Speed (k) - Avg/Max	16-20/80	720/75	11-15/63	16-20-50	5-10/40	16-20/60	11-15/75-80	11-15/45
E. Wave Height (ft) - Avg/Max	<8/50	78/22	2-4/16	4-8/12	2-4/6	4-8/12-15	4-8/18-20	>8/25
II. Hose Information								
A. Floating Hose								
1. No. of hoses								
a) FOB*	3	2	2	2	2	2	1	4
b) Mainline	6	60	42	47	50	52	22	90
c) Tail	0	2	14	6	2	2	2	4
2. Avg Replacement (mo)								
a) FOB*	na	5	2	3 (2)	48	8	11	9
b) Mainline	na	6-1/2	27	12 (2)	48	48	24	24
c) Tail	na	2	24	6 (2)	48	12	24 (3)	4-6
B. Underbuoy Hose								
1. Hose Configuration	Lazy-S	Lazy-S	Chinese Lantern	Chinese Lantern	Lazy-S	Lazy-S	Lazy-S	Lazy-S
2. Avg Replacement (mo)	na	12-24	<24	6-12	>24	12-24	12-24	>24
3. Flotation	Buoyancy Tanks	Buoyancy Tanks	Bead	Bead	Buoyancy Tanks	Buoyancy Tanks	Buoyancy Tanks	Buoyancy Tanks

*FOB - First off-buoy

TABLE 5 (CONT'D)

ITEM / OPERATOR	9	10	11	12	13	14	15	16
I. Site Information								
A. Buoy Type	CALM	CALM	CALM/SMB	CALM	CALM	CALM	na	na
B. Water Depth (ft)	100-150	100-150	50-100	Over 150	100-150	100-150	50-100	50-100
C. Current (k) - Avg/Max	.5-1/1.2	1-2/2	2-3/>3	2-3/3-4	2-3/3.5	.5-1/1.5	2-3/6	1-2/2.5
D. Wind Speed (k) - Avg/Max	11-15/45	11-15/55	5-10/>20	11-15/55-60	>20/70	11-15/65	2.5-10/80	16-20/55
E. Wave Height (ft) - Avg/Max	>8/12	>8/14	4-8/12	>8/10-12	>8/35-40	4-8/15	2-4/10	2-4/16
II. Hose Information								
A. Floating Hose								
1. No. of Hoses								
a) FOB	9	5	na	1	1	1	na	2
b) Mainline	234	81	na	24	22	24	19	13
c) Tail	33	15(6)	na	1	2 Strings of 4	2	na	na
2. Avg Replacement (mo)								
a) FOB	5	7.4	12-14	(7)	6	6	na	na
b) Mainline	6	12+	24-36	(7)	18	na	6	na
c) Tail	4	8.7	6	(7)	12	na	na	na
B. Underbuoy Hose								
1. Hose Configuration	Lazy-S	Lazy-S	Chinese Lantern	Lazy-S	Lazy-S	Lazy-S	na	Lazy-S
2. Avg Replacement (mo)	6-12	12-24	12-24	(7)	12(8)	48	6-12	na
3. Flotation	Bead	Combination	Bead	Buoyancy Tanks	Buoyancy Tanks	Buoyancy Tanks	Buoyancy tanks	Buoyancy Tanks

TABLE 5 (CONT'D)

Item \ OPERATOR	17	18	19	20	21	22	23	24	25
I. Site Information									
A. Buoy Type	CALM	CALM	CALM	CALM	CALM	CALM	CALM	CALM	CALM
B. Water Depth (ft)	50-100	50-100	50-100	50-100	50-100	50-100	<50	50-100	50-100
C. Current (k) - Avg/Max	.5-1/3	<.5/.5	.5-1/1	.5-1/1.5	.5-1/1.5	.5-1/1	1-2/1	1-2/2	1-2/2
D. Wind Speed (k) - Avg/Max	11-15/60	11-15/60	11-15/60	11-15/70	5-10/73	5-10/54	5-10/30	16-20/56	16-20/54
E. Wave Height (ft) - Avg/Max	2-4/10	2-4/11	>8/10	>8/10	2-4/9	>8/6	4-8/3	4-8/6	2-4/10
II. Hose Information									
A. Floating Hose									
1. No. of Hoses									
a) FOB	3	3	7	2	6 6	5	2	14	7
b) Mainline	18	13	16	44	36 40	21	2	34	16
c) Tail	2	6	5	8	6 6	5	2	10	5
2. Avg Replacement (mo)									
a) FOB	18	24	36	12	48	12	12	24	24
b) Mainline	12	60	60	48	48	60	12	24	48
c) Tail	11	48	24	24	48	12	12	24	48
B. Underbuoy Hose									
1. Hose Configuration	Chinese Lantern	Chinese Lantern	Lazy-S	Lazy-S	Lazy-S	Chinese Lantern	Chinese Lantern	Lazy-S	Lazy-S
2. Avg Replacement (mo)	12-24	>24	>24	>24	>24	>24	12-24	12-24	>24
3. Flotation	Bead	Bead	Bead	Bead	Bead	Bead	na	Combination	Combination

TABLE 5 (CONT'D)

ITEM / OPERATOR	27	28	29	30	31
I. Site Information					
A. Buoy Type	CALM	CALM	CALM	SALM	CALM
B. Water Depth (ft)	50-100	50-100	50-100	na	50-100
C. Current (k) - Avg/Max	.5-1/1	1-2/3	.5-1/1	na	.5-1/2
D. Wind Speed (k) - Avg/Max	11-15/40	5-10/76	11-15/60	na	5-10/30
E. Wave Height (ft) - Avg/Max	78/10	78/42	>8/10	na	4-8/8
II. Hose Information					
A. Floating Hose					
1. No. of hoses					
a) FOB	1	1	7	na	2
b) Mainline	24	18	16	24	38
c) Tail	1	1	5	6	4
2. Avg Replacement (mo)					
a) FOB	36	6	36	-	12
b) Mainline	36	48	60	18	36
c) Tail	36	48	24	-	6
B. Underbuoy Hose					
1. Hose Configuration	Chinese Lantern	Lazy-S	Lazy-S		Lazy-S
2. Avg Replacement (mo)	>24	>24	>24	30(9)	>24
3. Flotation	Combination	Combination	Combination		Buoyancy Tanks

NOTES
(1) Not available
(2) Scheduled removal - not failure.
(3) Rail hose 6-12mo
(4) Two buoys
(5) Total of 3 SPM
(6) Six at OTT, Nine at FSV
(7) None changed after nine months of operation.
(8) Scheduled replacement.
(9) Underwater buoy

HOSE FAILURE SUMMARY DATA

MAINLINE HOSES (FLOATING) (20" AND 24" ONLY)

OPERATOR	HOSE DIAMETER (INCHES)	TIME IN SERVICE (MONTHS):	FAILURE TYPE	FAILURE LOCATION (HOSE LENGTH)	NUMBER OF OCCURRENCES
2	24	9.4	Crack or Fissure	At Nipple	3
	24	11.0	Crack or Fissure	At Nipple	5
	24	3.5	Crack or Fissure	At Nipple	5
6	20	27.0	Nipple Leak	At Nipple	1
	20	7.0	Handling Kink	Mid Bay	1
	20	35.0	Nipple Leak	At Nipple	1
	20	39.0	Hole	Mid Bay	1
	20	43.0	Hole	At Nipple	1
	20	18.0	Throughput	na	1
	20	35.0	Hole and Crack	Mid Bay	1
	20	40.0	Ship Broke Out	na	1
	20	48.0	Throughput	na	3
	20	48.0	Throughput	na	4
8	24	21.0	Crack or Fissure	Nipple/Hose Interface	40
9	24	5.5	Nipple Leak	At Nipple	3
	24	6.0	Kink	At Nipple	1
	24	7.5	Liner Failure	Nipple/Hose	2
10	24	6.5	Crack or Fissure	Mid Bay	1
	24	6.0	Liner Failure	Mid Bay	3
	24	12.0	Liner Failure	Mid Bay	5
	24	12.0	Liner Failure	At Nipple	4
	24	10.0	Nipple Pull Out	At Nipple	1
	24	5.5	Liner Failure	Mid Bay	3
	24	1.0	Nipple Leak	At Nipple	1
	24	4.0	Kink	At Nipple	2
11	24	7.0	Cut	Mid Bay	1
	24	18.0	Crack or Fissure	Nipple/Hose Interface	1
	24	na	Excess tension	All	24
12	24	6.0	Abrasion	6ft from nipple	3
13	24	10.0	Crack	Hose/Nipple Interface	1
	24	11.0	Crack	Hose/Nipple Interface	1
20	20	17.0	Hole	Mid Bay	11
	20	14.0	Nipple Leak	Nipple/Hose Interface	2
	20	15.0	Delamination	Nipple/Hose Interface	11

HOSE FAILURE SUMMARY DATA

FIRST OFF BUCY HOSES

(POSITION 1)

OPERATOR	HOSE DIAMETER (INCHES)	SERVICE LIFE (MONTHS)	FAILURE LOCATION (HOSE LENGTH)	FAILURE TYPE	TOTAL FAILURES (THIS TYPE)
2	24	13.0	Nipple	Hole, Crack, Delamination	1
	24	7.0	Nipple	Hole, Crack, Delamination	1
	24	4.0	Nipple	Hole, Crack, Delamination	1
	24	1.5	Nipple	Hole, Crack, Delamination	1
	24	1.5	Nipple	Hole, Crack, Delamination	1
3	20	2.0	Mid Bay	Kink	8
	16	2.0	Mid Bay	Kink	5
4	20	12.0	Mid Bay	Crack	1
6	20	10.0	Nipple/Hose Interface	Hole, Nipple Leak	4
	20	1.0	Mid Bay	Hole, Abrasion	1
	20	9.0	Nipple	Nipple Leak	1
	20	33.0	Nipple/Hose Interface	Nipple Leak	1
	16	7.0	Nipple/Hose Interface	Handling kink	1
7	24	6.0	Mid Bay	Abrasion	
	24	8.0	Nipple/Hose Interface	Crack	
	24	3.0	Nipple/Hose Interface	Hole	
	24	6.5	Nipple/Hose Interface	Hole	
8	24	9-12	Nipple/Hose Interface	Crack	5
	16	12.0	Nipple/Hose Interface	Crack	4
9	24	4.0	Nipple	Nipple Leak	
	24	5.5	Nipple	Nipple Pullout	
10	24	7.0	Nipple	Handling Kink	3
11	24	4.0	Nipple	Rupture	1
	24	1.0	Mid Bay	Cut	1
	24	8.0	Nipple	Leak	1
	24	4.0	Nipple	Split	1
	24	14.0	Nipple	Split	1
13	24	4.0	Nipple/Hose Interface	Crack	6
	24	3.0	Nipple	Nipple leak	1
20	20	2.0	Nipple/Hose Interface	Nipple Leak	4
	20	24.0	Mid Bay	Abrasion	1
	20	8.0	Nipple/Hose Interface	Handling Kink	4

TABLE 6

HOSE FAILURE SUMMARY DATA

TAIL/RAIL HOSES (12" AND 16")

OPERATOR	HOSE DIAMETER (INCHES)	SERVICE LIFE (MONTHS)	FAILURE TYPE	FAILURE LOCATION	NO. OF FAILURES REPORTED.
2	16	1.0	Cut		1
		5.0	Kink and Crack	Nipple	1
		6.0	Crack	Mid Bay	1
6	16	8.0	Crack, Abrasion	Mid Bay	2
		7.0	Crack and Abrasion	All Over	1
		6.0	na	na	2
		16.0	Crack and Abrasion	na	1
		17.0	Liner Failure	Nipple/Hose Interface	1
7	16	11.5	Abrasion	Mid Bay	1
		8.0	Kink	Mid Bay	1
8	16	5.0	Crack	Nipple/Hose Interface	9
		3.0	Throughput	Nipple/Hose Interface & Mid Bay	1
9	16	4.0	Liner	Mid Bay	2
		6.5	Liner	Mid Bay	1
		7.0	Flotation Material	Mid Bay	1
		6.0	Liner	Mid Bay	3
		6.5	Abrasion	Mid Bay	1
		7.0	Liner	Nipple/Hose Interface	2
11	12	4.0	Rupture	Mid Bay	3
19	16	5.0	Kink	Mid Bay	8
		10.0	Kink	Mid Bay	1
20	16	24.0	Kink	Mid Bay	3
		12.0	Crack	Mid Bay	1
		12.0	Nipple Leak	Nipple/Hose Interface	2
		3.0	Nipple Leak	Nipple/Hose Interface	-
		24.0	Delamination	Nipple/Hose Interface	2
25	16	24.0	Liner Failure	Mid Bay	2
		24.0	Time Up	-	3
28	16	10.0	Kink	Mid Bay	2
		9.0	Abrasion	Near Nipple	1
		25.0	Kink	Mid Bay	2

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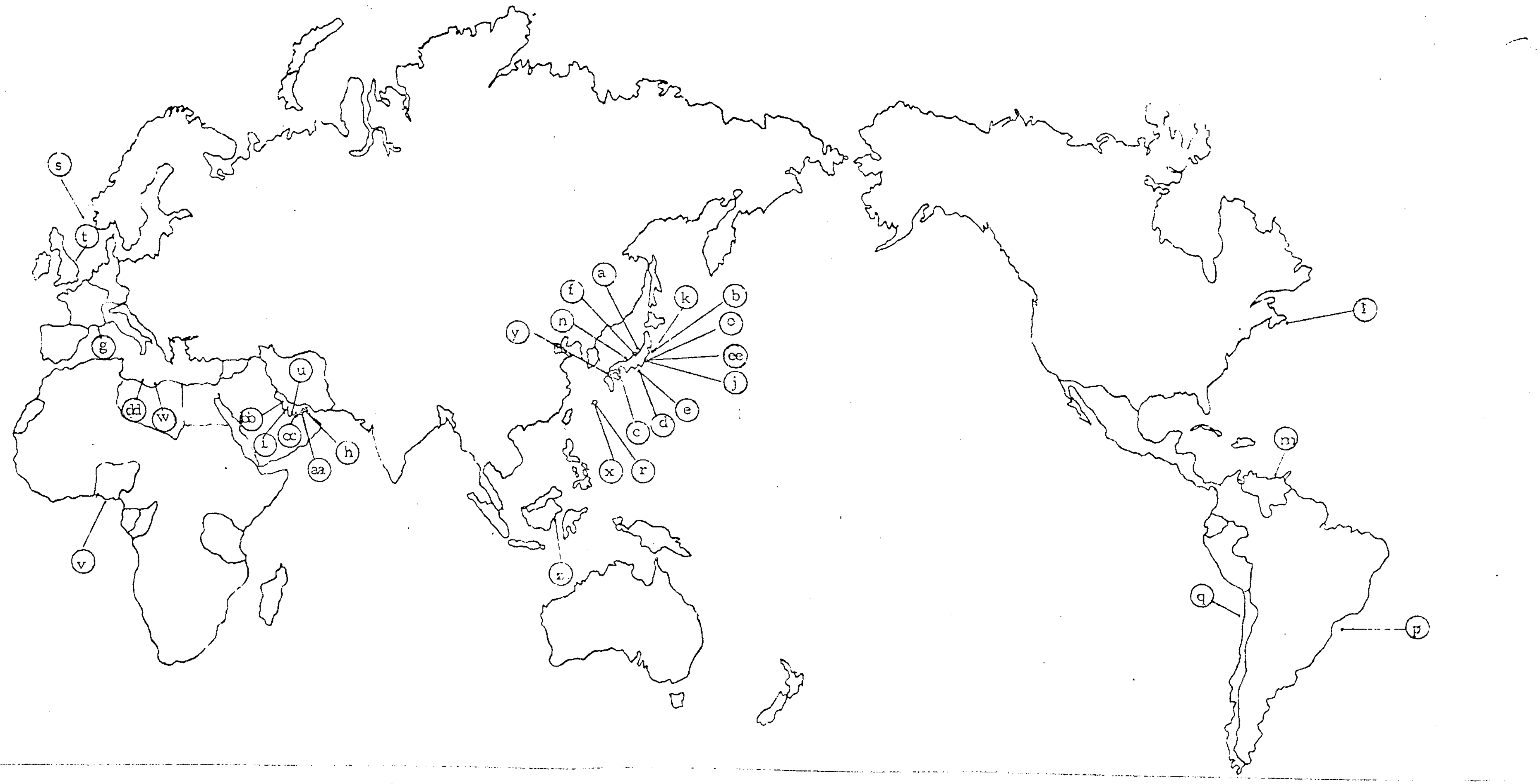


FIGURE (17) FAILURE DATA SUMMARY MAP

IV OUTLINE TO PROBLEM APPROACH

INTRODUCTION

Based upon the data and conclusions of the preliminary study of the problem it was apparent that a complete and comprehensive programme of work was necessary to achieve an improvement in large bore hose performance. Furthermore it was recognised that certain elements of such a programme might be:-

a) Outside the scope of work directed towards this thesis.

b) Complete within themselves.

c) The setting up of a procedure(s) as a short or long term study in order to aid the above objectives.

d) Severely complicated by Financial/Marketing restraints.

Considerable thought and effort were therefore expended in mapping out a framework to such a programme that it might be comprehensive, sufficiently structured to enable the package to be "sold" to the company and yet allowing flexibility of approach, timing and method so as to cope with changes of emphasis as the project progressed. The programme structure eventually put forward and adopted by the Company is shown by Figure (20).

It was intent that the programme be conducted such that results would be directed towards providing structural parameters that could be used to improve hose in service life. The ultimate mid-term objective was however to gain sufficient knowledge to design/build a "full scale dynamic test rig", capable of subjecting full scale hoses to loads which simulate actual service conditions. Through this device it was projected to duplicate failures that have been observed in the field, such that the mechanisms of failure could then be analysed and the effect of design changes determined under controlled laboratory conditions.

Before considering the programme in detail however it is worthwhile to note the peripheral activity of failure data collection. During the preliminary study it became abundantly apparent that the reporting, cataloguing and correlation of failure data left much to be desired. It was therefore considered, although generally outside the scope of the work undertaken by the writer, that the development and implementation of a hose failure reporting procedure was a primary immediate goal. Such a system was eventually developed by the Market Planning Department of the Company with reference to the writer on technical aspects. A questionnaire type format was devised with particular emphasis on flagging failures, site characteristics, and probable conditions under which those failures occurred. An example of this format is contained in Appendix (i), and is now completed by Company personnel visiting or communicating with a site from Director level downwards, the degree of completion dependent upon previously recorded data on the particular site.

Studying the programme in some detail it may be identified that the subject matter has been segregated into two major areas, namely "Macro" and "Micro", which call upon similar or complementary areas of activity in order to achieve the final aim. It is therefore worthwhile considering what is meant by the terms "Macro" and "Micro" and how the various activities integrate at various stages of the programme.

MACRO ACTIVITY

The term "Macro" has been employed in the outward looking or external sense of the word. Work conducted within this sphere has been directed at examining the hose as an element within a system with respect to the environment and to the system itself. In this context the hose has been regarded virtually as a black box (of assumed characteristics where necessary) and the effect of the environment and/or the overall systems response to the environment determined or monitored in the form of service loads; ie the determination of functional input data. Various broad techniques have been used during this part of the study which may generally be classified as shown below:-

MACRO → FIELD TESTING (FULL SCALE)
MACRO → MODEL TESTING
MACRO → MATHEMATICAL ANALYSIS

Each of the individual subject matter will be described in detail later in this thesis, however for clarity it is worthwhile to briefly consider the role and chronology of each aspect of the study.

FIELD TESTING

During the structuring of the work programme framework it became apparent that field testing would have to forefil an extremely important role. Firstly because of the diversity of service load estimates (previously noted by order of magnitude) an attempt to construct a mathematical or other model of the system would have been thwart with problems. The only acceptable test of a model is to compare its output with reality; if reality is not capable of definition then the model, at best, is of questionable use. Secondly because of the necessary time restrictions in relation to the construction of a full scale test facility, with respect of financial and marketing strategy considerations it was necessary that this aspect of the programme ran virtually concurrently with other study areas, or at worst only marginally behind. Major items to be considered, even at specification stage, required a basic knowledge of service loads, eg: are torsional loads significant with respect to other loading modes? What are likely maximum load magnitudes? etc. As stated above, mathematical prediction was similarly not considered to be adequate. Thirdly because of external pressures upon the Company it was important that they were <u>seen</u> to be doing something to overcome their problems. From a marketing overview, field testing demonstrates such far more readily than any amount of theoretical work.

Thus for a variety of reasons, some technical, others not, the setting up of a field testing project was given immediate priority. An initial site survey was therefore implemented in order to determine an appropriate terminal or terminals at which to conduct the study. Two principle criteria were set for the selection of the test site, first the site must have a record of moderate - severe sea conditions such that meaningful data could be obtained (preferably seasonally dependent conditions), secondly the area(s) must be accessible in order to maintain/monitor the equipment and such that repair or replacement of equipment may be made available within a reasonable period of time. Another extremely important factor was the requirement for the supporting co-operation of the operator of the terminal. Naturally work would be done at the convenience of the terminal work schedule, but naturally there would be times when certain delays might take place and the co-operation of all parties involved would be necessary.

Eventually a joint study was set up in conjunction with the Royal Dutch Shell Group and tests conducted at Shell BP's export terminal in Mid West Nigeria during the latter half of 1973. This study; the preparation, performance and evaluation is described in detail in Chapter (V).

MODEL TESTING

As will be described in Chapter (V) the field tests conducted related to a study, nominally of the floating hose system. It was also originally intended to conduct a similar study in relation to submarine hose applications. Upon reflection however it was considered that such a study might be subject to a number of complications not encountered during the afore mentioned study, namely:-

a) Failure pattern peaks in relation to submarine hose systems were found to be spread throughout the various components of the system(s) and not located at readily definable points as was the case with the floating system.

b) An attempt to instrument any area apart from those immediately adjacent to the buoy would be extremely complex and expensive.

c) It was apparent that the design of the submarine hose system contributed, to a great extent, to the environmental loads acting upon the hose.

Considering the latter point in some depth, it was determined that the detail design of such submarine systems, (typically Chinese Lantern or Lazy 'S'), was extremely suspect. Generally it was found that such systems were detailed on a drawing board, using crude drafting techniques, totally ignoring the structural characteristics of the hose elements. (eg: The writer during the course of his work viewed an internal memorandum of a large international oil company specifying a certain type of electrical flex which could be used to represent submarine hose in much the same way as flexi-curves are used by draughtsmen).

At best it was determined that such systems had occasionally been designed/evaluated using a crude model consisting of springs (representing hose elements) supported by counterbalance weights (buoyancy units) in air. It was not much wonder that divers had often spent weeks or even months installing such systems, laboriously modifying the buoyancy distribution on a trial and error basis, only to achieve a spate of early hose failures.

It was considered that considerable improvements could be made in this situation by some form of simple model testing, particularly if such tests could be conducted in a water basin rather than in air. The criteria whereby model testing had initially been rejected prior to the field study were again considered. However it was noted that two important aspects had changed:-

a) As a result of the field study, (although it be mainly related to surface hoses) an order of magnitude for service loads had been determined. It was assumed that such loads, although different in detail, could in general be considered applicable to submarine hose. This was based upon the basis that designs for the respective hoses varied very little and that the incidence of failure associated with submarine hose differed only marginally from that of their floating counterparts.

b) The "visual" configurations achieved during model testing could be compared directly with practice (ie: previously installed systems), and a comparative evaluation made between model and "real life".

A proposal was therefore put to the Company that some form of model testing facility be constructed. Initially such a unit would be used mainly to evaluate correct hose length, buoyancy distribution, manifold angles etc under extreme static envelope conditions, but later would be extended to enable dynamic input to be considered together with some form of load/stress determination within the model.

The proposal was accepted and a model test tank/peripheral equipment designed and installed during late 1974.

The initial test facility was purposely designed on rather a simple basis (ie: Dynamic input only allowed one degree of freedom for the simulated buoy motion) mainly due to the lack of knowledge in relation to meaningful input data. This facility is described in detail by chapter (VI).

Subsequent to the installation of the above facility and the evaluation of numerous submarine systems, (directly for customers and for Dunlop R & D) a further, more sophisticated unit has been installed (Late 1976). The latter unit enables greater scale models to be employed and provides a better simulation of buoy motion, (three degrees of freedom). This also is described in Chapter (VI).

MATHEMATICAL ANALYSIS

During the course of the work conducted as described previously it became apparent that either gaps could be filled or knowledge extrapolated by the use of mathematical models in conjunction with the measured data previously obtained.

A) FLOATING HOSE BEHAVIOUR PREDICTION MODEL -

The purpose of this study was to project the dynamic behaviour of a floating hose string at an SPM subjected to wave action. A floating hose string was modelled by splitting the total length into a number of elements and specifying each element in terms of weight, displacement and stiffness. The system thus defined was subjected to a regular wave train of specified height and period, and the maximum bending loads at each of the elemental modes calculated for different hose stiffnesses. (Chapter VII a)

B) BUOY CONNECTION HOSE CONFIGURATION -

As a direct consequence of the previously noted studied,results and observations made in relation to the Forcardos field study, and general failure statistics, it was noted that the loads acting upon this "first hose off the buoy" are governed to a great degree by the hose configuration. That is the weight, buoyancy and stiffness distribution of the hose in relation to the offtake manifold geometry.
The system was therefore mathematically evaluated, (in a static state), by building a model which enabled the equilibrium state to be determined for various combinations of the above quoted input parameters.
Such a study was considered necessary as there is a great variety of offtake manifold geometry in existence worldwide and an equal variety of hose designs (mainly differences in buoyancy distribution)

pertaining to be the ideal for that particular manifold geometry. In fact a analogous situation exists to that of the proverbial "Chicken and Egg". (Chapter VII b).

C) BUOY EXCURSION PREDICTION MODEL -

During the initial period of model testing it became apparent that very little information was available in relation to buoy excursion; (large lateral displacement of the buoy due to the mooring load of a tanker), let alone general buoy response characteristics.

The SBM terminal adopts a multi-chain system catenary anchor chains to effect anchoring of the buoy body. As such the anchorage system is based on a relationship between unit chain weight and buoy excursion. More specifically when a mooring force is applied, the buoy will move sideways, thus lifting sufficient chain from the seabed to restore the equilibrium.

The requirement of this study was to determine the relationship between buoy excursion and mooring force for various depths of water and anchoring system design. (NB: The relationship previously quoted by the buoy manufacturer and used by Dunlop gave an envelope of maxima excursion at various water depths as a squat cylinder - clearly incorrect!) (Chapter VII c).

D) SUBMARINE HOSE CONFIGURATION MODEL -

In conjunction with model testing it was considered desirable to build a computer model of a submarine system, such that a certain amount of design work could be evaluated before introduction of the physical model into the tank. Clearly, if possible, this would offer considerable benefits in reducing the number of test variables, thus reducing time and costs, at the model testing stage.

The computer model developed represents the initial work in this area in relation to a Chinese Lantern type system (Chapter VII d).

Nominally the aforegoing concludes the work grouped by the writer into the 'Macro' area.

MICRO ACTIVITY

The term "Micro" has been employed in the inward looking sense. That is work associated with treating the hose as an entity, the evaluation of its intrinsic properties and response to applied loadings.

In a similar manner that Macro activity revolved around the original Dunlop postulation that there was "Insufficient input design data upon which to base design modification or future designs" : Micro activity was based upon the parallel statement: "Given that the above (Macro) information could be made available, few design calculations or analysis techniques are historically available within the industry".

In a similar manner to the Macro study the area was subdivided, this time into two categories:-

MICRO
- CHARACTERISTIC PROPERTY EVALUATION
- THEORETICAL PREDICTION OF PROPERTIES

Again each subject will be described in detail elsewhere within this thesis, however for clarity each role is briefly considered at this stage.

CHARACTERISTIC PROPERTY EVALUATION

As stated previously in Chapter III, although hoses had been manufactured for many years little was known of their characteristics, that is apart from their burst capability.

It was therefore put to the Company that they should adopt a policy of measuring and recording as many properties/combination of properties as possible for their past, current and future generation of hose designs. In doing so it was hypothesised that characteristic properties could eventually be correlated with constructional design criteria in order to develop prediction formulii or models for specified characteristics. Additionally a basic knowledge of properties could considerably aid the Company's efforts in system design, Model testing etc. The Marketing potential evolved in being able to publish characteristics, (whilst competitors showed no signs of being able to do so), was also not overlooked or indeed understated.

The Company accepted that such work was both desirable and indeed necessary, thus a programme of such tests, as described below, was implemented.

In selecting the basic tests that were to be conducted the criteria considered were:-

1) Because of expenditure elsewhere (ie Model testing, Dynamic test rig etc), Capital expenditure should be kept to a minimum.

2) Tests conducted should cause minimal disruption to other factory activities, such as final inspection, transportation etc.

3) Tests should be aimed at providing the most immediate "customer orientated" results, rather than any protracted R & D study, although the latter as an integrated part of the former was acceptable.

Based upon these criteria the following three basic property tests were conducted:- (NB: Burst tests were not generally considered as they were already conducted periodically to fulfil specification requirements, although not documented at all well. Secondly, such tests required specially constructed specimens, necessary in order to effect pressure sealing of the fittings, and were destructive).

A) BEND TESTING:-

A system for bend testing of hose was standardised which enabled:-

1) Direct comparisons to be made between test specimens (ie Hose A v Hose B).

2) Calculation of bending rigidity, EI values.

3) Evaluation of EI against radius of curvature and internal pressurisation.

The test method devised has now been adopted by many oil companies/ 'turn-key' construction consortiums and it is projected that it may be written into the BMF specification as a standard, ex manufacture, final inspection test. (Chapter VIII a).

B) TENSILE TESTS:-

Various tensile tests were conducted, some to destruction, in order to evaluate figures relating to "tensile modulus" as well as ultimate tensile strength. Correlations were also made as to the effect of internal pressurisation upon these properties. (Chapter VIII b).

C) VOLUMETRIC DISTENSION TESTS:-

This test was devised in an attempt to replace the much distrusted, (with just cause), "temporary elongation under pressure" test; as a means of evaluating hose deterioration. Nominally the test consists of measuring the volumetric change of a hose during a pressurisation process. (Chapter VIII c).

THEORETICAL ANALYSIS

During the course of the characteristic property testing programme, certain phenomena were observed that required explanation; also additional techniques were found necessary in order to predict or correlate certain parameters. The following two are examples of such work:-

A) LARGE DEFLECTION OF BEAMS -

It became increasingly apparent that small deflection theory could not adequately predict the bending configuration of such a relatively flexible member as a hose. A technique was therefore developed that enabled such large deflections to be accommodated. In doing so a state of geometrical instability was noted which later was particularly useful in relation to the dynamic test rig (Chapter IX a).

B) BENDING STIFFNESS PREDICTION

The bending stiffness properties of hose structures were found to be relatively complex, the characteristic varying with radius of curvature and internal pressure. This study was an attempt to explain the latter of the two relationships. (Chapter IX b)

Referring back to Figure (20) - the programme structure - it may be noted that as an offshoot of the Micro study area was a project termed "Analysis of hose construction". This was included in the total study programme as the writer considered that in order to evaluate parameters, in general, relating to the external properties of the hose, it would be beneficial to also examine in some detail the internal hose construction. This could have been done, and indeed was done, by studying the various manufacturing processes and in addition examining failed hose specimens. It was however considered desirable to relate and document this knowledge in rather a more formal manner. It also helped to clarify various conflicting views on why certain components were included/omitted.

In order to forefil the above need a failure mode and effect analysis (FMEA) was conducted. (Chapter X).

FULL SCALE DYNAMIC TEST RIG

As stated previously the primary aim of the programme of work undertaken by the writer was to develop a full scale testing facility for the company. As reflected in the programme structure, the work conducted was aimed at supplying the needs in the development of such a facility although much was useful in its own right and/or enjoyed many spin-off usages.

A considerable amount of time was therefore spent working upon the many facets required for the introduction of such a piece of hardware. This is reflected in the extent of Chapter XI of this thesis.

In Chapter XI the various design parameters will be reviewed, the various initial rig concepts discussed leading to the creation of the final functional specification.

The design of the various elements are then considered together with the associated costings, time table and capital raising exercise.

Finally the installation/commissioning of the rig is reviewed together with limited feedback from the provisional tests conducted.

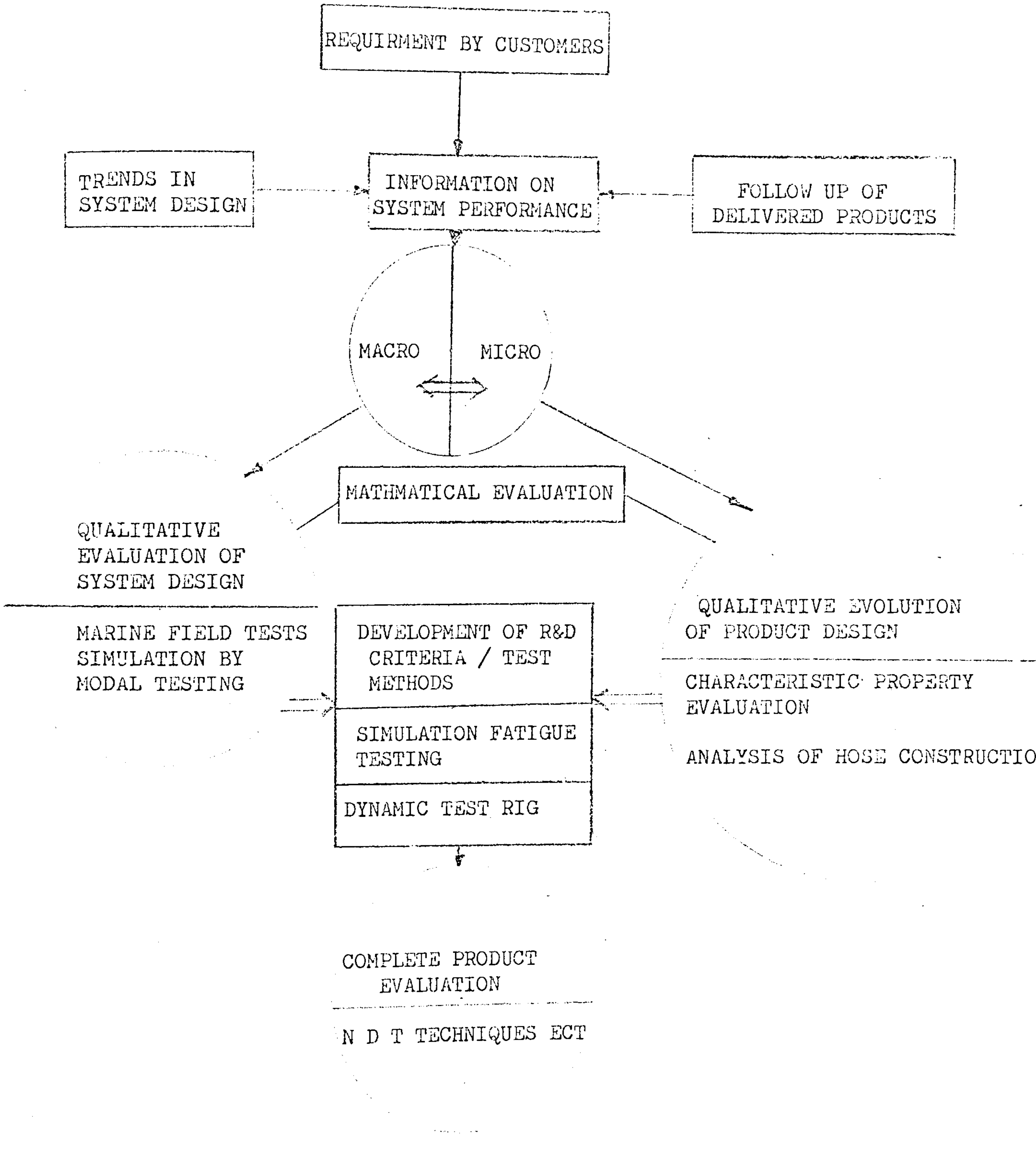


FIGURE (20) PROGRAMME STRUCTURE

V. FIELD TESTING

V.1 INTRODUCTION

As stated previously (IV.3), three major criteria were considered in selecting the test site for such a study of service loading:-

(a) History of moderate - severe weather conditions

(b) Accessibility

(c) Co-operation of terminal personnel.

The site finally selected, Forcados Terminal (Mid-Western Nigeria) - operated by Shell-BP, ideally forefilled such criteria.

(a) Forcados was regarded as a "bad" site, experiencing considerable periods of severe sea state - (Figure 21). In addition such weather conditions were of a seasonal nature, therefore ideal for determining the optimum period for conducting the test programme, (ie: "Wet Season").

(b) As a major export terminal Forcados employed two buoys, thus one could be released periodically for the installation of equipment, retrieval of data etc, without seriously disrupting terminal operations. The buoys, together with an offshore operations platform, were also only situated some twenty miles offshore. Daily servicing by shore based vessels or helicopter/platform/berthing vessels, made direct access to the equipment readily available.

Whilst it was realised that Nigeria was not the ideal location in terms of international communication, transportation, location of equipment spares etc. it was considered that the above stated benefits coupled with the resources of Shell-BP outweighed such draw backs.

(c) As a large inter-nation organisation, Shell (Royal Dutch Shell Group) operate the largest number of SPM's worldwide. As such, Shell were naturally anxious to give assistance to any party concerning the improvement of reliability of such installations. In this regard, and having reviewed the outline test programme, Shell offered financial assistance by:-

(1) Sharing the capital equipment costs on a 50/50 basis

(2) Providing all necessary services in Nigeria; accommodation, transportation etc. free of charge.

A co-operative study arrangement was therefore agreed between Shell and Dunlop.

In addition Shell-BP (Nigeria) Limited, as an operator of four SPM's in Nigeria, each generally of bad service history, were equally anxious to provide maximum co-operation.

Having selected the optimum site available, considerable time was naturally spent in establishing the test criteria/programme, selecting equipment etc. much of which is described by the following sections. Figure (22), however shows a schematic of the test programme development against the time scale involved. It is hoped that this will serve to give a general overview of the field study.

V.2 TEST APPROACH

It was clear at the outset of the project that there were two main angles from which to approach the problem. These being:-

(a) Direct measurement of the loads applied to the hose via the buoy connection. (Possibly involving the use of strain gauges, load cells etc).

(b) Measurement of the relative motions of the hose with respect to the buoy system and environment from which applied loading might later be synthesised possibly involving the use of accelerometers, gyroscopes etc.

No work, to my knowledge, had been undertaken in this particular area previously. Thus no guidelines were available as to which approach might lead to the more meaningful or "readily-adaptable" results. The basic worry when weighing these alternatives was the possible interaction of hose construction etc. (ie: would modifications carried out in view of the test results then invalidate the further use of those results). In elementary terms the problem was one of determining the "more" independent of the two variables, (ie: determining the dependent and independent variable). Stating the problem thus, perhaps over simplifies it, but it does bring out the essence of the problem.

Consideration of the problem in greater depth proved to be inconclusive, possible load versus deflection relationships being dependent upon a number of parameters such as "slenderness ratio" (ie: bore/length), stiffness, buoyancy distribution etc.

Thus with very little to choose between the approaches from the final analytical viewpoint, economic and practical factors were considered. From these viewpoints option (a) (Load measurement) clearly emerged favourite. This is clearly indicated by the following points:-

(1) Capital cost of option (a) equipment considerably lower than option (b).

(2) System and equipment required for load measurement less complex than option (b), thus less susceptible to problems commonly associated with "onsite" work.

Based on the major reasons stated plus further information not described, option (a) was selected. However, as a move towards option (b) it was decided to Photographically record basic motions of the system.

A complete knowledge of the forces acting was considered to be of greater value with a knowledge of prevailing environmental circumstances, particularly those at peak load occurrences. It was noted that weather parameters were visually estimated at six hourly intervals by Shell BP personnel, however, although this was considered adequate for current and wind parameters, it was not for sea conditions.

Clearly, relating individual waves to force peaks etc., (requiring buoy mounted equipment) would be extremely difficult and expensive. It was considered that a record of the general wave profile during each test period would suffice. For this purpose it was intended to utilise a wave recorder already mounted beneath the Forcados offshore platform (approximately one mile from the test buoy). The camera, primarily intended for monitoring hose motion, would also provide a valuable secondary source of visual wave data.

It was concluded that the system thus selected optimised the cost: Information ratio.

V.3 TEST EQUIPMENT

In order to measure and monitor the forces and motions described in the previous section a Data Logging System was designed and developed.

Designing for extremes of environment conditions, required extremely careful selection of instrumentation. The monitoring equipment chosen was selected with these conditions in mind, together with other important criteria such as robustness, long term stability and certain power and size limitations. Great importance was attached to modular design in order to facilitate fault diagnosis and replacement. Back-up sub-assemblies were designed into the system at critical points, thus enabling full operation to be maintained even after a number of component failures.

The measuring system comprised three main units - load measuring spool, instrument capsule and photographic system.

(Figure 23)

The load measuring spool converted mechanical loads into electro-analog signals using strain gauge techniques. The instrument capsule provided the facility for simultaneously recording the outputs of four strain gauge bridges on magnetic tape. It also provided the power requirements for itself and other two units, the total system being activated by a short wave radio link.

(Figure 24)

The Photographic system provided the facility for visual recording of the hoses by time lapse photography. A link between these later units was provided by synchronisation between film and tape.

LOAD MEASURING SPOOL

Initially it was thought that it might be possible to implant strain gauges within the hose during its construction. This approach appeared to offer the following advantages.

(a) Overall loads, of the stated modes, could be measured at any point.

(b) Effects resulting from the super-position of these loads could be monitored, ie: total loading.

(c) By implanting gauges both radially and longitudinally within the carcase, stress distributions could be determined.

However, after careful consideration, this was thought to be too adventurous and complex for the initial research, particularly with respect to the probable on-site conditions. (Determination of stress distributions etc. could be performed at a later stage, recreating loadings via a dynamic test rig).

Having thus decided to concentrate on the transmitted loads, (ie: load acting between the buoy and hose flanges), it appeared logical to gauge a metal, rather than rubber structure. (Gauge techniques being that much simpler). In this respect the existence of a short installation spool adjacent to the manifold simplified the system considerably, the measuring spool being a replica of the installation spool, suitably instrumented with strain gauges. The geometry of the system was, (very importantly) therefore, not altered.

At the outset it was considered that even order of magnitude results would be a significant improvement on the present situation. None-the-less an accuracy of ± 10% of measured value was considered of more use as a design tool. The lowest forces of interest (Axial load) were considered to be approximately 8 tons, it was therefore desirable to measure to ± 1 ton.

Alternatively the spool piece had to be able to withstand 240 tons, and to do so without appreciable distortion (<3%). As a result the spool stiffness required was quite high, thus requiring a correspondingly sensitive measuring system. (Similar considerations also applied to the measurements of bending moments and torsion couples).

The problem was further complicated by the need to keep the spool piece short, (identical dimensions to that of the installation spool). So short in fact that it was no longer possible to theoretically predict its calibration accurately because of the disturbing effect of the end flanges.

Conventional strain gauge techniques and arrangements were employed to measure the discrete loads without recourse to strain circle analysis.

(Figure 25)

(See Appendix 12 for Gauge Diagrams etc).

Because of the possibility of high temperature differentials across the spool under certain conditions, self temperature compensating gauges were used in addition to compensating gauge arrangements. (The maximum anticipated temperature differential of 40°C, (normally only 2° - 3°C), could produce apparent strains of up to half the operating range without these precautions). The entire gauge system was duplicated, thus providing secondary systems for all load channels, for alternate use in cases of failure.

Since any system employing strain gauges is particularly sensitive to high humidity multiple precautions were taken to avoid moisture ingress. (The standard requirement for the earth isolation of the gauge circuits was 500MΩ. Individual gauges, terminations, etc., were encapsulated in a silicone rubber compound. A Monel shield, (designed to add minimal structural stiffness), was welded above the gauge system, creating an integral waterproof structure in addition to providing physical protection.

Cables connecting the gauge system to the instrument capsule were contained within a reinforced flexible hose. The function of this being both protective and waterproofing.

The joints at either end of the communications hose were protected by a series of waterproof gaskets and "shrink fit" plastic sleeves. Cables projecting from the "buoy end" of the hose were also protected by shrink fit sleeving and terminated in waterproof Hydro-lock connectors.

Because of the particularly long cable lengths required, (connecting the gauges to the bridges), it was essential that any shunt resistance and capacitance were not sufficient to influence the measurement. Since the minimum desired signal of 5 micro-strain corresponded to a shunt resistance of approximately 3.4M ohms or its equivalent shunt capacitance of 5 pico farads, it was evident that the closest attention was required for the insulation of the circuits. This included tracking resistance on the plugs etc., and the possibility of variations in shunt capacitance due to flexure of cables etc.

(The effect of cable flexure was to be determined during the Calibration procedure).

PRESSURE MEASUREMENT

It was decided, along with measuring the hose loading, to measure the line pressure. This was prompted by the occasional occurrence of pressure surges during valve closures etc.

A pressure transducer was installed to measure both normal and surge pressures at the buoy.

INSTRUMENT CAPSULE

Once connected to the instrument capsule, any four of the nine channel inputs, (2 x axial load, 2 x V. Bending, 2 x H Bending, 2 x Torsion + 1 x pressure), could be connected (via bridge balance and conditioning units) to a frequency modulated tape recording system. These instruments, together with radio receiver, power supply and auxiliary circuits, were contained within a water tight G.R.P. housing purposely designed and constructed.

The instrument and camera housings mounted on the upper structure of the buoy were considered to be in a Zone 2 safety area. (ie: Explosive risk due to presence of oil vapour/electrical equipment).

On the basis of Zone 2, applicable safety measures were:-

(i) Restricted breathing

(ii) Flame Proofing

(iii) Intrinsic Safety

(iv) Hermetic Sealing

(v) Purging (+ monitor)

Of these it was considered that the equipment could best be made to conform to (i), ie: restricted breathing standards (BS 4683 part 3, 1972).

The basic requirement of weather and waterproofing for such an exposed location virtually demanded hermetic sealing. Multiple external clamps on silicone greased 'O' ring joints were used to seal the enclosures. The Hydro-lock through connectors used were designed for submarine duty.

As an additional precaution the instrument capsule was mounted on Anti-shock mountings within a heavy duty mesh security cage.

Within the capsule individual conditioning units were used to excite each of the four strain gauge bridges under observation. These modules, Vibrometer 81CF units, provided both bridge balance and amplification facilities. Operating from a stabilised D.C. supply these units operated on the carrier frequency principle at 8 KHZ. (Clearly an A/M system would have suffered from severe loss of sensitivity and noise "pick-up" because of the long cables etc). The output could be varied by a combination of step and variable gain controls thus providing an input sensitivity of 20 - 400 mv f.s.d. and a variable D.C. output of 0 to $\pm$ 10V.

The principal features of this unit were its extreme robustness and stability, (temperature zero drift $<$ o.02%/^{o}C: Long term zero drift $<$ 0.2%/Month).

(Appendix 7)

The selection of a suitable tape recorder was extremely difficult. Although many recorders could fulfil the signal reproduction requirements they could not meet the necessary continuous run or stringent size and power requirements. (It was because of these requirements, and Computer Analysis considerations, that types of U.V. or pen recorders were dismissed).

This problem was finally solved by the Medilog 4 - 24, miniature 4 channel F.M. Cassette recorder, manufactured by Oxford Instruments Limited. Both its size and power requirements had been reduced by the use of integrated circuits and miniaturized tape drive system. The drive system, running at a tape speed of 2mm/sec, gave a run time of over twelve hours with a standard C-60 Cassette.

Because of the extremely slow tape speed the signal band width was restricted to 10HZ. The shortest wave period anticipated was approximately 5 seconds (corresponding to 0.2HZ). Since the hose excitation thus produced could not be expected to be sinusoidal, it was necessary to measure harmonics also up to a frequency of several HZ. It was considered that the recorder giving a roll-off of 3 db at 10HZ would be adequate.

In order to achieve such a low tape speed and yet retain good signal reproduction characteristics a pulsed modulation carrier system was employed, (ie: the input signal modifies the mark space ratio of the carrier). As in any F.M. system, tape speed stability was of paramount importance. The tape servo system gave the following characteristics: Long term stability $<0.5\%$; short term flutter $<2\%$ in the measurement band.

(Appendix 8)

The power supply for all instrumentation was provided by Non-rechargeable dry batteries. There were four groups of batteries:-

(i) Internal to the tape recorder

(ii) 36 v supply for conditioning units. (stabilised to 22V)

(iii) 12 v supply for the camera and its auxiliaries (stabilised to 7.2V)

(iv) 24 v supply for the radio receiver.

Supplies (i) to (iii) were sized, after allowance for deterioration in tropical storage, so as to operate the equipment for a period of not less than nine hours.

The radio supply (iv), the only one in continuous use, was sized for a minimum of seven days continuous operation.

The complete system was energised, as stated, by a short wave radio link. The receiver, mounted within the instrument capsule, could be activated on receipt of a signal from the transmitter installed on the offshore platform or service vessel. This radio system operated on a frequency of 73.075MHZ, (later changed to 87.15MHZ), purposely chosen outside normal operating channels. As an additional safeguard against accidental operation, the signal was frequency coded. Design of the relay link ensured full operation of all systems five seconds after "latch-up". For the purpose of calibration, testing etc., it was possible to override the radio link manually from the instrument capsule.

(Figure 26)

The photographic equipment was also contained within a heavy gauge water tight housing to the same specifications as described previously. For clear vision a double window assembly and windscreen wiper were employed.

(Figure 27)

The camera used was Beaulieu S.P. 16mm Cine equipped with a 5.9mm wide angle Angenieux lens with servo controlled diaphragm. Time lapse filming was employed, the filming rate being variable from six to sixty frames per minute. Using a 200 feet magazine, (8,000 frames), this gave a maximum run period of seventeen hours. In order to synchronise the film and tape systems, the following facilities were provided;

(a) light emitting diode illuminated "in-frame" every 100th frame and

(b) simultaneous transmission of a 10 HZ (square wave) pulse train onto channel 1 of the tape recorder, overwriting the data.

"Slate" operation was used for initial synchronisation and labelling of the film.(A board was positioned in front of the camera lens during this sequence giving details of recording date, film number etc., the marker unit operating the synchronisation facility with each frame.)

In order to trim the camera for various lighting conditions servo control of the diaphragm was employed via an electro mechanical drive system controlled by a photo-cell. During normal operation the accuracy of this control system was within 1/3 of an f stop. Current limiting circuits were included to prevent servo overload in very low lighting conditions.

The G.R.P. camera housing was located on a turntable mounting system, above the "right hand" buoy offtake, (Figure 23). The mounting position and equipment were so designed to enable changes to be made to the field of view, both in rotation and elevation.

CALIBRATION

In any measurement chain precise calibration of the total system is vital to obtaining accurate and repeatable results. It is possible to calculate such characteristics from theoretical stress analysis and previously defined component parameters. However, when commissioning a new system, especially one for use under unfavourable conditions, it is essential to perform an initial physical calibration. The calibration of the total system was particularly important in this instance because of possible signal degradation and noise generation.

In addition, a physical calibration of the load measuring spool afforded an opportunity to determine possible interaction between the various channels, also to check the temperature stability, waterproofing etc. The system proved satisfactory in all respects:-

(1) The discrete nature of the signals (pure axial loading, pure bending and pure torsion) was confirmed.

(2) Heat was applied to both spool and instrument capsule in order to check their temperature stability; apparent strain generation or drift was barely perceptible.

(3) The previously mentioned cable dynamic capacitance asymmetry was determined by flexing the cables; no effect was noted.

(4) Prolonged spraying with water jets produced no moisture ingress.

To perform the load calibration, the load measuring spool was fixed firmly (encastre) to a load bearing stanchion. Bending moments or torque were applied to the free face of the spool via a rigid torque arm to which force was applied by a winch or jack system. Accurate measurements of the forces applied to the beam were registered with a load cell, thus allowing the applied moments (torques) to be calculated.

(Figure 28)

Axial load was applied by internal pressure with the flanged ends

blanked off, the load being calculated from a knowledge of the internal load area and the applied pressure. The residual effects of hoop stress were calculated and the calibration curve suitably modified. The pressure transducer was calibrated using a dead weight calibration rig. Calibration curves for all nine channels (four main and four back-up on the spool, together with the pressure transducer) were prepared, one of which is shown.
(Figure 29)

Having established the initial calibration it was necessary to define a procedure for the re-establishment of this state on the monobuoy. Two major problems were immediately obvious: the amplifiers offered variable gain control and, once installed on site the system could not be "unloaded" for zero drift correction.
The first of these problems was tackled using the apparent strain resistor method: (ie: applying a resistor across the terminals of one arm of the bridge to create a positive or negative output equivalent to that of a static load.)
A series of such resistors was calculated for the desired range of loads. These were then applied across the bridge arms, the output being compared with the appropriate calibration curve for the resistor "load equivalent".
(Figure 29)

The second of the two problems was rather more difficult to solve in that it was not possible to create an identical control bridge network to connect in place of the spool circuits for zero balance purposes.

Before installation on the buoy, each channel was balanced by its respective conditioning unit. The gain was then set and the calibration established as described above. (At this stage the system was identical to the initial calibration state). A standard control bridge, of high tolerance resistors, was then substituted for each circuit in turn. Static voltage offsets (due to discrepancies between the true and dummy circuits) were noted.

On the buoy all channels could be returned to the true zero state by using the control bridge and setting the static offsets to those values previously established. In this manner, both zero balance and gain could always be established.

Procedure routines and standard on-site record sheets were prepared and followed to ensure the consistent validity of on-site results.

V.4 TEST HISTORY

The original consignment of test equipment arrived in Nigeria at the beginning of June such that the tests could be commenced at the start of the "bad weather" season.

Because of some confusion concerning the import agents, Custom authorities and transportation, some considerable delay was encountered before the equipment reached Forcados, (late June).

On arrival, the majority of the equipment was found to be in good condition. However, the packaging of the spool and communications hose was found to be damaged. An electrical test of the spool circuits later indicated defects in two of the four modes.

During further tests on the equipment the radio receiver (73.075 MHz - advised by Shell-BP) was found to activate via one of the Forcados voice channels. (Although the activating signal required was frequency coded, this code was found to be generated during certain speech patterns). Two further operating frequencies were then advised by Shell-BP. (Subsequently the radio system was modified to 87.15 MHz and found to function correctly).

An examination of the fixtures mounted on the buoy, (modifications requested by Dunlop) indicated a number of dimensional discrepancies against the drawings submitted to the Hague. Shell-BP were requested to reposition bolts etc., accordingly.

A "Back-up" spool was dispatched to Nigeria at the beginning of July. After some delay, (Customs clearance, transportation etc) the spool arrived at Forcados. On this occasion no visible damage was apparent.

Because of constant use of the Buoy, spool installation did not commence until the beginning of August. However due to severe sea conditions it was not possible to complete the installation. (Up to this time the spool and floating hose string had been bolted into position, the communications hose however had not been properly secured). At this stage the supply vessel sheared one of its mooring lines. Because of the apparent danger, the turntable was released and the 'hose gang' taken from the buoy after temporarily securing the free end of the communications hose.

On return to the buoy the following day, the communications hose was found to be severed immediately adjacent to the spool thereby enabling water to enter the sealed system. Inspection indicated that the free end of the hose had been secured by the hose gang to a non-rotating section of the buoy. Thus releasing the turntable had made the consequences inevitable.

A further "back-up" spool was dispatched to Nigeria which arrived by Mid-August. On this occasion, Customs clearance etc., proved to be less of a problem. The spool was subsequently safely installed and the test programme implemented at the beginning of September.

After two weeks operation, the tests were brought to a temporary halt due to failure of the signal cables within the communications hose. (During a submarine hose change, (with the turntable locked); motion of the 24" horizontal swivel caused overstressing of the communications hose, resulting in the cables severing).

The spool was removed from the buoy and returned to Forcados for repair. Damage was found to be not severe. Repair was implemented and completed with only the loss of two "back-up" bending channels.

The spool was available for re-installation one week after removal. However, the existence of excessive terminal stock levels caused a further three week delay before the buoy became available for installation purposes.

The test programme thus recommenced during the first week of October. Tests then continued at regular intervals throughout October and early November. Initial on-site analysis, (tapes played back via a U.V. recorder) indicated that interesting and valuable results were being obtained.

The programme was brought to a close by joint agreement between Shell-BP and Dunlop, (Hague and Lagos meetings) in mid-November.

V.5 RESULTS

The analysis of the recorded data was made in the following manner, (schematically shown by Figure (30)):-

(a) Upon recovery of the magnetic tape from the buoy, it was initially replayed via an ultra-violet recorder, so producing a visual record of the loadings. (Refers Figure (31 a-c)).

(b) Sea state data (manually recorded*) in terms of wave height, period and direction was analysed and schematically represented for each recording period (Figure 32).

(c) Film relating to each test period was processed and similarly recorded in relation to (a) and (b), Figure (33).

(d) By examing items (a), (b) and (c) in conjunction sample periods of each record were selected for detailed analysis; in order to perform the latter the records were sampled and digitised before processing by digital computer. (Detailed method described in Appendix (11).

The following parameters were used to describe the analytical zone, ($\simeq$15 minutes real time):-

Mean Value

RMS

Standard Deviation

Max

Min

Range

Probability Density

Autocorrelation

Power Spectral Density

Figure (34 a-f) represents typical results of such an analysis. Generally the results obtained may be summarised as follows:-

(a) AXIAL LOAD:

Mean:	0 ⟶ + 10 Tons (Tensile)
Dynamic Range:	± 11 tons (Max. range)
Distribution:	Approximately Gaussian
Frequency Range:	2 - 30 seconds (Period)
Dominant Frequency:	2 - 6 seconds (corresponding to short period waves or "sea").

(b) TORSION:

Mean:	Zero (± 10 ft. ton - rare occurrences)
Dynamic Range:	± 36 ft. ton (Max. range)
Distribution:	Approximately Gaussian
Frequency Range:	2 - 30 seconds
Dominant Frequency:	2 - 6 seconds

(c) VERTICAL BENDING:

Mean:	− 5 ⟶ − 20 ft. ton
Dynamic Range:	± 16 ft. ton (Max. range)
Distribution:	Approximately Gaussian
Frequency Range:	2 - 30 seconds
Dominant Frequency:	2 - 6 seconds

(d) HORIZONTAL BENDING:

Negligible due to the efficient action of the horizontal swivel installed on the buoy.

(e) LINE PRESSURE:

No surges detected. Small variations noted however, suspected cause vertical motion of the buoy.

Range of Variations: $\pm$ 20 lb/in^2 (Max. range)

Frequency of Variations: 4 - 8 seconds

* Originally the Forcados project had been planned such that extensive load/sea state correlations could be analysed. To forefil this requirement Shell-BP were to make available data from their platform mounted wave recorder (pressure type), located approximately one mile from the test site. The measurement method, one five minute sample/hour, being appropriate for such analysis.
Unfortunately upon recovery of the data tapes and subsequent attempts at analysis, (the process was subcontracted to a UK based survey company), the results were found to be un-decipherable.
Such lack of quantitive wave data was a serious set back to any correlation analysis. However a secondary source of wave data, in terms of six hourly visual estimates, was still available for analysis.

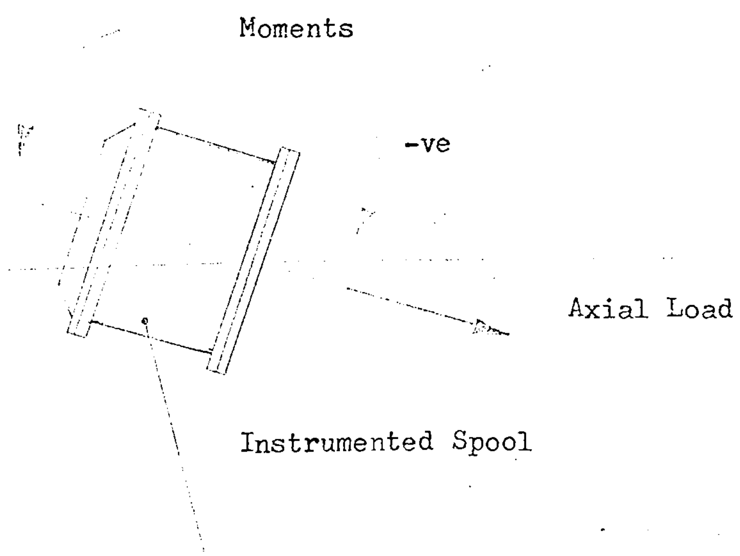
CONSIDERATION OF TEST RESULTS:

(a) MEAN LOADS (STATIC OFFSETS):

Generally it may be considered that to a greater extent static offset loads are a function of factors which are relatively stationary in their own right, and not upon dynamic variables such as wave motion.

As averaged values throughout the tests conducted the following are typical mean loads:-

Axial	:	+ 5 ton
Torsion	:	Zero
Vertical Bending	:	- 15 ton ft.



1. AXIAL LOAD

One obvious possible cause for such static tensile loads is the effect of surface current acting along a relatively long hose string.

Considering that the shear force acting at the hose surface may be approximated to:

$$\tau = 1/2\, \rho \bar{u}^2 f$$

for a 24" bore hose (ignoring the effects of fittings etc) the relative roughness of the hose surface approximates to 0.0043 resulting in a friction factor of 0.04.

Thus assuming a "wetted" surface area of 80% and a hose string length of 1000 ft.

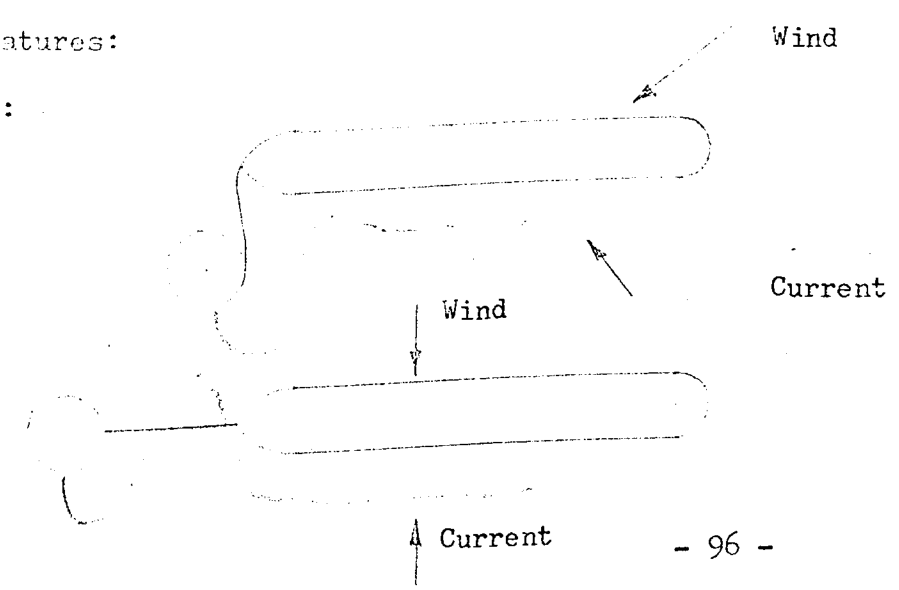
The drag force $F_D \simeq 390\, \bar{u}^2$ lbf where $\bar{u}$ = current stream vel. (ft/sec).

Therefore considering a current range of 1 - 3 knots (typical for the Forcados location); the resultant drag force becomes:-

$\bar{u}$	F
1 Kt	0.5 Ton
3 Kt	4.5 Ton

One further aspect however also requires consideration. During tanker loading, the configuration of the hose string may be determined by many features:

eg:

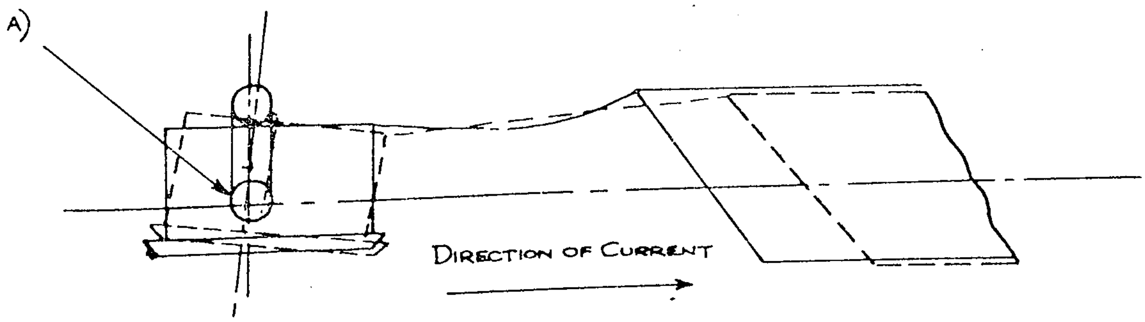


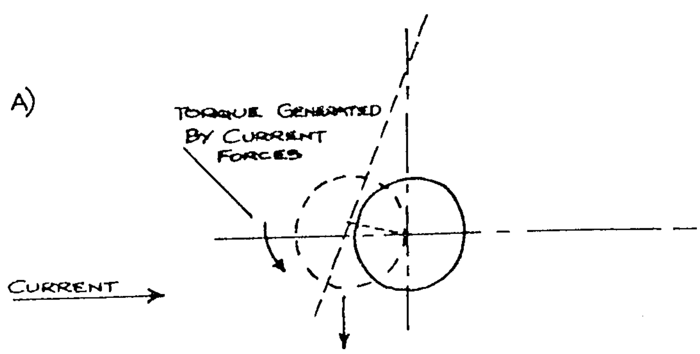
The hose string may thus tend to assume configurations as shown, thereby inducing tensions far greater than those previously calculated.

2. TORSION

Generally, no significant static torsion level was detected during the series. This was as expected, since torsional loads may only occur as a result of differential rotation of the buoy/hose system. Virtually by definition, such motion is oscillatory and thus is not applicable to static situations.

Occasionally, a minor degree of static torsion was detectable. Examination of the data describing the appropriate periods suggests a possible explanation:-





Resulting from a combination of environmental loads, (current, wind etc), the tanker establishes a large mooring force. This, as a result of the buoy geometry and anchoring system causes the buoy to tilt as indicated by the diagram. In doing so the pipe arm manifold effectively rotates and alters the buoy connection hose configuration. It is estimated that with buoy tilt of (5-10^{o}) torsional loads approaching 10 ton ft could be produced.

(NB: Buoy tilt may be noted from a number of the photographic records, Figure (33), exhibited as a sloping horizon, of approximately 7 deg.)

3. BENDING (VERTICAL)

Calculations were performed in order to predict probable "static deflection" bending loads, the method employed based upon small deflection theory, incorporating Macauley's Method in a numerical solution. (The latter was necessary as the resultant hose configuration determines to a great extent the bending loads which act upon the system, ie: nett buoyancy distribution).

Similar calculations were performed for four type connection configurations.

	FLANGE ANGLE	HOSE FLOATATION DISTRIBUTION
(a)	15 deg.	2/3 Distribution
(b)	30 deg.	2/3 Distribution
(c)	15 deg.	1/2 Distribution
(d)	30 deg.	1/2 Distribution

Case (a) being the situation at Forcados.

The following results were obtained:-

	BENDING MOMENT (TON.FT)	REACTION FORCE (TON)	(Both at the instrumented spool)
(a)	-11.5	0.7	
(b)	-26.5	0.7	
(c)	-16.5	0.5	
(d)	-30.0	0.5	

A reasonable degree of correlation would therefore appear to exist between the bending moment measured and that which might be theoretically predicted.

(NB: The relationship between buoy manifold angle/position and connection hose floatation geometry varies considerably around the World. Initially horizontal hose connections were made employing fully floated main line type hose. These were however prone to rapid failure and replaced by variations typical of which are the four cases considered. To the writer's knowledge no in depth theoretical analysis has been made of this situation, a state of personnel experience/ preference continuing to prevail. Chapter VII (1) briefly considers this state although much work remains to be conducted.

4. INTERNAL PRESSURE:

During the test series no serious pressure peaks or surges were detected. Additionally it was noted that the hoses were never operated at anything approaching their design pressure rating. (Maximum pressures recorded approximated to 150 lbf/in^2, whereas the design pressure rating was 225 lbf/in^2).

(NB: This situation prevails at many terminals throughout the world and one wonders if the significant pressure rating safety factors thus engendered could not be reduced in order to achieve more desirable properties in other areas, eg: Flexibility etc).

It was noted however that minor pressure fluctuations were always present, these being of an order of 0 - $\pm$ 20 lbf/in^2. These were eventually found to approximately correlate to buoy heave motion, ie: vertical motion generated by wave action. This is perhaps not surprising as the effective static head will vary with buoy heave, thus resulting in a pressure varient, in this magnification also being produced as a result of twin 20 inch bore submarine lines combining to form a single 24 inch floating line.

The presence of these relatively small pressure variations may at first appear to be insignificant, however it should be noted that several million such variations could occur in a year, (dependent upon predominent wave period range), and as such could contribute to the general fatigue situation.

(b) DYNAMIC LOADS

1. PROBABILITY DENSITY

It was observed, Figure (34), that the probability density functions plotted, in each case, approximated to a Gaussian distribution. (NB: It may be noted that the Axial Load distribution exhibits a 'spike' towards the negative asymptote. This feature may be explained by the camera synchronisation pulse overwritten on the axial channel).

By its nature a normal distribution implies the presence of both a restoring force and damping; most physical activities exhibit this feature; not to do so would imply some ideal state. Although not profound in its nature, the approximation of such physical phenomena to a normal distribution does lend confidence to its validity. Having established such a distribution it then becomes possible to assess the probability of any load occurring, from a knowledge of the standard deviation.

Expressing the standard deviation as unity:

Percentile :	99.99	99.9	99	95	70
Probability :	1/10,000	1/1000	1/100	1/20	3/10
Deviation :	3.9	3.3	2.6	2.0	1.0

ie: 70% of the loads fall within the range $\pm$ 1.0 std. deviation.

The above aspect becomes particularly important in a study of fatigue and fatigue life. For example, the probability of a particular load occurring may be used to derive the test programme as a basis for a cyclic fatigue test, (Dynamic Test Rig). Eventually this could also lead to the prediction of fatigue life for a set of known criteria.

2. AUTOCORRELATION:

The correlogram is logically the first step in the statistical analysis of a time series, since its form indicates at once if the series is worthy of further attention.

High early correlations and the general smoothness of the curves are evidence of coherence, a feature present in nearly all environmental time series. The coherence being imposed by the physical processes which produce the variations. Besides their smoothness, the shape of the various correlogrames generally are simular. This resemblance, together with their high initial correlation and general smoothness, indicates that all of the series are based upon a similar physical occurrence. Examination of the correlograms indicates a reasonably strong occurrence which persists for approximately five seconds. The latter approximately corresponds to the shorter period waves rather than the longer period (20-30 second) swell. Some evidence was found which indicated a very weak recurrence of approximately twenty seconds, this was however not at all clearly exhibited.

3. SPECTRAL DENSITY:

The importance of the power spectrum is as a means of showing the relative importance of the power, or variance, of oscillations in different wave bands. Generally if two series have strongly dissimilar spectra, then they are unlikely to be related in any predictable fashion. The converse is not always true, but the presence of peaks near to the same frequencies does imply that they might be related.

Analysis of the periodograms indicates a number of interesting features:-

(a) All of the periodograms conform in general shape/appearance. As such this would appear to suggest a relationship(s) between the data together with the exciting external motion.

(b) The maxima peaks exhibited by all of the periodograms fall within the frequency range 0.3 - 0.15 Hz (ie: 4-7 sec period). This would indicate the frequency range through which maximum power (excitation) occurs. Once again this approximates to the shorter period "locally generated" waves.

(c) A large proportion of the periodograms exhibit a plateau towards the low frequency range as well as a maxima peak. The average nature of this plateau persists over the frequency range 0.15 - 0.05 Hz. (7-20 second period). It thus would indicate the presence of the longer period waves or swell of that order of frequency.

(d) The very steep "fall off" at the low frequency end illustrates the restriction of motion to a restricted narrow wave band.

(e) In reviewing the relative magnitudes of the PSD quantity it may be noted that axial load and bending traces exhibit similar and higher values than the equivalent torsional trace. This would suggest a closer correlation between the former quantities than the latter. The buoy/hose configuration might of course lead one to this assumption without the PSD evidence.

(Naturally PSD values will be of usefulness in generating meaningful test programmes in any load simulation testing).

V.6 CORRELATION ANALYSIS

The case when one dependent variable can be estimated only in terms of a single independent variable is rather uncommon. For this reason an establishment of possible interactions is required, together with a weighing of their possible significance.

The correlation coefficient has some of the properties which we require in order to obtain a measure of the linear association between variables. However, before using the correlation coefficient for this purpose we must be aware of two serious pitfalls. The first is that the coefficients relate to the sample and not to the 'population', so they will show sampling variations about the true value. Secondly, is assessing the significance levels of the results, these are based upon the assumption that the variables involved are normally distributed. Although previous analysis has shown this approximately to be the case, it is difficult to be certain particularly with moderately small samples. Departures from normality can have serious effects depending almost entirely on the few extreme cases.

As stated previously the project had been planned with load/weather correlations in mind. However the failure of the wave recorder was a serious set-back to such analysis.

However an alternate source of "visual sea condition observations" was however available. (NB: These were observations set up by the project personnel at the offshore platform as a back-up to the recorder and to provide immediate data for provisional analysis. The estimate was collected by recording the average wave height, against a 'wave staff' appended to one of the platform legs. These were calculated for five minute periods on a six hourly interval basis).

In using these estimates however a number of qualifications should be noted:-

(1) Because of the 'estimate' nature of the data, serious discrepancies may exist. (This was heightened by the danger of obtaining such data, the recording area being in the wave/splash zone of the platform).

(2) The nature of the estimate process, one estimate/six hour period, makes direct correlation extremely difficult; especially with reference to the unpredictability of the Forcados weather situation.

(3) Information relating to wave period(s) was not provided.

In order to conduct the analysis the relevant wave data for each of the tests conducted were extracted and the equivalent load data averaged throughout the observation period. Thus series of values were generated for each of the twenty independent tests conducted, (see table below).

TWIN NO: IN CORRELATION SERIES	AXIAL LOAD 'STD. DEV' 1 (TON)	TORSIONAL LOAD 'STD. DEV' 2 (TON FT.)	BENDING LOAD 'STD. DEV' 3 (TON FT.)	WAVE HEIGHT (AVE. OBSERVED VALUE) 4 (FT.)	PLOT IDENTIFIER
1)	1.6000	5.6000	6.4000	3.0000	5
2)	4.4000	5.8000	9.2000	5.4000	6
3)	4.0000	3.4000	10.3000	3.5000	7
4)	4.0000	9.4000	7.8000	5.0000	8
5)	3.3000	6.6000	5.4000	4.0000	9
6)	3.7000	4.2000	6.4000	4.0000	10
7)	1.6000	8.4000	5.6000	4.5000	11
8)	2.5000	4.8000	4.6000	3.7000	17
9)	4.8000	10.0000	10.0000	6.0000	19
10)	3.4000	7.0000	6.6000	4.8000	20
11)	3.2000	13.0000	2.6000	7.5000	2
12)	2.5000	9.2000	4.0000	5.0000	4
13)	4.4000	12.8000	8.0000	8.0000	16
14)	4.1000	9.8000	7.2000	7.2000	18
15)	2.5000	18.4000	9.6000	9.6000	1
16)	1.5000	5.2000	5.9000	5.9000	3
17)	2.4000	10.6000	7.3000	7.3000	12
18)	2.7000	12.0000	8.5000	8.5000	13
19)	1.9000	10.0000	9.2000	9.2000	14
20)	3.8000	15.0000	10.5000	10.5000	15

Each series was then correlated with each other and the results output in the form of a correlation matrix. The calculations were based upon the "Welford updated sums of squares" technique. (Details given in Appendix (11)).

CORRELATION MATRIX V1.0

	1	2	3	4	Series No.	Mode
1)	1.00000				1	Axial load
2)	0.02091	1.00000			2	Torsion
3)	0.36797	0.24918	1.00000		3	Bending
4)	-0.00079	0.86241	0.41331	1.00000	4	Wave Height

The output, shown above, did not immediately appear to indicate a great degree of correlation, except that between wave height and torsion, a coefficient of 0.86 being indicated.

On reviewing this situation however, it was decided to introduce a further variable into the calculation, internal hose pressure. Three further correlations were run, each relating to an internal pressure range.

$$P = 90 - 150 \text{ lbf/in}^2$$

	1	2	3	4
1)	1.00000			
2)	0.14249	1.00000		
3)	0.74105	0.10054	1.00000	
4)	0.64644	0.71912	0.45938	1.00000

$P = 40 - 90 \text{ lbf/in}^2$

	1	2	3	4
1)	1.00000			
2)	0.40047	1.00000		
3)	0.84671	-0.03933	1.00000	
4)	0.83802	0.79029	0.42850	1.00000

$P = 10 - 40 \text{ lbf/in}^2$

	1	2	3	4
1)	1.00000			
2)	0.73708	1.00000		
3)	0.74560	0.82308	1.00000	
4)	0.74560	0.82308	1.00000	1.00000

Immediately it may be noted that in most cases the coefficients have increased in magnitude by such group segregation.

Using the 't' test in order to test the significance of the results:-

$$t = \frac{r \quad N - 2}{\sqrt{1 - r^2}}$$

where r = correlation coefficient
N = No. of sample (size)
t = Students 't' value

The coefficients relating to significance levels of 5% , 1% and 0.1% are listed for each test.

PRESSURE RANGE	COEFFICIENTS AT SIGNIFICANCE LEVELS		
	5%	1%	0.1%
$90 - 150 \text{ lbf/in}^2$	0.63	0.76	0.87
$40 - 90 \text{ lbf/in}^2$	0.96	0.99	-
$0 - 40 \text{ lbf/in}^2$	0.88	0.96	-

Although not highly significant it may be seen that the correlations would appear to carry significance levels of 5-10%.
Because of the relatively low significance of the results, (probably because of the small sample size), it was decided not to proceed to a regression analysis but to merely plot the results and thereby estimate the regression line, (Figure (35)).

Reviewing Figure (35):-

(a) AXIAL LOAD

Three distinct groups of data points are apparent each relating to a pressure range band, loads increasing with increasing pressure. It may be noted that considerable scatter exists, however it should be remembered that the lines indicated refer to quite broad pressure bands rather than discrete values.
Using the lower pressure range grouping as a datum it may be determined that the middle and upper bands represent load increases of approximately 120% and 230% respectively.
It is shown elsewhere within this thesis (VIII.2) that tensile stiffness is also a function of internal pressure, the following results relating a hose of similar design to that used in the Forcados project:-

PRESSURE (lbf/in^2)	STIFFNESS (Ton)	AVERAGE STIFFNESS
0	24×10^2	
		$26.5\ 10^2$
50	29×10^2	
		$33.0\ 10^2$
100	37×10^2	
		$41.0\ 10^2$
150	45×10^2	

It may be seen that whilst conforming qualitatively with the load results, increasing stiffness only explains some 20% of the load increases in either case. As yet the effect of compound situations, ie: bending + tension etc. and the result changes in stiffness etc. have not been evaluated. The latter may produce additional information in explaining the situation described.

(b) TORSIONAL LOAD:

Whilst considerable scatter of the plotted points exist, no distinct groupings or trends are apparent.

Referring to stiffness evaluation tests again (XI.8), in the case of torsional stiffness initial test rig results have indicated this property to be independant of internal pressure. A typical value for the hose under consideration approximates to 52.0×10^2 Ton ft^2. Using this value it may be calculated that the corresponding deflections relating to the torques measured would fall within the range $\pm$ 0.2 deg/ft. Thus over the first hose element a differential twist of $\pm$ 6 deg would be encountered. This figure compares very well with the visually (from the photographs) estimated value of $\pm$ 10 deg.

(c) BENDING MOMENTS:

As in the case of axial load, distinct "pressure range groupings" are apparent. The relationships are not as clear in this case, although that may well reflect the positioning of point Δ_2, which would appear to be suspect.

Again taking the lower pressure rating group as a datum, the middle and higher pressure groupings would indicate increased loads of 40% and 100% respectively.

Once again referring to stiffness properties, it can be shown that bending stiffness increases with both increasing pressure and radius of curvature. For various radii of curvature (ie: 10ft - 35 ft) it may be shown that the changes in stiffness, presented in the equivalent manner to the above loads, result in increases of 30% and 70% respectively, (refer to Chapter VIII.1).

(NB: The stiffness properties denoted in this chapter and/or the relationship(s) with pressure etc., were not available at the time of conducting the Forcados tests. But were evaluated subsequent to that period in order to explain certain of the phenomena observed.

V.7 SUMMARY/CONCLUSIONS:

(a) It was most gratifying to prove that the approach adopted and the equipment designed and built for the project enabled the required information to be generally obtained. Problems were encountered but generally these were overcome without seriously affecting the project results.

The most problematical and detrimental component equipment failure was the wave recorder, (provided by the host oil company). This failure further illustrates the design philosophy which must be employed, (and indeed was employed for the other components), for equipment for use in hostile environments:-

(1) Built-in redundancy or back-up should be provided.

(2) "On-the-spot" analysis of results should be possible/conducted, in order to immediately assess the validity of results/equipment malfunctions etc., thereby avoiding expensive periods of no productivity.

(b) The results obtained suggested that the problem of rapid hose failure of the buoy manifold hose is primarily due to fatigue rather than excessive loads being encountered which exceed the design limits or capabilities of the hose. That is not to say that such occurrences do not or indeed cannot take place under conditions more severe than those encountered during the test period.

(c) The most significant features of the results obtained were:-

(1) The presence and magnitude of torsional forces acting on the hose. (Prior to this study torsional forces occurring in service were regarded as negligible and consequently had historically been ignored in the design/study of hose).

(2) The predominance of the shorter period waves, (ie: 3-7 second period) in generating the most significant differential buoy/hose motion and resultant higher stress levels.

(NB: This now would seem to explain the discrepancy between hose failures off W. Africa as compared to the Arabian Gulf, the latter having a significantly greater failure rate. Initially wave conditions were considered to be similar, indeed W. Africa appeared to be marginally worst. However wave period data would now indicate that much shorter period waves predominate in the Arabian Gulf).

(3) The significance of internal pressurisation, or rather its effect on other properties or induced loads was identified; thereby encouraging further studies into this aspect.

(d) Sufficient data was obtained upon which to base the design of a full scale simulation fatigue test rig.

(e) Initial data was obtained to enable a basic prediction model to be implemented with reference to predicting system loads from site wave data. It is indeed realised that this aspect is far from complete and requires additional field study(s) together with more accurately defined wave data.

V.8 FINANCIAL EFFECT

The costs associated with the Forcados project were as follows:-

		£
1.	Load measuring spool piece 3 off including, structure gauging, testing etc.	4,000
2.	Data logging equipment including recorder, playback unit, radio equipment, housing etc.	7,000
3.	Camera recording unit including camera, housing synchronisation equipment etc.	3,000
4.	Transportation of equipment to Nigeria:	2,500
5.	Analysis of results, computing time etc.	800
6.	Miscellaneous associated 'on' costs	1,500
	TOTAL STUDY COST:	18,800

The above total was divided on a fifty percent share basis between the two companies involved, ie: Dunlop and Shell BP.

NB: Costs such as the following were paid directly by the individual companies:-

DUNLOP : Test Personnel Salaries.
Equipment design overheads etc.
Test personnel travel and associated expenditure in Nigeria etc.

SHELL-BP : Installation of Equipment on Buoy.
Buoy/Terminal access costs (eg: launch time etc)
Personnel accommodation charges.

The resulting cost to Dunlop was thus:-

	£
50% share of Capital charges:	9,400
Peripheral expenditure as indicated above:	5,000
TOTAL:	14,400

RETURN ON INVESTMENT:

At the time of conducting the above study Dunlop held an approximate fifty percent share of the Shell-BP hose market.

As a consequence of the Forcados study, in terms of goodwill, technical achievement, product improvement* etc; Dunlop have since that period obtained virtually one hundred percent of Shell-BP's business. (Naturally they periodically purchase competitors products for comparative evaluation purposes).

At 1977 prices the above market is valued at £450,000 per annum. Thus the gross contribution to Company profits over the four years since the study is approximately £300,000.

* As a result of the information gained from the study, particularly with regard to the magnitude (measured) and concentration (photographic evidence) of the bending loads a re-design of the off-buoy hose was initiated. This briefly comprised of stiffening the reinforced (buoy) end of the hose, extending the reinforcement further into the body of the hose and tapering the reinforcement/(stiffness) over a much greater length of hose than previously.

Field feedback of the above hose design from Forcados has indicated the following results:-

Average Service Life (Original Design) - 3- 6 months

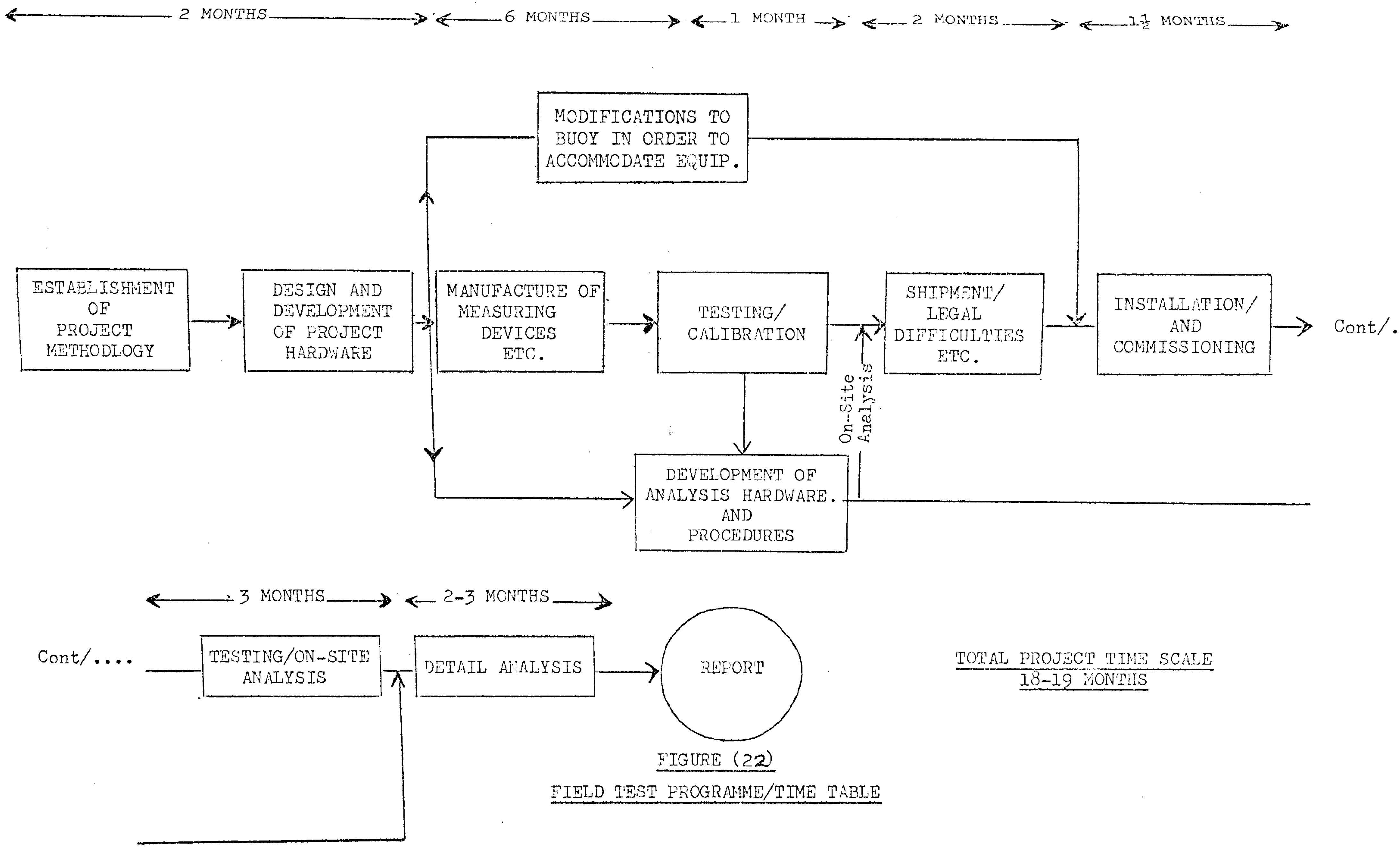
Average Service Life (Revised Design) - 12-18 months +

It may readily be seen that the increase in service life achieved was somewhat greater than 300%.

FIGURE (21)

SHELL-BP FORCADOS BUOY

'AVERAGE YEAR'



TOTAL PROJECT TIME SCALE
18-19 MONTHS

FIGURE (22)

FIELD TEST PROGRAMME/TIME TABLE

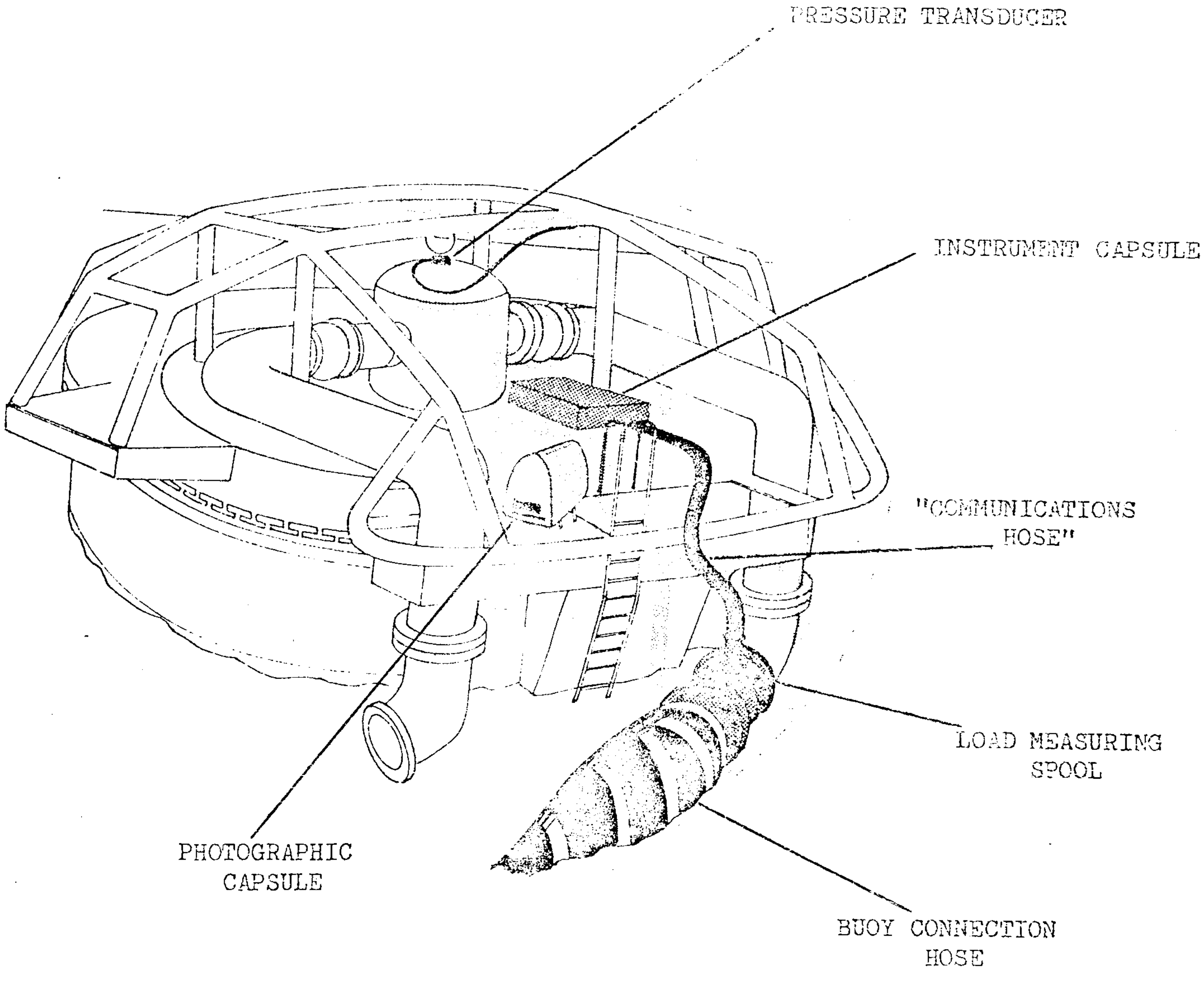


FIGURE (23)

LOCATION OF EQUIPMENT ON SPM

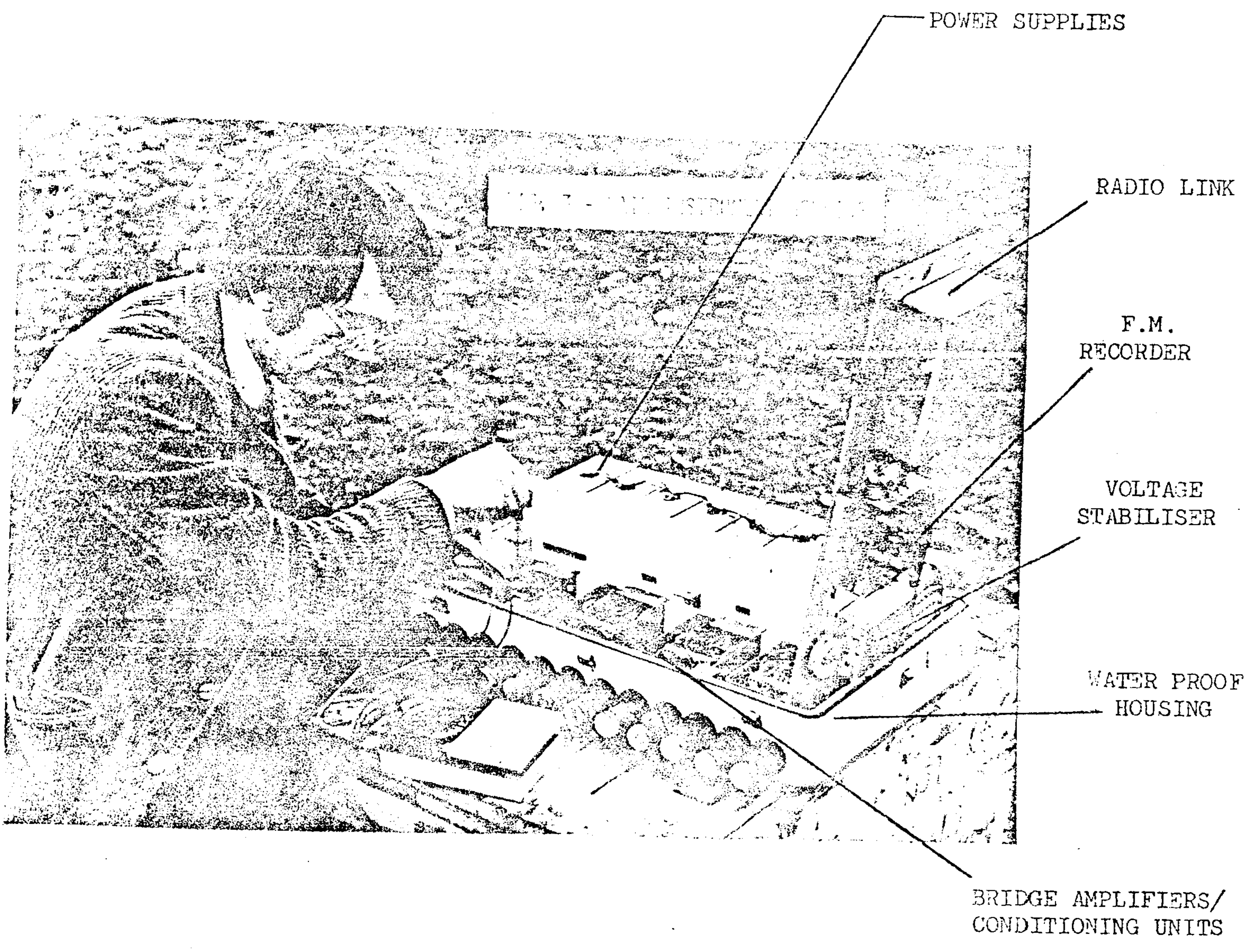


FIGURE (24)

INSTRUMENT CAPSULE.

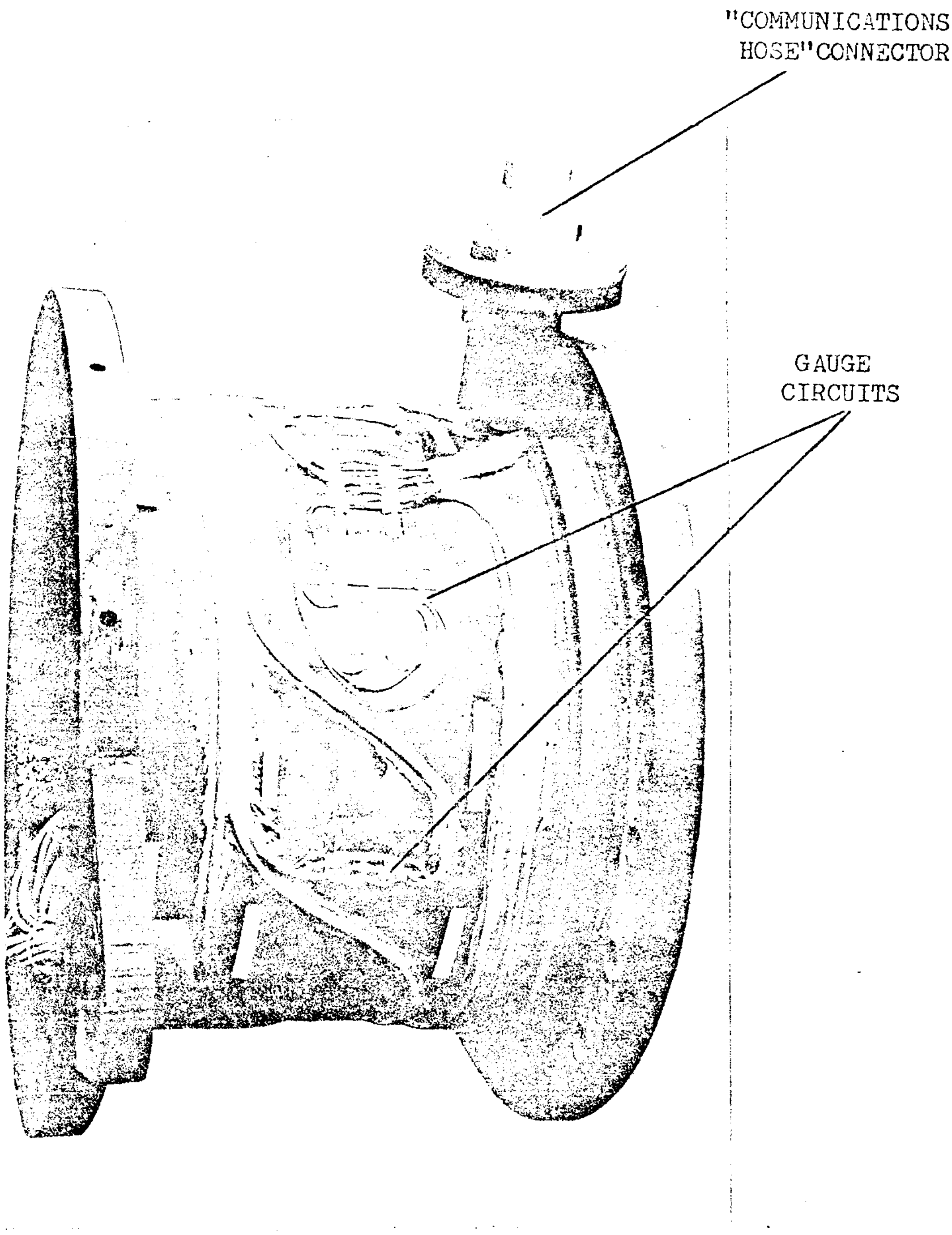


FIGURE (25)

SPOOL (PRIOR TO ENCAPSULATION

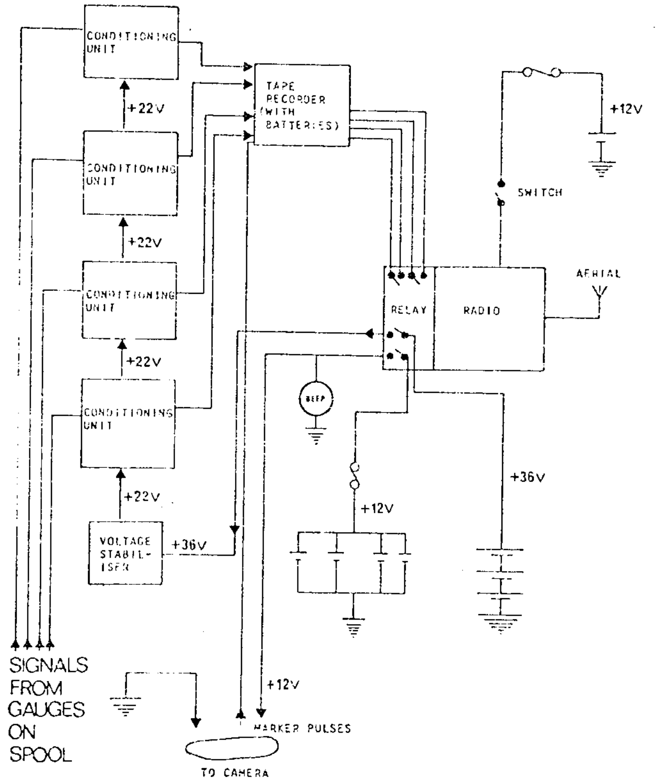


FIGURE (26)

INSTRUMENT CAPSULE - BLOCK DIAGRAM

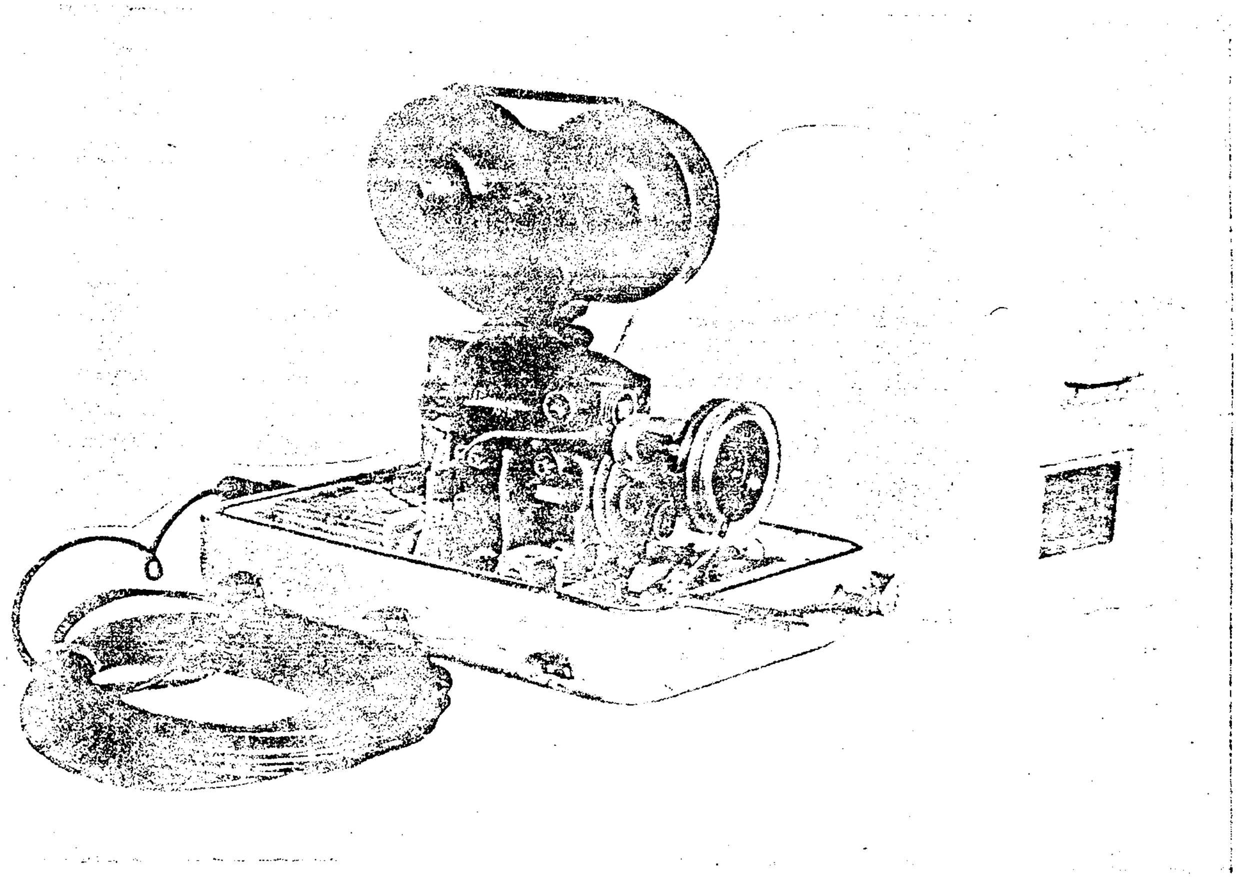


FIGURE (27)

PHOTOGRAPHIC SYSTEM AND HOUSING

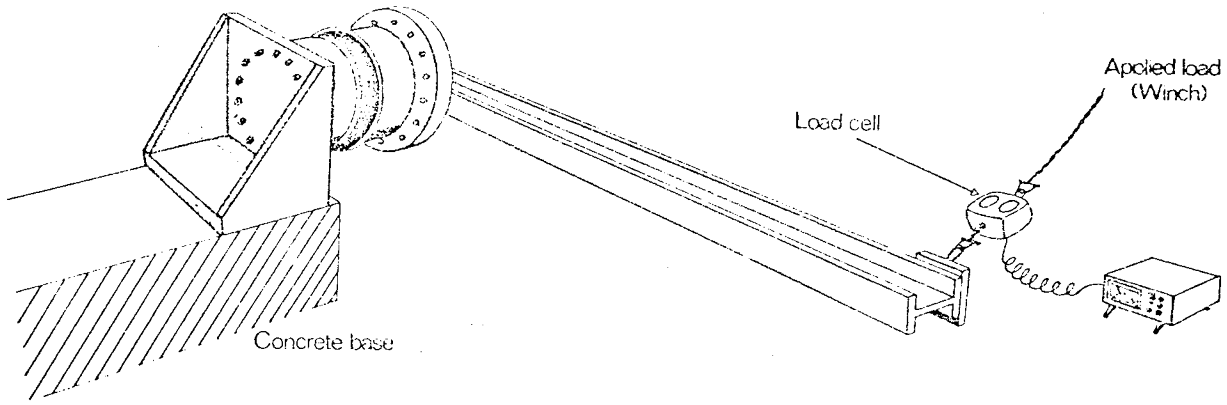
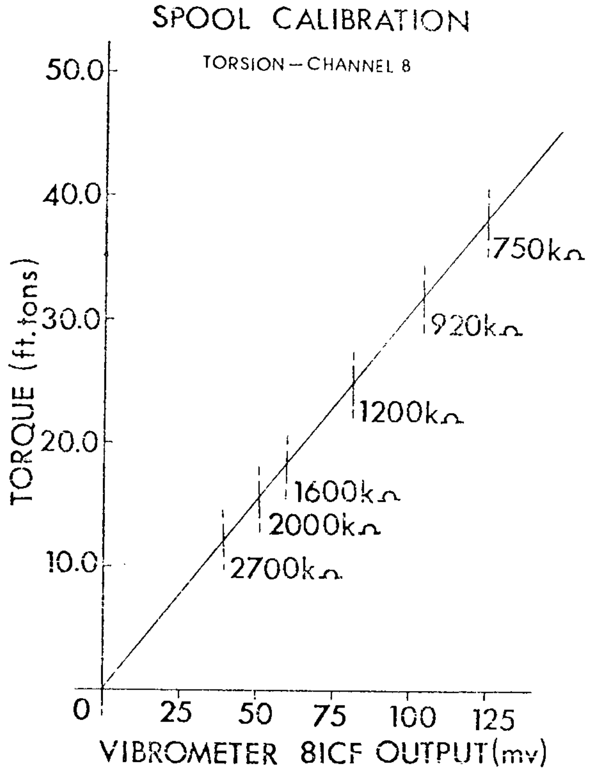


FIGURE (28)

SPOOL CALIBRATION



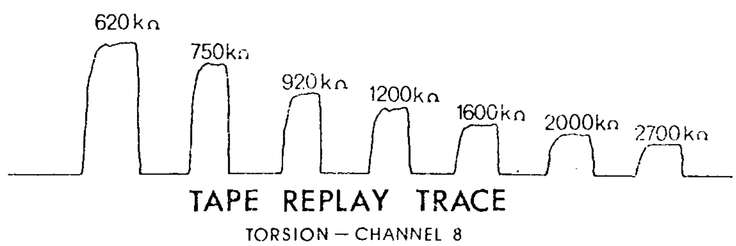


FIGURE (29)

TYPICAL CALIBRATION CURVE/RESISTOR EQUIVALENT

ANALYTICAL ZONE

(Approx. 15 min)

DATA RECORDER → LOW PASS FILTER → UV RECORDER → 1CH 2Ch ⋮ 4CH

(1) Marine & Weather conditions recorded.

(2) Simultaneous photographic deflection data.

(3) After seeing wave profile in conjunction with (1), (2) a zone was selected for detailed analysis.

DATA RECORDER → LOW PASS FILTER → SAMPLE +A-D CONVERTER → COMPUTER

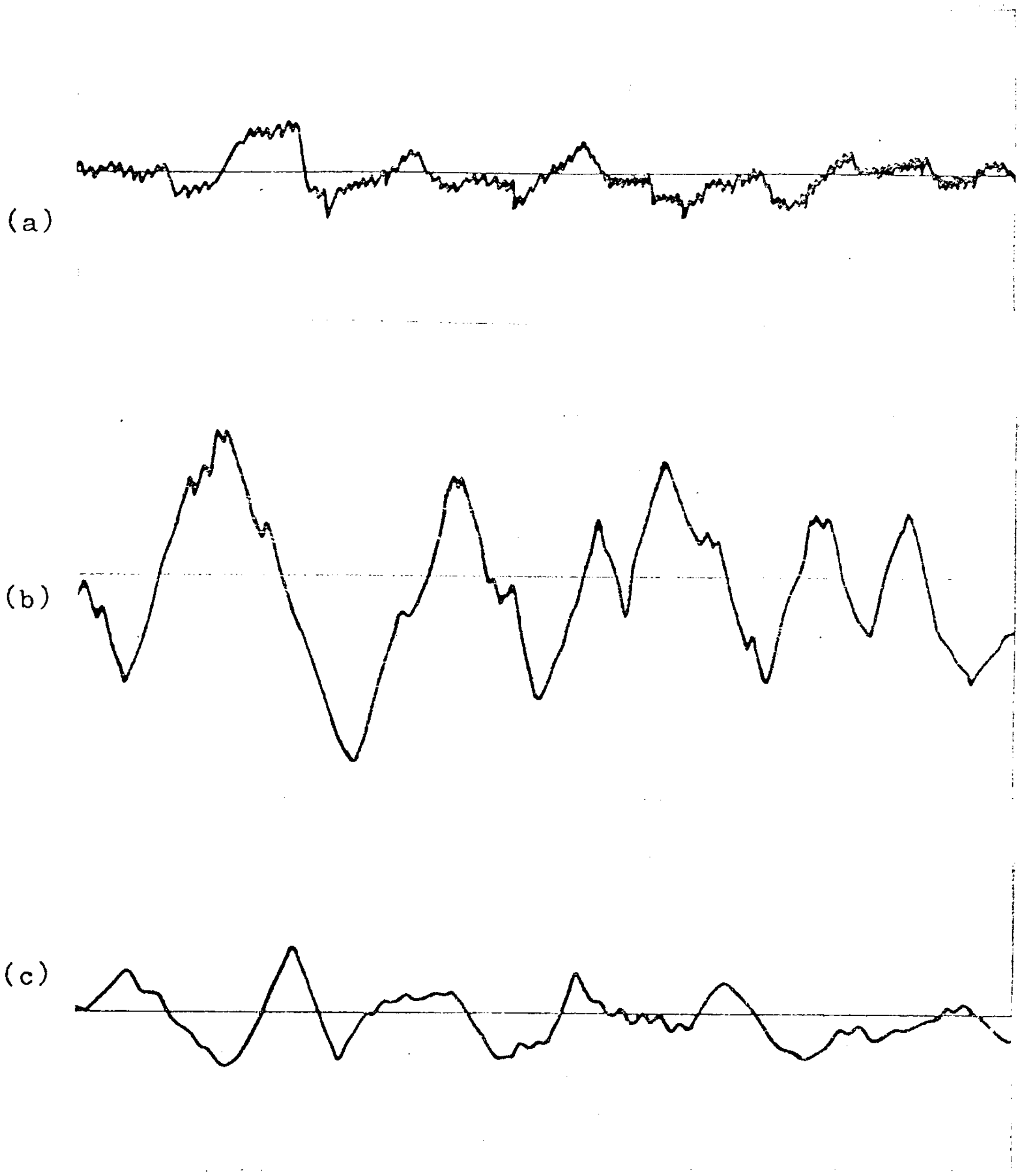
Input of analytical zone.

Noise Eradicator

Output Mean, RHS, Std. Dev. etc.

FIGURE (30)

DATA ANALYSIS SCHEMATIC



Loadings: (a) AXIAL - 1Cm - 9.3 TON
(b) TORSION - 1Cm - 9.7 TON FT.
(c) V.BENDING - 1Cm - 12.0 TON FT.

TAPE: 005/03
Date: 14:10:73
Time Scale: 1Cm -2.26 SEC.

FIGURE (31)

U.V. LOADING TRACE

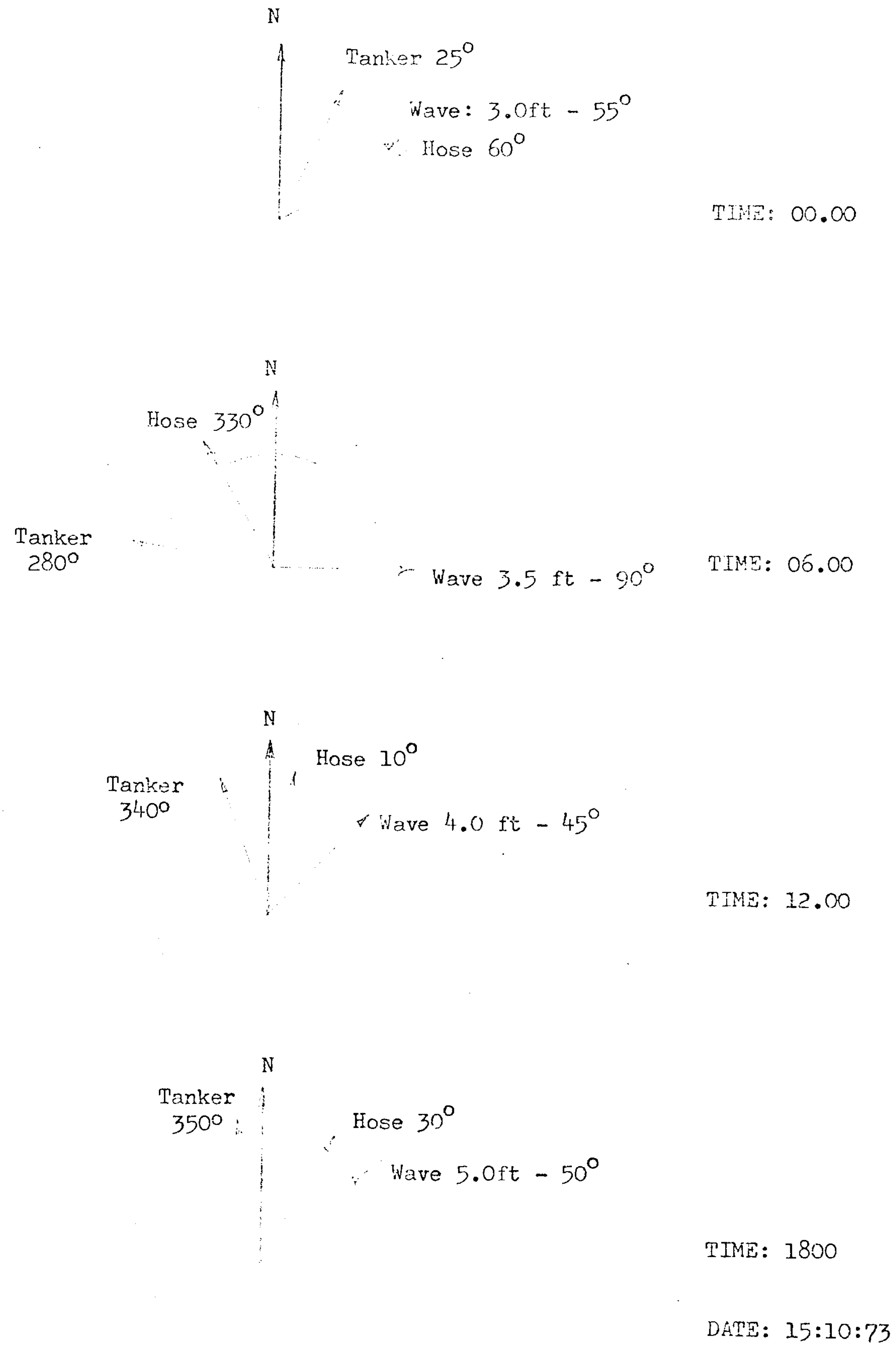
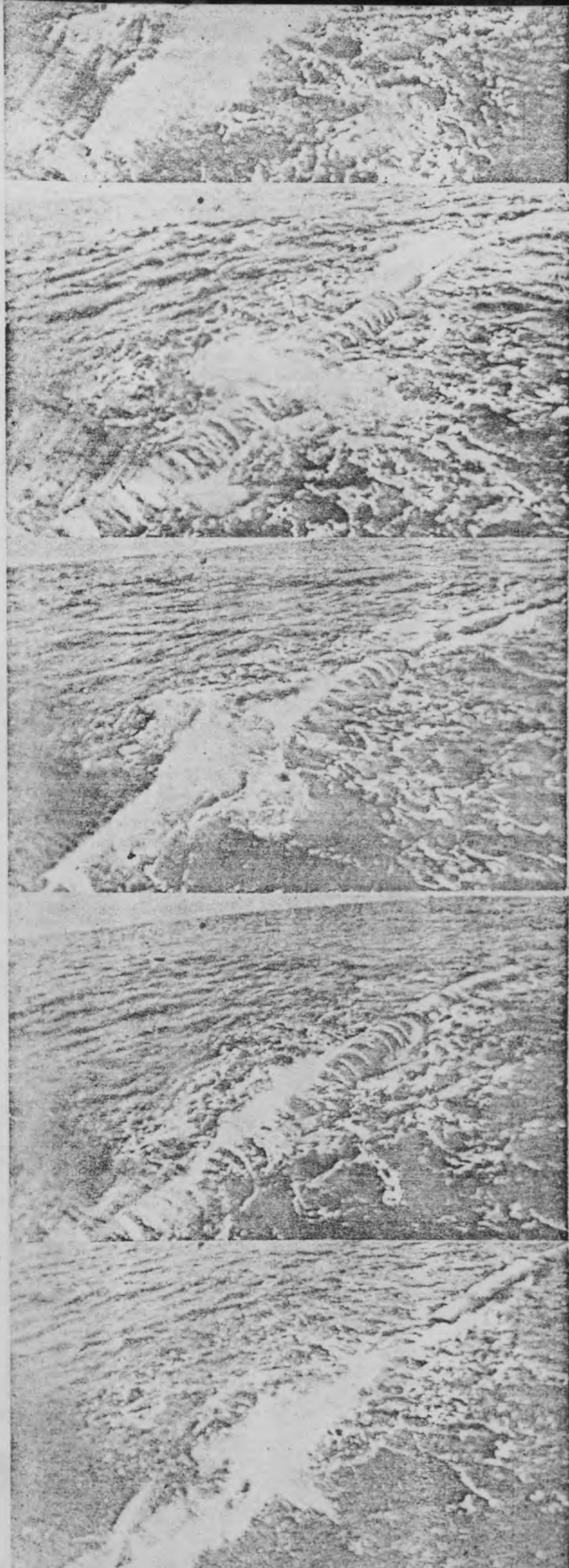


FIGURE (32)

SEA STATE SCHEMATICS

TEST 005/02
10.10.73



1

2

2.6S INTERVAL

3

4

5

FIGURE (33)

DUNLOP FORCADOS HOSE TESTS

TAPE 701 DATE 29.10.73 DURATION 15.16 – 15.40

SUMMARY STATS

SER. NO.	MEAN	RMS	STD DEV	MAX	MIN	RANGE
1	23.8	24.1	4.2	38.1	13.8	24.2
2	-32.5	32.9	4.6	-20.0	-49.6	29.6
3	-46.38	46.48	3.14	-34.88	-54.9	20.0

TIME PLOT OF FIRST 30 TERMS IN SERIES

XIAL LOAD (TONS) TORSION (FT. TON) VERT. BEND. (FT. TON)

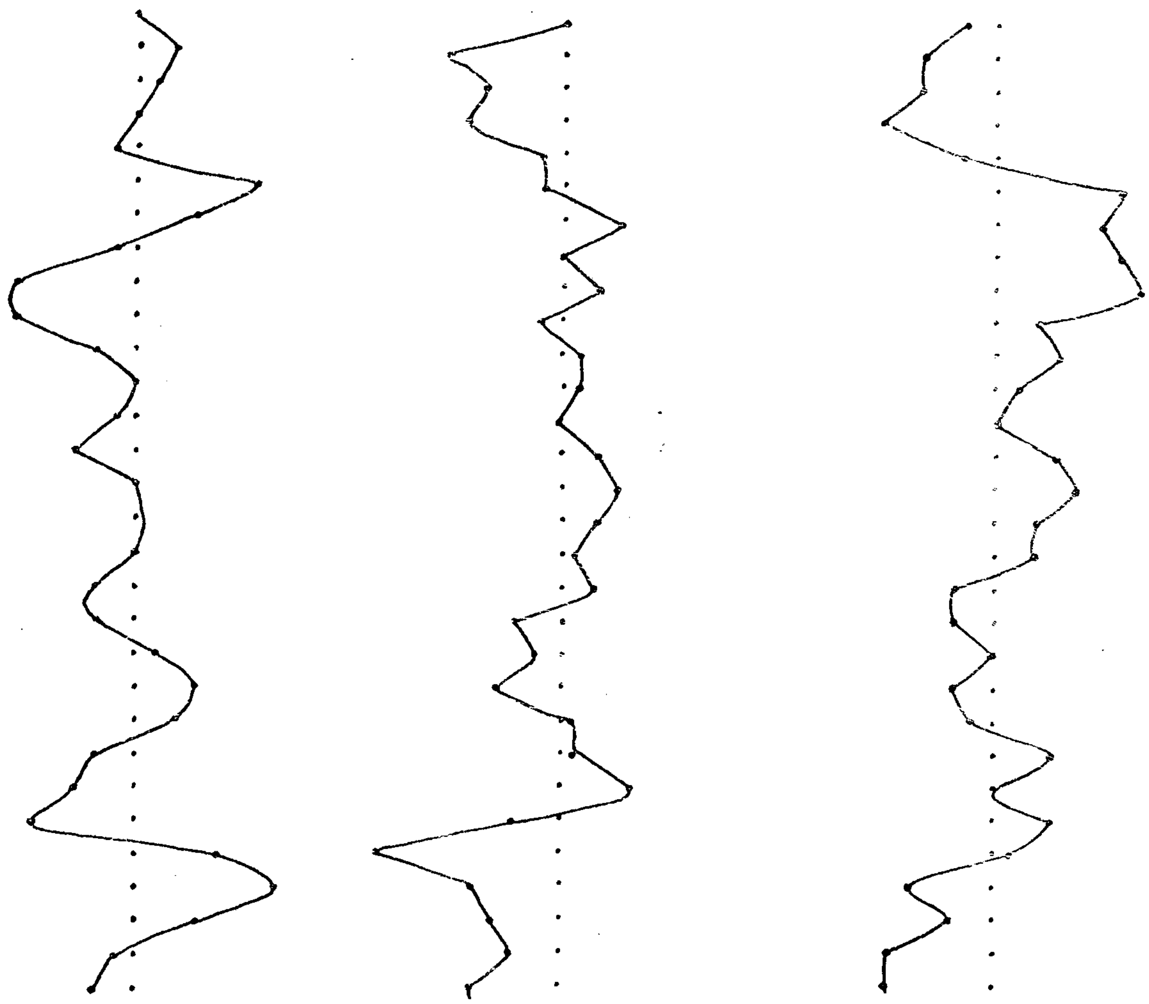


FIGURE (34)(a)

SER. NO. 1 PROBABILITY FUNCTIONS 30 CLASS INTERVALS

Mid Mark	Freq	HISTOGRAM	Gaussian	HISTOGRAM
-9.52	0.3%		0.6%	
-8.71	1.4%	XX	0.9%	X
-7.90	1.5%	XX	1.3%	X
-7.09	1.8%	XX	1.9%	XX
-6.29	4.6%	XXXXXX	2.5%	XXX
-5.48	4.1%	XXXXXX	3.3%	XXXX
-4.67	5.4%	XXXXXXXX	4.1%	XXXXXX
-3.87	6.7%	XXXXXXXXXX	5.0%	XXXXXXX
-3.06	6.4%	XXXXXXXXX	5.9%	XXXXXXXX
-2.25	6.6%	XXXXXXXXX	6.6%	XXXXXXXXX
-1.44	7.3%	XXXXXXXXXX	7.2%	XXXXXXXXXX
-0.64	8.5%	XXXXXXXXXXXX	7.5%	XXXXXXXXXXX
0.17	7.8%	XXXXXXXXXXX	7.6%	XXXXXXXXXXX
0.98	5.4%	XXXXXXXX	7.4%	XXXXXXXXXXX
1.78	6.4%	XXXXXXXXX	7.0%	XXXXXXXXXX
2.59	5.1%	XXXXXXX	6.3%	XXXXXXXXX
3.40	5.4%	XXXXXXXX	5.5%	XXXXXXXX
4.21	2.9%	XXXX	4.6%	XXXXXX
5.01	3.2%	XXXX	3.8%	XXXXX
5.82	1.9%	XX	2.9%	XXXX
6.63	2.4%	XXX	2.2%	XXX
7.43	1.6%	XX	1.6%	XX
8.24	1.2%	X	1.1%	X
9.05	0.6%		0.8%	X
9.86	0.1%		0.5%	
10.66	0.7%	X	0.3%	
11.47	0.1%		0.2%	
12.28	0.4%		0.1%	
13.09	0.2%		0.1%	
13.89	0.1%		0.0%	

SER. NO. 2 PROBABILITY FUNCTIONS 30 CLASS INTERVALS

Mid Mark	Freq	HISTOGRAM	Gaussian	HISTOGRAM
-16.54	0.3%		0.0%	
-15.55	0.0%		0.0%	
-14.57	0.2%		0.1%	
-13.58	0.5%		0.1%	
-12.59	0.3%		0.2%	
-11.61	1.1%	X	0.4%	
-10.62	0.8%	X	0.6%	
- 9.63	1.3%	X	0.9%	X
- 8.65	1.9%	XX	1.5%	XX
- 7.66	2.0%	XXX	2.1%	XXX
- 6.67	2.9%	XXXX	3.0%	XXXX
- 5.69	3.7%	XXXXX	4.0%	XXXXX
- 4.70	5.2%	XXXXXXX	5.1%	XXXXXXX
- 3.71	5.9%	XXXXXXXX	6.2%	XXXXXXXXX
- 2.73	7.7%	XXXXXXXXXXX	7.2%	XXXXXXXXXX
- 1.74	7.4%	XXXXXXXXXXX	8.0%	XXXXXXXXXXX
- 0.76	8.4%	XXXXXXXXXXXX	8.4%	XXXXXXXXXXXX
0.23	8.5%	XXXXXXXXXXXX	8.5%	XXXXXXXXXXXX
1.22	7.8%	XXXXXXXXXXX	8.3%	XXXXXXXXXXXX
2.20	7.8%	XXXXXXXXXXX	7.6%	XXXXXXXXXXX
3.19	7.8%	XXXXXXXXXXX	6.7%	XXXXXXXXXX
4.18	6.5%	XXXXXXXXX	5.7%	XXXXXXXX
5.16	4.7%	XXXXXXX	4.5%	XXXXXX
6.15	3.0%	XXXX	3.5%	XXXXX
7.14	2.0%	XXX	2.6%	XXX
8.12	1.5%	XX	1.8%	XX
9.11	0.6%		1.2%	X
10.09	0.2%		0.8%	X

SER. NO. 3 PROBABILITY FUNCTIONS 30 CLASS INTERVALS

Mid Mark	Freq	HISTOGRAM	Gaussian	HISTOGRAM
8.19	0.4%		0.3%	
7.52	0.2%		0.5%	
6.85	1.1%	X	0.8%	X
6.18	1.2%	X	1.2%	X
5.52	2.2%	XXX	1.8%	XX
4.85	3.2%	XXXX	2.6%	XXX
4.18	4.6%	XXXXXX	3.5%	XXXXX
3.51	6.7%	XXXXXXXXXX	4.5%	XXXXXX
2.85	7.0%	XXXXXXXXXX	5.6%	XXXXXXXX
2.18	6.5%	XXXXXXXXX	6.6%	XXXXXXXXX
1.51	8.0%	XXXXXXXXXXXX	7.5%	XXXXXXXXXXX
0.84	7.3%	XXXXXXXXXX	8.2%	XXXXXXXXXXXX
0.18	9.1%	XXXXXXXXXXXXX	8.5%	XXXXXXXXXXXX
0.48	9.0%	XXXXXXXXXXXXX	8.4%	XXXXXXXXXXXX
1.15	6.7%	XXXXXXXXXX	7.9%	XXXXXXXXXXX
1.82	5.5%	XXXXXXXX	7.2%	XXXXXXXXXX
2.48	5.0%	XXXXXXX	6.2%	XXXXXXXXX
3.15	4.6%	XXXXXX	5.1%	XXXXXXX
3.82	3.8%	XXXXX	4.0%	XXXXXX
4.49	2.2%	XXX	3.0%	XXXX
5.16	2.1%	XXX	2.2%	XXX
5.82	0.9%	X	1.5%	XX
6.49	1.2%	X	1.0%	X
7.16	0.9%	X	0.6%	
7.82	0.2%		0.4%	
8.49	0.0%		0.2%	
9.16	0.2%		0.1%	
9.83	0.1%		0.1%	
10.49	0.1%		0.0%	
11.16	0.1%		0.0%	

SER. NO. 1 AUTOCORRELATION MAX.LAG.28

LAG(S)	AUTO. CORR.
0.0	1.000
0.7	0.614
1.4	0.071
2.1	-0.111
2.8	0.106
3.5	0.337
4.2	0.323
4.9	0.101
5.6	-0.038
6.3	0.056
7.0	0.218
7.7	0.287
8.4	0.186
9.1	0.106
9.8	0.081
10.5	0.144
11.2	0.171
11.9	0.160
12.6	0.110
13.3	0.101
14.0	0.113
14.7	0.119
15.4	0.131
16.1	0.129
16.8	0.151
17.5	0.134
18.2	0.123
18.9	0.091

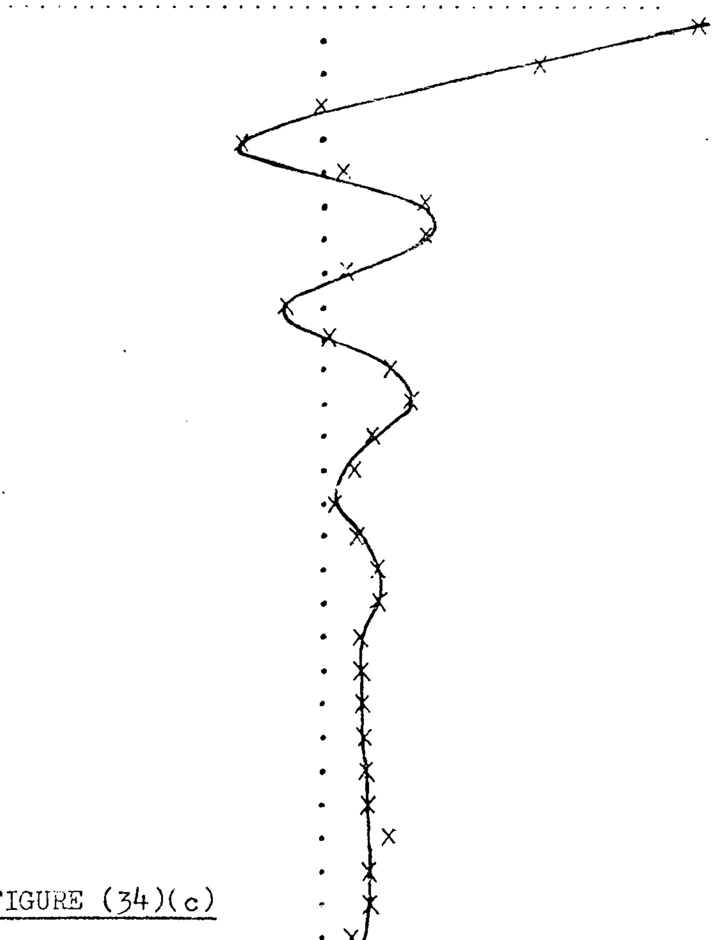
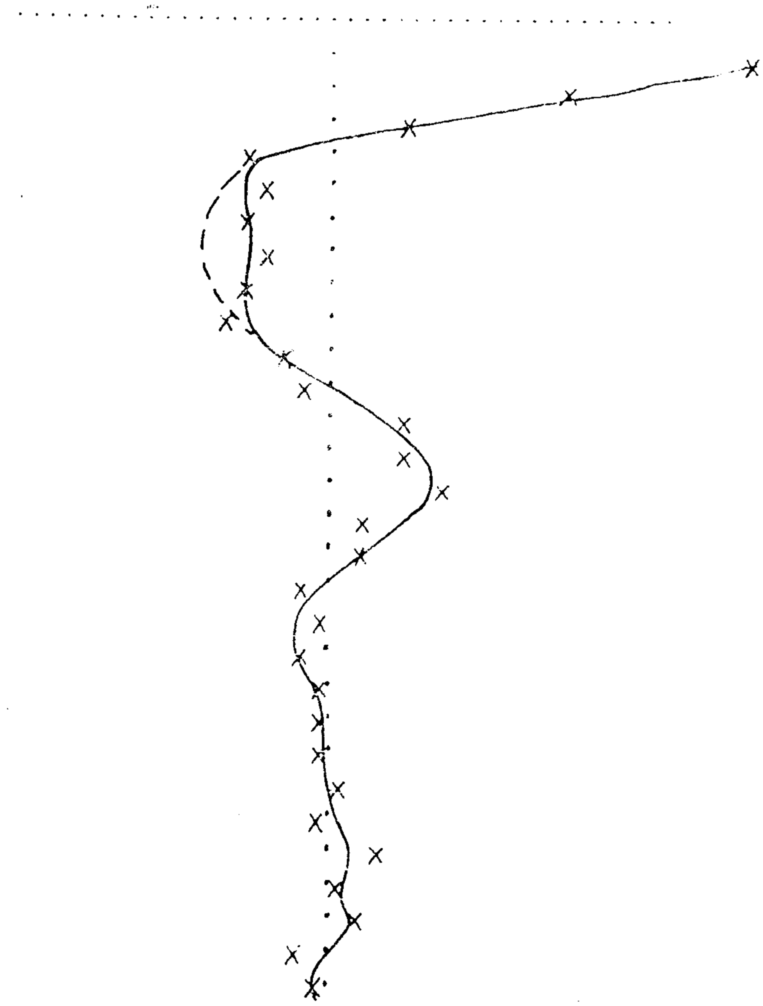


FIGURE (34)(c)

SER. NO. 2 AUTOCORRELATION MAX.LAG.28

LAG(S)	AUTO. CORR.
0.0	1.000
0.7	0.569
1.4	0.181
2.1	-0.214
2.8	-0.180
3.5	-0.212
4.2	-0.196
4.9	-0.233
5.6	-0.253
6.3	-0.139
7.0	-0.091
7.7	0.170
8.4	0.162
9.1	0.269
9.8	0.058
10.5	0.078
11.2	-0.072
11.9	-0.007
12.6	-0.082
13.3	-0.035
14.0	-0.012
14.7	-0.005
15.4	0.042
16.1	-0.003
16.8	0.120
17.5	0.006
18.2	0.077
18.9	-0.088
19.6	0.006



SER. NO. 3 AUTOCORRELATION MAX.LAG.28

LAG(S)	AUTO. CORR.
0.0	0.999
0.7	0.638
1.4	0.325
2.1	0.121
2.8	-0.191
3.5	-0.108
4.2	0.032
4.9	0.076
5.6	0.308
6.3	0.252
7.0	0.032
7.7	-0.035
8.4	-0.279
9.1	-0.318
9.8	-0.087
10.5	-0.024
11.2	0.205
11.9	0.385
12.6	0.168
13.3	0.084
14.0	-0.094
14.7	-0.305
15.4	-0.168
16.1	-0.094
16.8	0.056
17.5	0.355
18.2	0.291
18.9	0.168

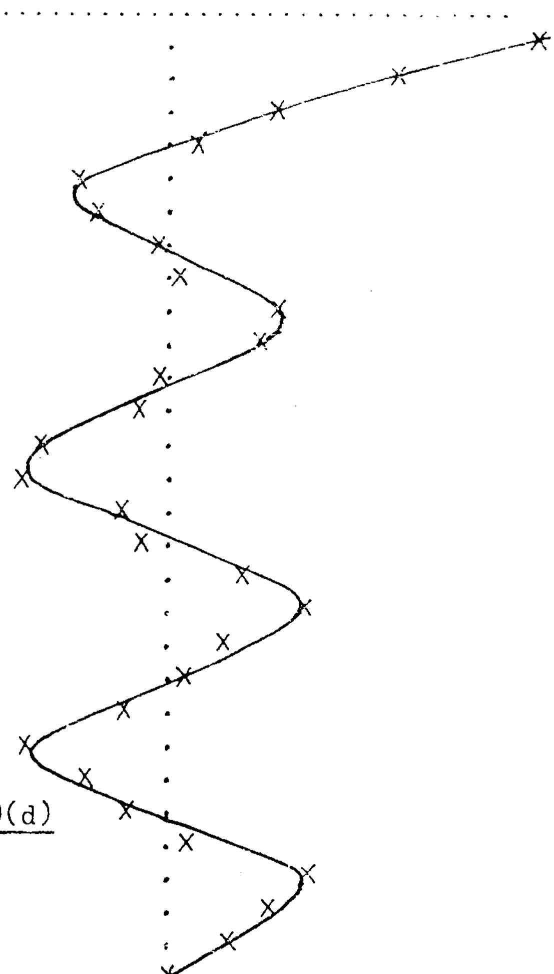
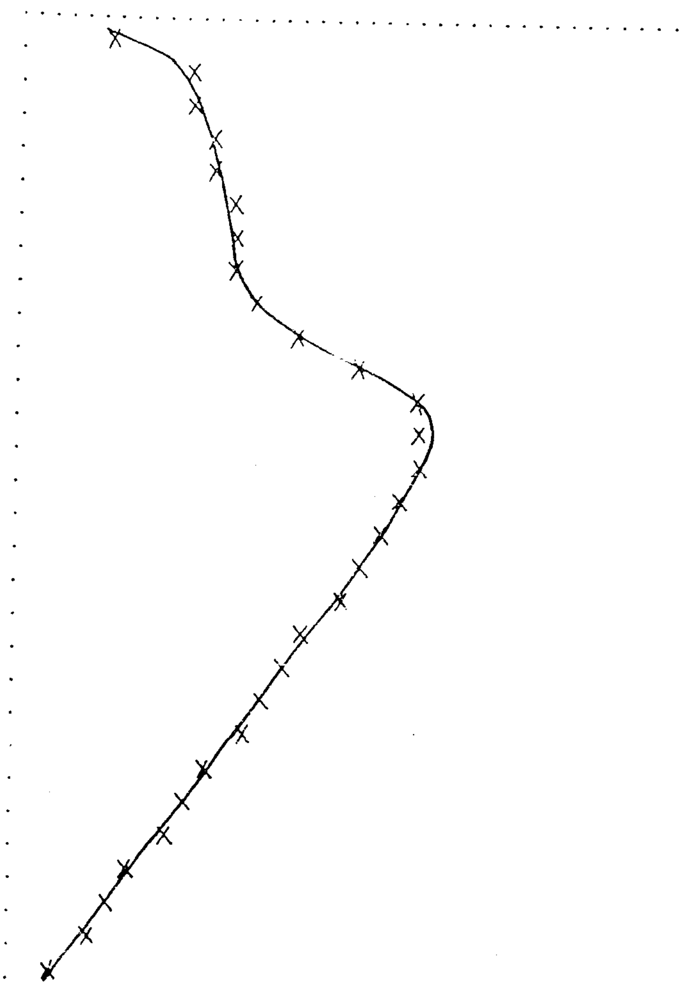


FIGURE (34)(d)

SER. NO. 1 PERIODOGRAM

FREQ(HZ)	PSD
0.000	15.4
0.025	22.5
0.051	23.6
0.076	24.5
0.102	25.7
0.127	26.8
0.152	27.3
0.178	27.8
0.203	29.2
0.229	33.2
0.264	39.2
0.279	44.0
0.305	45.4
0.330	44.5
0.355	42.7
0.381	40.6
0.406	38.4
0.432	36.1
0.457	33.7
0.483	31.3
0.508	28.7
0.533	26.1
0.559	23.5
0.584	20.8
0.610	18.1
0.635	15.5
0.661	13.1
0.686	10.5
0.711	7.7



SER. NO. 2 PERIODOGRAM

FREQ(HZ)	PSD
0.000	1.7
0.025	0.6
0.051	1.3
0.076	6.2
0.102	15.4
0.127	23.6
0.152	26.7
0.173	27.6
0.203	29.8
0.229	33.0
0.254	35.0
0.279	35.4
0.305	34.7
0.330	33.6
0.335	32.2
0.381	30.2
0.406	27.8
0.432	25.3
0.457	22.7
0.483	20.2
0.508	17.6
0.533	15.0
0.559	12.3
0.584	9.6
0.610	
0.635	5.7
0.661	5.3
0.686	3.6
0.711	0.8

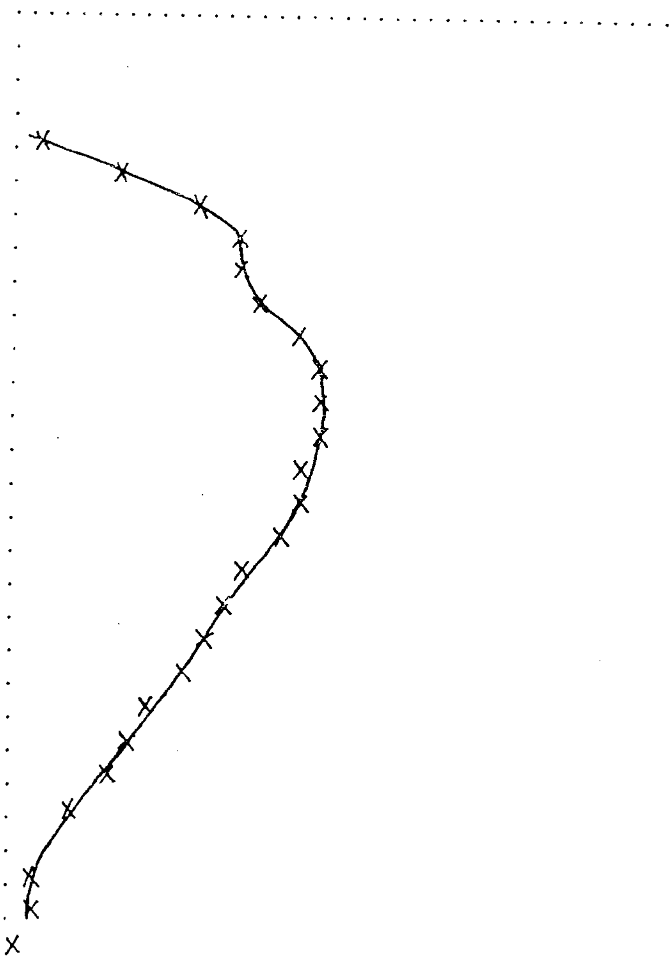
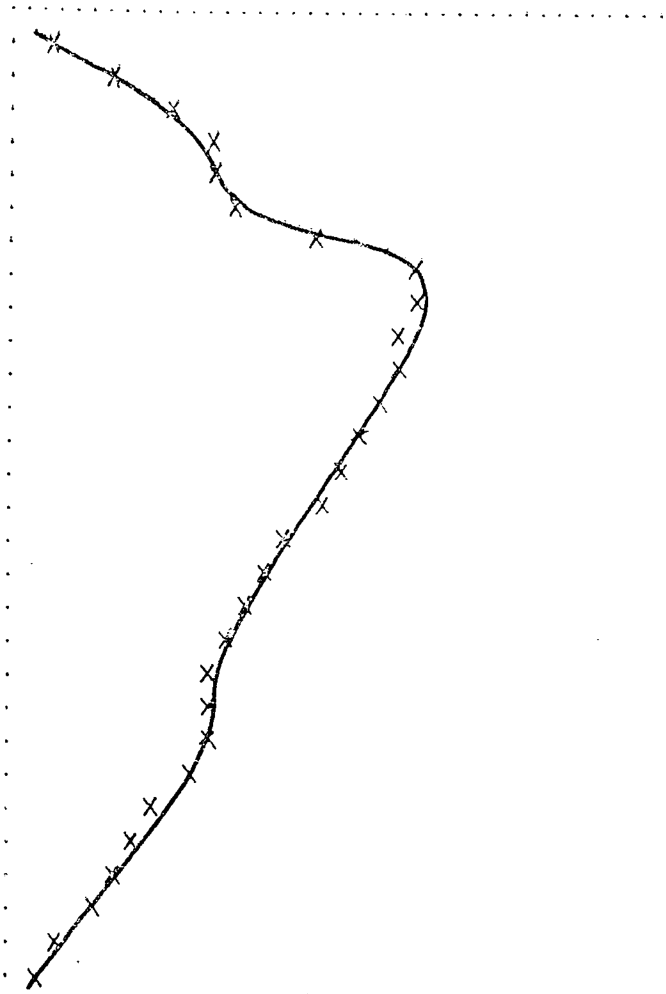


FIGURE (34)(e)

SER. NO. 3 PERIODOGRAM

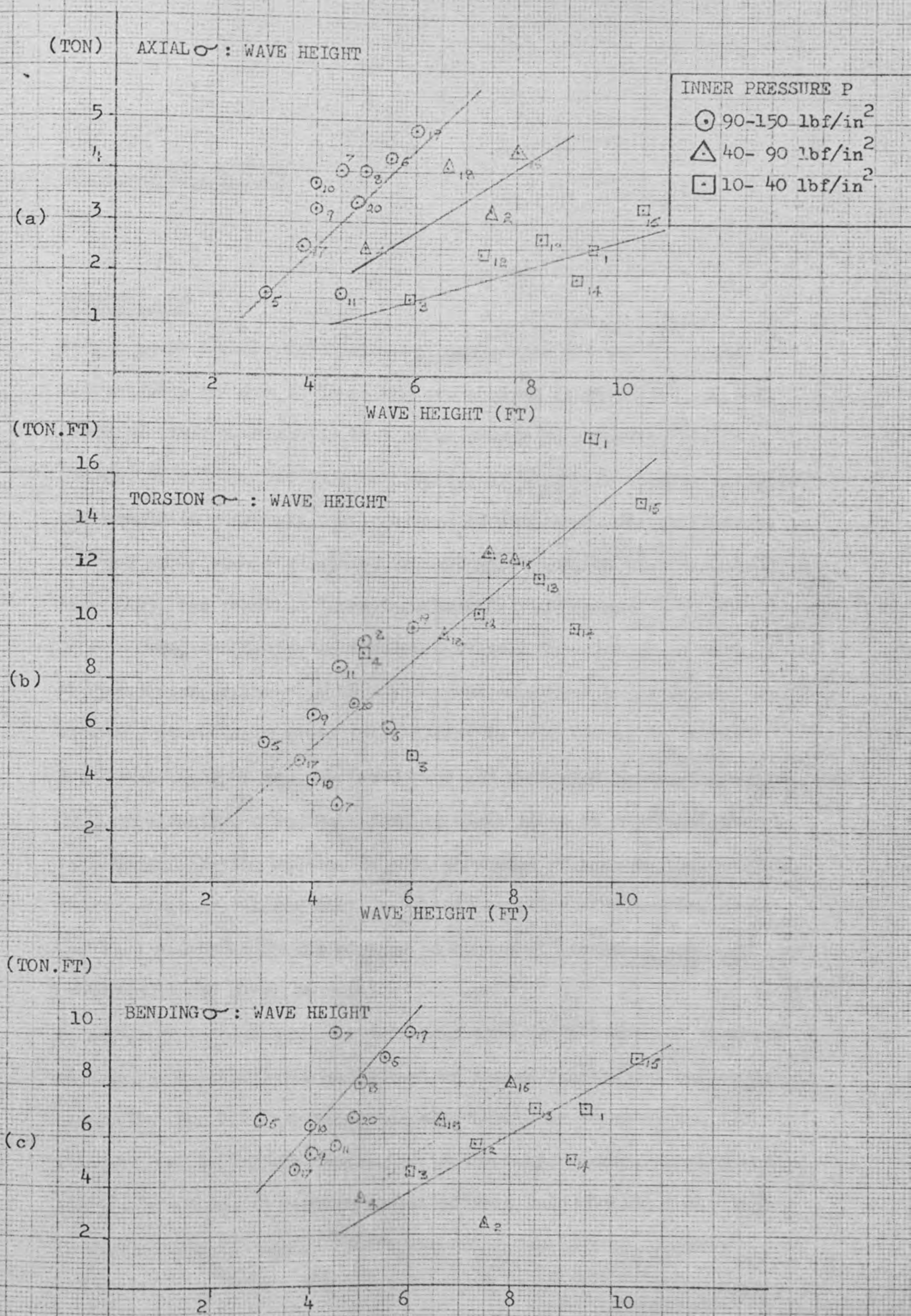
FREQ(HZ)	PSD
0.000	9.2
0.025	15.0
0.051	20.0
0.076	24.1
0.102	25.7
0.127	27.1
0.152	35.3
0.178	44.1
0.203	45.3
0.229	43.7
0.254	42.1
0.279	40.3
0.305	38.3
0.330	36.3
0.355	34.1
0.381	31.7
0.406	29.4
0.432	27.1
0.457	24.6
0.483	23.4
0.508	23.9
0.533	23.0
0.559	20.5
0.584	17.9
0.610	15.3
0.635	12.6
0.661	10.0
0.686	7.3
0.711	4.6



PRUS : 32 ELAPSED TIME : 1:13.70
NO EXECUTION ERRORS DETECTED

EXIT

FIGURE (34)(f)



(a)
(TON)
AXIAL σ : WAVE HEIGHT
INNER PRESSURE P
90-150 lbf/in²
40- 90 lbf/in²
10- 40 lbf/in²
WAVE HEIGHT (FT)
(b)
(TON.FT)
TORSION σ : WAVE HEIGHT
WAVE HEIGHT (FT)
(c)
(TON.FT)
BENDING σ : WAVE HEIGHT
WAVE HEIGHT (FT)

VI MODEL TESTING

INTRODUCTION

Single point mooring terminals are complex systems which present a formidable problem to those who are engaged in their design. Not least of these problems is that of underbuoy hose configuration design. Experience and opinion, neither of which are totally reliable, can result in extremely costly errors of judgement. The fallibility of such experience and opinion is well known and, on considerable occasions, has resulted in endless trouble and expense.

In reviewing the design techniques available in relation to such system design engineering the writer was staggered to find how crudely, potentially fundamental decisions were taken. As stated previously much of such system design was conducted in drawing offices by a draughtsman, using some form of flexi curve to represent the hose string, having little, if any, knowledge of hose or their characteristics. At the best "dry land" mock-ups of submarine systems had been attempted, using springs to represent the hoses and counterbalanced wires to provide 'buoyancy'.

In addition to such work oil companies had, for a number of years, been using sophisticated model testing basins for the evaluation of oil tanker design. Latterly these had also been used to study the SBM itself, although mainly restricted to the buoy, tanker, mooring and anchoring systems. A certain amount of work had simultaneously been conducted into the design of the submarine hose system. However because of the rather large modelling scales adopted in order to accommodate the tanker/buoy, (typically 1/53), in shallow wave and current basins the hose models, at best, were extremely suspect. (A typical comparison would be that of a piece of rather flimsy laboratory rubber tubing). As such model tests, in this respect, were held in dubious regard, and most designs still originated from a drawing board.

Aware of the shortcomings therefore, of previous attempts at model testing, the writer put forward the following proposals:-

1) Dunlop should consider installing model testing facilities to concentrate on the model testing of the submarine hose system, neglecting the other aspects of the SPM operation, ie: buoy, mooring system etc. (Direct consideration of the latter would involve extremely large and complex facilities, outside the scope of a hose manufacturer).

2) All necessary test input in respect of the hose configuration, such as buoy motion etc., could either be gained from external sources, (customers model tests, calculations etc), or calculated within the Division - refer to VII (d). It was recognised that this would involve various approximations but these were considered acceptable when compared against the alternatives.

3) Because of 1) and 2) the buoy may be replaced by a mechanical simulator captable of simulating the buoy motions required.

4) By virtue of the above, wind/wave generation would not be required, therefore a relatively deep low surface area basin or tank could be utilised. (ie: Wind/Wave basins require large surface area basins in order to maintain stable test conditions, including "beaches" to overcome wave reflection etc. By virtue of such size the

Vertical dimension has to be minimised in order to reduce power requirements in the wave generation equipment, and capital cost).

5) Using a tank as put forward by 4), generally depth/relative size is no longer of such importance and thus the modelling scale may be reduced, (ie: size of models increased). It was postulated that the maximum scale possible to ensure reasonably representative model hoses was 1:20.

6) Current, nor the effects of current, could be readily simulated in such a tank. However this would not be a serious problem as most SPM sites do not experience significant current flows and even that encountered tended to approximate to surface layer flow, therefore not of great importance to the submarine hose system.

The above proposals/comments were considered by the Divisional Management, who finally agreed that some form of model testing facility should be commissioned.

At this stage however it was very difficult for either Divisional Management or indeed the writer to accurately assess the benefits of such a facility. (Apart, that is, that it would be considerably better than most previous attempts at submarine system design). Bearing these reservations in mind it was finally decided to install a model test tank of limited capability. Thereby funding the expenditure directly from the Division, rather than seek capital sanction expenditure from the parent Company, (Dunlop Holdings Limited).

The initial model testing unit was therefore an advancement of the drawing board. A model could be examined, in three dimensions, underwater in a static condition or a very simple dynamic state, (vertical sinusoidal motion of a buoy simulator).

This test arrangement is described in section (b).

The above model testing unit was installed during 1974, and has been in use continually from that date. As a result of the work conducted, it became obvious that:-

a) Considerable benefits could/were being derived from such a facility.

b) Confidence in the accuracy of the results was high, (field feedback etc).

c) Improvements could/should be made in order to improve the total modelling capability.

d) A number of proposed studies could not be accommodated by the tank because of size limitations (particularly N. Sea installations). That is without predjudicing the current modelling scale.

e) More work was available than could be accommodated by the existing unit.

On the basis of these rather conclusive arguments a decision was made during early 1976 to apply to the parent Company for capital sanction in order to install a large revised testing unit. Approval was gained, (refer to section (j)) and the new facility installed during late 1976.

This also is described in section (b).

(NB: The 'old' test unit was resited during the installation of the new unit together with the dynamic test rig, see figure (36)).

b) DESCRIPTION OF FACILITIES

The model testing unit consists of two model test tanks, housed in a purpose designed building, (Figure 36) monitored from a central control room, together with associated model preparation and changing areas.

SMALL TANK

The smaller tank of the two designed to accommodate model systems at a scale of 1/20. From (Figure 37), it may be seen that the tank is constructed from modular panels, totally externally braced and internally welded. Tank dimensions are respectively 12ft long x 8ft wide x 8ft deep, capacity 4800 gallons.

Above the tank is mounted a hydraulically activated buoy motion simulator, restricted to one degree of freedom - described in Section (c).

The tank is fitted with portholes to enable observation and or filming of the model system tests, and is wired for video and strain gauge recording.

Tank capabilities in terms of water depth, etc at a model scale of 1/20 are given in Table (9).

Generally the small tank is used for simple tests (ie: restricted dynamic input) for shallow water systems.

LARGE TANK

Designed to accommodate model systems at a scale of 1/20, 1/15 and 1/10. (Figure 38) shows that the tank is of similar construction to that of the smaller tank, but in addition has a high level gantry surrounding the upper panels. (NB: The gantry forms an integral part of the external tank bracing (locking ring) required because of the volume of water contained. It may also be noted that the portholes are smaller than the previous tank because of the stressing of the necessary reinforced glass). Tank dimensions are 20ft long x 12ft wide x 16ft deep; Capacity 24000 gallons.

Once again a hydraulic buoy motion simulator is mounted above the tank, although of a more sophisticated design than the former unit, three degrees of freedom are available - described in Section (c).

Again the tank is wired for video and strain gauge recording.

Tank capabilities, at each of the modelling scales are given in Table (10).

The large tank is used for all tests on deep water systems, (scale usually 1/20) and for the more sophisticated tests on shallow water systems, (scales of 1/15 and 1/10).

The tank designs are described in more detail in (17).

CONTROL ROOM

The control room, Figure (39), is situated adjacent to the tanks and houses:-

- Motion simulator controls/read out
- Video Camera Controls/Recording System
- Strain gauge data logging system
- Personnel communications system

The test programme controller directs the model tests from the control room from where he is in communication with the camera operators, diver etc.

MODEL PREPARATION AREA

Adjacent to the test area is small laboratory specifically for the preparation and testing of model hoses, load transducers etc as described in latter sections. A changing/shower room/diving equipment store is adjacent to the assembly laboratory (Figure 40).

c) BUOY MOTION SIMULATION

In the case of a relatively small mooring buoy, (as is the case of most SPM systems), the motions of the buoy, when occupied by a tanker are, except the vertical heave motion, mainly dictated by the loads in the bow hawser. A knowledge of such buoy motion enables the minimum and maximum distances between the suspension point of the underwater hose of the buoy and the pipeline end manifold (PLEM) to be determined. This data is indispensable for a proper design of the underwater hose system. The horizontal motions of the buoy are normally largest in extreme operational conditions when the buoy is occupied by a tanker; the vertical motions are largest in survival conditions without a tanker moored to the buoy.

An example of the envelopes of motion of the hose suspension point of an SPM buoy in extreme conditions is shown in Figure (41).

Up to now a proper explanation for all phenomena that occur with regard to the behaviour of the system tanker - single point mooring is not available. However much effort is being put into this subject, and there follows a brief review of typical calculations that may be performed.

These methods of calculation involve on the one hand a mathematical description of the important characteristics of the SPM - ship system, including non linear restoring forces and the inertia and damping of the buoy and ship for various modes of motion, and on the other hand a mathematical description of the environmental forces acting on the elements of the system.

Due to the non linearities of the system, equations of motion are integrated on a step by step basis using small increments of time. Generally it may be assumed that the environmental forces are due to:

- WIND
- WAVES
- CURRENT

WIND FORCES

The formulations for the lateral and longitudinal wind forces and yawing moment are of the following type:

$$F_w = \tfrac{1}{2}\rho_a V_w^2 A_w C_{dw}$$

In the formulation the coefficient C_{dw} is determined from model tests in wind tunnels for different heading angle of the vessel and wind.

CURRENT FORCES

The formulation for this type of force is similar to those used for the wind forces:

$$F_c = \tfrac{1}{2}\rho V_c^2 A_c C_{dc}$$

The value of the current force coefficient C_{dc} is determined by oblique towing tests or by tests in current for different angles between vessel and current (1). Both wind and current forces are constant values for a constant speed of wind and current.

WAVE FORCES

The forces due to waves may be split up into two components:

- wave forces and moments which are proportional to the wave height (1st order wave forces)
- wave forces and moments which are proportional to the square of the wave height (2nd order wave forces or wave drifting forces).

The 1st and 2nd order wave forces in regular waves (the only system presently considered*), may be calculated from the pressure distribution on the hull using Bernouillis equation:

$$p = -\rho gz - \rho\frac{d\phi}{dt} - \tfrac{1}{2}\rho\,|V|\,V$$

where ϕ is the first order velocity potential. The first order wave force is found by integrating the pressures acting on the hull whereby the 2nd order contributions of the pressure components are neglected. The mean wave drifting force is found by integrating all second order pressure contributions. In determining the 1st order wave force only the components $-\rho gz$ and $-\rho\frac{d\phi}{dt}$ are taken into account. All three components of the pressure contribute to the mean drifting force.

*(Currently only regular wave patterns are considered, mainly to avoid the added complexity of calculation required for irregular waves. This certainly does involve a certain degree of non similarity between 'real life' and the model, however it is considered that the approximation is still sufficiently valid to yield useful results).

On the basis of the aforegoing, a knowledge of the local environmental condition pertinent to the SPM site, and certain assumptions (or facts) in relation to the probable type of vessel to be used, approximations as to the ship mooring forces may be derived. Typical output from such calculations are shown by Figure (42).

It then remains to translate such forces into equivalent horizontal motion of the buoy and to calculate the heave response of the buoy to the required wave pattern.

In order to calculate both, it is necessary to consider the chain anchor system of the buoy. The principle of such a system is that of a multi leg catenary, see Figure (43), and is based upon a relationship between unit chain weight and buoy excursion. More specifically, when a mooring force is applied, the buoy will move sideways thus lifting a certain amount of chain off the sea bed.

When the weight of lifted chain is sufficient to balance the mooring force a new equilibrium position is formed and no further excursion occurs until the load is changed. In order to prevent the chain from ever coming into a fully tensioned position, causing shock loads, the chain length is such that a minimum section of it remains lying on the sea bed under maximum load conditions. (Often dictated by the breaking strain of the mooring hawser).

In order to conduct the required calculations a mathematical description of the catenary anchor system is thus vital. A method of generating such characteristics is dealt with by Chapter VII (d).

Calculation of the vertical buoy motion is conducted by calculating the wave pressure distribution on its hull and using this as the input force into the anchorage system. Generally however, because of the relative size of waves in comparison to water depth etc a simplification to a double spring mass system may be considered, Figure (44).

By a superposition of both sets of calculations it is therefore possible to derive approximations as to form buoy motion envelopes, as shown by Figure (41). How then is this envelope reproduced mechanically in the model tank?

SMALL TANK

As the originally installed tank, the motion simulator used is of a very simple design, Figure (45). From the photograph it may be seen that the simulator consists of a single hydraulic actuator mounted on a fixed carriage assembly above the tank. The static vertical position of the actuator may be raised or lowered or a screw and pinion device to accommodate changes in water level (tide)/Water depth. Since the assembly is fixed relative to the surface of the tank, effective buoy excursion is produced by relative movement of the model PLEM with respect to the simulator.

It will be noted that whilst the simulator may produce dynamic heave motion (vertical) of the buoy only static excursion motion (lateral) may be attained. The latter is a severe limitation of the facility (dictated by cost considerations) and requires that the motion envelope, (Figure 41) be split into a series of vertical strips, studied independantly and later superimposed. The latter however was found to not always be necessary as many cases of excursion have been determined whereby the displacement could nominally be considered as static (wind/current dominated forces) and only the buoy heave motion considered superimposed upon the said static excursion.

Heave motion, produced by oscillation of the hydraulic actuator, may be controlled in terms of amplitude and frequency, and is nominally sinusoidal in form. However facilities exist, and have been used, to inject the signal input from an external source, ie: signal generator, magnetic tape etc.

The maximum capability of the simulator is naturally determined by its amplitude and response characteristics, also by the model scale employed. However, considering a scale of 1:20, the following apply:-

Max heave motion	: $\pm$ 15ft
Max excursion	: 50ft
Max Heave frequency	: 1/5 Hz at full stroke.

LARGE TANK

Based upon the experience gained from the small tank simulator, the same designed for the large tank was of a more sophisticated design, Figure (46). As may be seen the simulator is mounted upon a mobile carriage system above the tank which enables X, Y and Z motion of the simulator mounting with respect to the tank. The simulator itself, although restricted to motion in a single plane, is allowed three degrees of freedom. Thus heave, excursion (or sway) and tilt of the buoy may be reproduced.

It may be noted that the simulator consists of two hydraulic actuator units which are attached at a variable position pivot on the vertical ram.

Static displacement of the carriage assembly coupled with simulator dynamics may be superimposed to produce most buoy motion patterns, naturally limitations do exist. Although dynamics may only be produced in the one plane, it is possible to rotate the model hose system with respect to this plane in order to study the effect of directional phenomena.

Control of the simulator is produced in a similar manner to that described previously for the smaller unit. This time however the control function is doubled (ie frequency and amplitude of two actuators), and is coupled with phase control between the two systems plus mechanical adjustment of the simulator geometry. Once again sinusoidal input is nominally used although similar "random" input facilities are available.

Specifying the capabilities of the simulator is a similar manner as previously used, (again at a scale of 1:20).

Static Displacement	Z	=	$\pm$ 60ft
Static Displacement	Y	=	$\pm$ 40ft
Static Displacement	X	=	$\pm$140ft
Max Heave Amplitude	(Z)	=	$\pm$ 30ft
Max Heave Frequency		=	1/5 Hz at full stroke.
Max Sway Amplitude	(Y)	=	$\pm$ 40ft
Max Sway Frequency		=	1/5 Hz at full stroke

(d) MODELLING CRITERIA

The choice of the model scale depends on a number of factors, the most important being:-

WATER DEPTH: Each tank has a maximum water depth

ACCURACY OF RESULTS: The larger the scale factor, the smaller the models, the lower the forces in the model and the accuracy with which such forces/characteristics may be adjusted or measured.

CAPABILITY: Of generating the required buoy motion at a particular scale, (magnitude and frequency considerations).

Taking the above into account a scale factor $\propto$ for the model tests must be chosen. This means that if the linear scale factor is $\propto$, all full scale linear dimensions will be reduced accordingly. The scale factors for other properties, weight, inertia etc depends upon the principle of similitude adopted. This in turn, is dependent on the phenomena which determine the dynamic behaviour of the prototype and the model. In reality, the dynamic behaviour of bodies located in or near the water surface, (ie the buoy), are dominated by forces due to the action of waves and forces due to the inertia of the body. If the body is far removed from the water surface the motions are dominated by friction forces and inertia. (In the case of a submarine hose system it is reasonably assumed that the majority of motion is buoy generated). The law of similitude between prototype and model for the case that the behaviour is dominated by the action and the inertia of the body has been formulated by William Froude and hence is known as Froude's law. This states that for dynamic similitude the following condition must be satisfied.

$$\frac{V_m}{\sqrt{g_m L_m}} = \frac{V_p}{\sqrt{g_p L_p}}$$

where: V_m, V_p = a velocity in the model or prototype

g_m, g_p = gravitational constant in the model or prototype

L_m, L_p = a characteristic length in the model or prototype

Assuming that $g_m = g_p$

and that $\frac{L_p}{L_m} = \alpha$ = linear scale factor

it follows that: $\frac{V_p}{V_m} = \sqrt{\alpha}$ = scale factor of speed

and $$\frac{\frac{L_m}{V_m}}{\frac{L_p}{V_p}} = \frac{t_p}{t_m} = \sqrt{\alpha} = \text{scale factor of time}$$

The scale factor for weight and force becomes:

$$\frac{F_p}{F_m} = \lambda\alpha^3$$

where: λ = ratio between specific gravity of sea water at the SPM location and the fresh water in the model tank.

The scale factor for frequency becomes:-

$$\frac{W_p}{W_m} = \frac{1}{\sqrt{\alpha}}$$

where: W = frequency in radians/sec or cycles/sec etc.

(e) MODEL CHARACTERISTICS

To yield data from which realistic facts can be interpreted it is absolutely essential for every model component to be correctly scaled.

Models of ancillary equipment (Figure 47) eg: buoyancy tanks, submarine floats, spreader bars, swivel unions etc., are constructed from a variety of materials; wood, metal, synthetic foam, plastics etc. In practically all cases, components are made as rigid as possible since the tests are aimed at the determination of rigid body behaviour and not at the determination of elastic behaviour of construction elements. Weight and/or buoyancy of the models are determined and adjusted; centre of gravity is determined and adjusted by means of inclining tests in air while the mass distributions are checked by means of pendulum tests.

Model hoses are similarly prepared, however additional features relating to bending, axial and torsional stiffnesses are also considered. It is worth, therefore, to consider in greater detail the preparation of each length of model hose.

PREPARING THE HOSE MODEL

These hoses in reality consist of strings of large bore flexibles, each string of hoses consisting of elements with lengths of nominally 30 - 40ft bolted together. The bore of these hoses may vary between 12 and 24 inches in reality. A hose element consists of a flexible middle section made of a fabric/wire/rubber composite matrix with semi rigid/rigid end sections terminating in a flange.

Models of such hoses are made in a manner similar to reality and consist of a steel wire, variable pitch helix embedded in a low modulus natural rubber compound, (open weave fabric mesh is also used in the smaller scale, larger size models). The extremities of each model element ending in a rigid part with a 'connector' to which the next section may be attached, Figure (48).

Originally by a basic approach of experimentation, (eg: size/pitch of helix, rubber modulus, hardness, s.g. etc), followed by carefully controlled batch production the following properties were scaled:

- LENGTH/DIAMETER
- UNDERWATER WEIGHT
- BENDING STIFFNESS
- AXIAL STIFFNESS
- TORSIONAL STIFFNESS (1/15 and 1/10 scale models only - not 1/20)

At this stage it would be useful to review the tests conducted to ensure the uniformity of the above characteristics.

The first test conducted is a check on the models underwater weight, (either "full" of oil of specified gravity or, water), both series of models are made; (NB: water is always maintained within the hose, the effective oil filled condition being simulated by reducing the weight of the model carcase). This is performed by immersing the model in water, Figure (49). Minor adjustments are then made by the addition of small ballasting weights.

The second test ensures that each of the lengths of hose has the required bending characteristics. Such characteristics are complex and dependant upon such factors as internal pressure and radius of curvature. The test is therefore a repetition of a series of tests performed on the equivalent full size hose, the model conforming to a fixed configuration for a predetermined load, Figure (50). Series of models are often made corresponding to differing internal pressures.

Usually a minimum of two are used in the test programme, zero pressure - buoy unoccupied and working pressure 150 - 225 lbf/in^2 - tanker loading.

The third test (only conducted for 1/15 and 1/10 scale models*) is for torsional characteristics. The model is mounted in a torsion rig and its angular twist measured for various torque loadings (Figure (51)).

*(It was found impossible in the 1/20 scale model series to simultaneously reproduce all of the full size characteristics. As the most difficult to model, and considered to be of least importance to the test, the torsional characteristics were not reproduced. The models infact had a much lower torsional stiffness that the equivalent in reality, approximately -50% -75% depending upon bore size).

The final test conducted is for axial stiffness, (extension under load). This characteristic is determined by mounting the model in an extensometer rig and plotting its load - deflection curve (Figure (52)). Minor adjustments are then made to the model by adjusting the tension in an internal synthetic cord running between opposite end couplings.

It may readily be seen that a consideraole amount of work is conducted on the models prior to introduction into the tank to ensure accuracy and authenticity of the test results.

(f) TEST RECORDING/LOAD MEASUREMENT

VIDEO RECORDING

During the course of the test series the motion and resultant configuration of the submarine system are recorded using still and cine photographic techniques, (the latter from underwater if required), and also by video equipment.

In the case of the latter three video cameras are used, two external to the tank viewing through portholes and the third an underwater unit operated by a diver. Each camera has a calibrated grid lens system in order that deflections, radii of curvature etc may later be analysed from the recording.

During the course of the tests, each camera is simultaneously displayed on its own mounter housed in the control console, (Figure (53)). Any picture may then be selected by the test controller to be transferred onto video tape. The selected picture is simultaneously displayed on a fourth monitor. In addition a "caption" camera may be over-written onto the video tape recorder in order to identify each video sequence to the test conditions.

At all times during the course of the test the test controller is in audio contact with the camera operators and diver such that cameras can be panned, refocussed etc and adjustments made to the model (Figure (54)).

During the final sequence of a test programme the recorded video tape is edited onto a second recorder, additional captions inserted and an audio commentary added. (NB: At this stage the model scale motion (frequency) is reduced to the equivalent motion in reality to avoid later confusion when reviewing the tapes.

LOAD MEASUREMENT

Axial forces, bending moments (in mutually perpendicular planes) and torsion (1/15 and 1/10 scale series only), are measured in the underbuoy hoses. These are measured by means of specially designed/ constructed strain gauge transducers that may be fitted at points where hose elements are "flanged" together or connected to ancillary equipment. The length of the transducer is such that they correspond to the length of the rigid hose fittings, in this way no additional discontinuities in the curvature of the hoses are retained.

Before continuing further it would be worthwhile to briefly outline the design/construction of the transducer. In designing the transducer the following criteria had to be met:

a) Ability to work underwater

b) Sufficiently sensitive to measure relatively small loads whilst retaining sufficient rigidity in the transducer body to accommodate survival load peaks.

c) Give nominally a linear response over the output range required.

d) To be constructed from a material such that additional loads (ie: transducer weight) were not introduced into the model system.

e) Ease of introduction into the model underwater by a diver.

The above criteria were eventually resolved via two basic designs employing copolymer (Delrin) as the shell material. Delrin was chosen because of its nominal neutral buoyancy in water, modulus characteristics and machinability.

The first design, thin walled tube concept, was adopted for the 1/20 scale series models and was capable of measuring the stated loads with the exception of torsion, (Figure (55)). As may be seen from the photograph one end of the transducers could be located into a model hose coupling via a screw thread whilst the other is "snapped" into its appropriate counterpart.

The second design, double perpendicular barbell concept, was a modified design for the 1/15 and 1/10 scale series of models and was capable of measuring torsion in addition to the previous loads, (Figure (56)). The method of connection into the model may be seen as replaceable plastic "snap-in" rivots.

Both transducers employ linear foil strain gauges, (45° rosettes used for torsion measurement), bonded to the delrin surface by a specially developed adhesive system. The gauges, terminations etc are finally baked to remove all traces of moisture and encapsulated in a low modulus silicone rubber compound to effect water-proofing. Waterproof strain gauge cables are used to connect the transducer to the surface connection box, joints at the transducer once again being protected/sealed by the silicone encapsulant.

(NB: During initial work with the transducer(s) it was found that considerable temperature gradients were built-up within the tank(s) due to the effect of the bottom heating coil and overhead arc lights. Thermally generated strains were recorded as the transducer(s) were moved within the tank, the latter naturally occurring during motion of the hose system under test. Because of the restrictive size/design of the transducers it was not possible to resolve this effect via compensating networks for all of the measuring grids. Temperature self compensating gauges were therefore employed with a temperature coefficient of $\alpha = 65 \times 10^{-6}$, corresponding to the thermal expansion coefficient of delrin.

Five such transducers may be used in conjunction with the current measuring system (ie: up to 20 channels of data). The system employed is a FM system, which simultaneously heats (energises) all of the data channels and continuously samples one channel from each transducer for display (Figure (57)) and Appendix (6).

(NB: Both the video and strain logging systems can be used in either of the test tanks, ie: a test is run in one tank whilst the other is prepared for the next test series etc).

Before introduction into the tank each transducer is balanced (in conjunction with its associated balance network of the measuring system), and calibrated on a purpose designed calibration rig. (Figure (58)).

LOAD RECORDING

Static signals are read directly from a digital display/store. Simple dynamic signals (slow moving) are recorded in the same manner. Generally dynamic signals are recorded on a multi-channel ultra violet recorder or magnetic tape recorder, depending on the number and/or complexity of signals to be measured.

Signals recorded on UV paper strip are used to read off maximum and minimum values and for checking purposes. Signals recorded on magnetic tape are digitised and used for more complex analysis. (Often however similar information may often be extracted from the UV strip using graphical techniques).

From the signals recorded during tests, the normal statistical quantities analysed are:

- MEAN VALUE
- ROOT MEAN SQUARE VALUE
- MAXIMUM VALUE
- MINIMUM VALUE
- SIGNIFICANT VALUE : Mean value of the one-third highest peak to trough values.

If required distribution characteristics can also be calculated. As can be seen from the above, one signal measured during one test can already yield a large amount of data. Although all these quantities describe the behaviour of the signal, the amount of data is too weildy when it comes to using such results for design purposes. In many cases the designer only requires a reliable value for the extreme of a particular signal.

(g) INPUT DATA/TEST PROCEDURES

INPUT DATA

Several aspects must be considered for the test programme:

- Behaviour and system loads in extreme environmental conditions when the terminal is unoccupied. During these tests the so-called survival conditions are investigated. The system loads for such a sea state combined with the probability of occurrence of the sea state form the basis for determining the probability of survival or system damage.
- Behaviour and system loads with the terminal occupied by a tanker. During these tests the upper limits (with respect to the environmental conditions) for operating are investigated. The results of such tests combined with the probability of occurrence of limiting sea conditions form the basis for the evaluation of workability.
- Behaviour and system loads under environmental conditions which occur a large percentage of the time. These results, combined with fatigue data (if known) can form a basis for maintenance scheduling.

In order to set up a model test programme, the following environmental information should be known:

1) Water depth and tidal variation.

2) Current : Speed, direction, variations in speed and direction.

3) Waves: Probability of occurrence of a given significant wave height and period. Dependence of significant wave height on mean period. Dependence of significant wave height on the direction of the waves.

4) Wind : Speeds and directional relationship with above quantities.

Generally published wave, wind and current data are used or a consulting body used to obtain the same, for example wave height/period probability data, see Figure (59).

From the environmental data a number of sea conditions with respect to wind, wave and current are selected under which the tests will be carried out. At this stage experience gained from previous tests or from experience gained with existing, similar installations is effective in reducing the number of different sea conditions possible to a level which is acceptable from the point of view of limiting the number of tests. Whilst still remaining sufficient to gain results representative of the particular system.

TEST PROGRAMME

When the environmental conditions and the configuration of the terminal are known, the test programme may be set up. Generally a limited selection of environmental conditions to be tested is made from the wide range of conditions which may occur. The selection is usually as follows:-

SURVIVAL TESTS - conditions are chosen which have a probability of occurrence of once in 50 or 100 years. These conditions are usually with high winds, waves and current from the same direction.

EXTREME OPERATING - conditions are chosen with a probability of occurrence of 1 to 10%. Often these conditions will be selected which, from experience, are known to give high forces due for instance to the relative directions of wind, wave and current.

NORMAL OPERATION - moderate conditions with the buoy occupied or unoccupied, conditions are chosen to have a probability of occurrence in the region of 50%.

The number of different sea states is generally as follows:

Survival tests with the buoy unoccupied

1 to 3

Operational tests with a tanker moored to the buoy

3 to 6

Tests in moderate conditions

1.

The sequence of tests is often as follows:

One test in survival conditions and two or three under operating conditions.

From the results of these tests it may be concluded that some characteristics of the system need to be altered, for instance: manifold angles, hose string length, buoyancy distribution etc. The first tests would then be followed by two or three tests in which the effect of the changes are checked. This part of the programme consists of, as it were, quick look tests and preliminary optimization tests. After the first part, a series of tests are carried out under operating conditions to investigate the influence of various input combinations. These tests form the bulk of the programme and may number from 10 to 30 tests. Generally not all possible combinations of parameter variations are tested, however. After this part of the programme, a number of tests are usually devoted to the optimization of some specific aspect of the system.

VI.8 TYPICAL TEST RESULTS

Rather than to illustrate the results of one particular test in depth it is intended in this section to discuss various aspects of system design, their effects as illustrated by model testing and basic guidelines for their application.

(a) INPUT DATA

Whilst it might appear almost naive to stress the importance of obtaining accurate and reliable input data it is a point that unfortunately is frequently overlooked. Although most companies institute a survey, prior to the selection of a terminal siting, these generally are of little value to a system designer. Often the only useful information relates to water depth, tidal range and bottom topography. Particularly with respect to wave and current conditions information is usually very scarce or non existent. Faced with such a situation, designers often over specify in order to 'safe-guard' their decision. However this procedure often tends to 'back-fire' in relation to submarine hose system design. For example the survival and maximum operating for a buoy are often selected on rather an arbitrary statistical basis. Frequently the survival condition is chosen to be the 100 year storm condition, and maximum operating some fraction of the above figure.

The results of such arbitrary decisions may be best illustrated by the following example:-

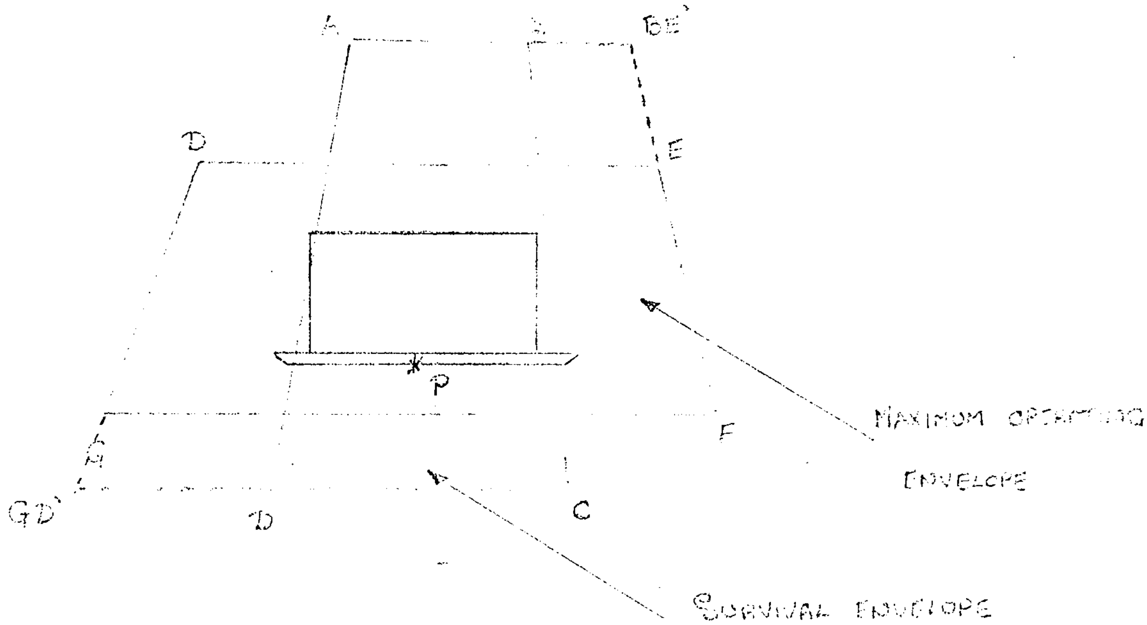
SITE	:	Location Uruguay
WATER DEPTH	:	(Mean) - 19m
TIDAL RANGE	:	1.2m
SURVIVAL WAVE	:	8.2m
MAXIMUM OPERATING WAVE	:	2.2m

NB: Maximum operating is defined as that condition in which a tanker could remain moored to the buoy.

It may readily be seen that in the worst case the wave height represents in excess of 40% of the water depth. On querying the above information it was determined that records over the most recent eighteen month period indicated maximum wave heights of $\leq$ 3m, a significant difference. However as the above figures were already written into all of the relating specifications no person was prepared to replace such by realistic values. (The oil industry is well known for its conservatism and resistance to change!).

On the basis of the above therefore a totally unique system required development, which was rather an unsatisfactory solution to the problem. As may be seen from Figures (130) and (131) the eventual solution incorporated the use of swivels (mounted beneath the buoy), a totally new and untried concept. (Various attempts have been made to incorporate swivels at the floating buoy manifold with very little success, generally the swivel bearing 'breaking-up' extremely rapidly). Figure (131) clearly indicates however that such were necessary in order to accommodate the wide variation in water depth required.

Also of extreme importance is a true and accurate determination of buoy excursion under the various conditions to be studied. Generally this situation may be represented in very simplistic form by the following diagram; (refer also to Chapter VII.2):-



EXCURSION ENVELOPES OF POINT P

Generally two excursion envelopes are of interest. The survival envelope where no tanker mooring load is present, but is subject to maximum wave forces; and the maximum operating envelope where wave forces are reduced but replaced by large directional mooring loads. These respectively tend to be of the shapes indicated.

Because of lack of understanding (or more correctly the apparent inability to calculate buoy excursion), it was frequent practice within the industry to quote a single excursion value as being representative for all conditions. It may readily be seen that such a practice, ie: extending the above envelopes to points BE' and GD' significantly increases the required performance of the hose system; often to a stage whereby the whole concept becomes extremely suspect.

Unlike most other engineering systems, designing for extreme values is detrimental in relation to submarine hose design. Frequently there exists a conflict between the system which gives optimised performance for a large range of operating conditions, and that which proves necessary in order to accommodate the large extremes of low statistical probability. Frequently the latter case prevails, (as the specified design criteria) and the resulting system although capable of accommodating design conditions, (encountered very rarely ie: every 10 years etc., if at all); results in a badly engineered system which gives rise to higher failure rates under normal conditions than should have been the case.

In relation to hose system design a compromise question should be asked; "Is it more advisable to design a system for optimised performance for some 90%-95% of conditions encountered and accept failure under the remaining 5% of situations, or does one design for the extreme and accept a general higher failure incidence for lesser conditions". Frequently the probabilities are such that former is by far the most economical sound decision. However due to a lack of appreciation by the vast majority of personnel specifying such system the latter generally prevails. In both cases however the need for reality and accuracy in specifying input data is paramount.

(b) WHICH SYSTEM?

Generally two systems predominate submarine system design:-

LAZY 'S' - FIGURE (132)

CHINESE LANTERN - FIGURE (133)

As may be seen from the Figures the terminology is self-explanatory. There are however variants on these, particularly in the former case where either buoyancy tanks or individual floats might be employed as the buoyancy system.

The prime conceptual difference between the two systems is the way in which they derive their respective configurations. In the case of the Lazy 'S', this is governed predominantly by a balance between gravitational and buoyancy forces, ie: one is trying to achieve neutral buoyancy. Structural stiffness plays a relatively minor role in the resultant configuration perhaps apart from those areas immediately adjacent to buoy and Plem. However the converse is true for the Chinese Lantern system. Here the configuration is governed predominantly by system stiffness and lesser affected by both weight or buoyancy. These initial criteria lead one to the first statement concerning the appropriate use of each system. In order to make the best use of, and to reinforce these characteristics, the Lazy 'S' is better suited to deeper water installations, ($\geqslant$ 100ft) whilst the Chinese Lantern performs best in shallow water (50-100ft) systems. Having identified one major characteristic of these systems, hence a guide to choosing that most appropriate the following is an attempt to further classify each system in terms of its advantages/ disadvantages:-

	LAZY 'S'	CHINESE LANTERN
a) WATER DEPTH	Best suited to depths of 100ft, may be used for shallower sites although problems occur in relation to system stiffness and accommodating the necessary hose length in the area available.	Becomes significant at depths between 50-100ft when used at greater depths, the prime characteristics of the system stiffness becomes a lesser influence, requiring the increasing balance of body forces by adding buoyancy.
b) ABILITY TO ACCOMMODATE LATERAL BUOY EXCURSION	Generally much greater scope than equivalent Chinese Lantern. Degree increases with water depths, as generally does the requirement.	Relatively restricted by conceptual design however for shallow water systems excursion are generally not great.

	LAZY 'S'	CHINESE LANTERN
c) EFFECT OF INTERNAL PRESSURISATION	Little effect, the configuration only marginally altering.	Because of the inter-relationship between pressure and stiffness, pressurisation can have a significant effect on configuration.
d) SPECIFIC GRAVITY OF CONTENTS - (PARTICULARLY IMPORTANT IS SYSTEM REQUIRED TO BE WATER FLUSHED DUE TO ENVIRONMENTAL REASONS).	Can have a significant effect particularly for a float supported system. In the float supported case, the removal/loss of a float can also induce significant configuration change (See Figure 134). This is because an attempt is made to achieve a neutrality of forces on a "step by step" basis throughout the system. In the case of a tank(s) supported system a similar "overall" situation	Little effect, except where used in deeper water installations.

	LAZY 'S'	CHINESE LANTERN
	is achieved, but by concentrating buoyancy in a single location. This latter case is less susceptible to such changes as incurred by individual float loss.	
e) USE OF MULTIPLY HOSE STRING	Not best suited for multiply string operation. Generally because of the cost of additional Plems (required if the strings were to be independent), such systems require the hoses to run in parallel. The latter requires the use of movement restricters or spreader bars to restrict lateral differential hose movement which could cause local loading/abrasion.	Ideal for such use, three, four or even more strings may be geometrically displaced about the buoy centre line. Each is a system in its own right, and does not affect the others during installation or operation.

	LAZY 'S'	CHINESE LANTERN
	Generally the above do not work well, and in the case of a float supported system float abrasion/loss can be a problem (refer to d). Generally only two or at maximum three strings may be installed as such. Even then they interact during both installation and operation.	
f) STABILITY (UNDER WAVE INDUCED FORCES)	Generally the Lazy 'S' is a very stable system. However in a certain situation a phenomenon termed "cork-screwing" occurs, refer to Figure (135). "As the buoy moves towards the Plem the curvature of those hoses beneath the buoy increase. At a	Under some circumstances - particular for less stiff systems. Vertical buoy motion induces a three dimensional configuration to occur, (refer to Figure 136). This tends to induce large torsional forces at both the buoy and the Plem, which is detrimental to system performance and hose life.

LAZY 'S'	CHINESE LANTERN
critical curvature the system distorts from a nominally planar configuration into a three dimensional mode which when viewed appears as a cork-screw. Whilst appearing to be rather alarming the above has found to be only a transient instability and in actual fact is beneficial in reducing system loads - particularly bending moments.	

Whilst an attempt has been made to develop a critique of system(s) performance based upon the results of model testing it must be stressed that each installation is inevitably a compromise. This in many cases has resulted in the need to model test alternate systems to enable a responsible selection decision to be made. The phrase "Hoses for Courses" is very appropriate in this sense.

(c) INITIAL SYSTEM OPTIMISATION

One of the greatest benefits derived from the model testing and facilities described, is that of the initial system optimisation. These tests generally being of the form of static and a few dynamic tests in order to assess the ability of the system to accommodate design extremes coupled with the normally prevailing conditions. During these initial tests many variants in the form of:-

: SYSTEM TYPE

: HOSE LENGTH (TOTAL + INDIVIDUAL ELEMENTS)

: MANIFOLD ANGLES

: BUOYANCY DISTRIBUTION ETC

may be rapidly evaluated. Using both visual and load monitoring techniques the effect of change can rapidly be measured and categorised.

It is during this stage that most of the major design decisions are made. (During model testing it has been found that differences in hose length of, eg: 5ft in 100ft, can be significant when reflected in measured system loads). Generally during latter testing, the bulk of the dynamic tests, relatively little additional system modification of any significance results. In other words a law of diminishing returns operates. These latter tests however to generate considerable information regarding system loads with respect to operation conditions which are extremely useful both for test rig input data and later should enable typical system life prognosis to be made.

(d) GENERAL CONCLUSIONS

- The hose loads are in general greatest at the buoy end of the hose string.

- The hose loads at the buoy are more sensitive to changes in the sea conditions than to changes in the hose configuration.

- The hose loads at the Plem are to a large extent dependent on the hose configuration.

- The influence of the presence of the tanker in the loading of the hose(s) near the buoy is relatively small. However the converse is true for the Plem hoses.

- Visual observation is indispensable for the judging of the general performance of a submarine hose system.

Because of the diversity of systems tested, and their individually tailored design only a brief resumé of generalised results has been made. For greater details of individual tests refer to the appropriate model test report, reference (13). (NB: The release of some of these reports will be subject to approval from the oil company involved).

(i) ACCURACY OF RESULTS

For a given set of model and environmental conditions, errors, uncertainties and inaccuracies in the results of the tests come from the following sources.

- SCALE EFFECTS
- ERRORS IN MEASURING DEVICES/SYSTEM
- USE OF REGULAR WAVES
- BUOY MOTION SIMULATION
- INABILITY TO MODEL CURRENT

SCALE EFFECTS

Tests are carried out according to Froude's law of similitude. This means that scale effects will occur in phenomena many dependent on friction effects since these are dependent on equality of the Reynolds number.

The greatest influence of frictional effects during the conduction of tests is that of flow around the hoses and/or buoyancy units etc. Generally buoyancy chambers approximate to flat submerged buoy type structures, the current flow around such bodies being three dimensional. That part of the flow encountering sharp corners will result in flow separation in the model as well as in reality. This means that generally the flow around such bodies match quite well with reality, and consequently, scale effects due to friction will be small. They are estimated to be in the region of 0-10%.

However, the hoses are of a slender cylinder form, the flow around such elements will be of a two dimensional form. In principle the model with these then shows different points of flow separation to the full scale system. In this case we must look at the local Reynolds number re model and prototype and determine the difference in drag coefficients C_d. For circular cylinders the drag coefficients C_d are shown to a base of log re in Figure (64), taken from (5). Model tests are practically always carried out at sub-critical Reynolds numbers. This means that the C_d values are always approximately equal to 1. Prototype Reynolds numbers are often in the super-critical region. The C_d value in this region is, however dependent upon the roughness of the cylinders as shown in Figure (60). In the case of actual installations, quite high roughness values may be reached due to the marine growth. This tends to increase the C_d values for the prototype thus bringing it closer to the model values. In general, model values of such "current" forces will be somewhat higher with values being from 0 to 20% above those in reality.

SENSITIVITY OF MEASURING DEVICES

The load transducers are generally capable of measuring forces with an accuracy far in excess of the minimum accuracy required. They are accurate to approximately:-

BENDING MOMENTS	1%
AXIAL FORCE	5%
TORSION	3%

Errors may also be made in reading values off UV traces, sampling and digitising signals recorded on magnetic tape.

These are dependent on the trace width, noise levels, sampling rate etc.

Generally errors may amount to approximately 2 to 5%.

USE OF REGULAR WAVES

In irregular waves the height of the incoming waves is a slowly varying quantity. Since the wave drift force (2nd order wave forces) is proportional to the square of the wave height, this will also show slowly varying or low frequency components, (see figure (65)). As may be seen from this figure, peaks in the wave drifting force are associated with the occurrence of groups of higher waves. In principle the components of the wave drifting force in irregular waves have frequencies from 0 to infinity, however, the main components are concentrated between the frequency 0 and the frequency of the waves from which the force originates.

Moored ships generally constitute mass spring systems with natural periods for horizontal motions in the order of magnitude of 50 - 500 seconds. These frequencies are outside of the wave frequencies and in the range of the frequencies of the drifting forces. Damping in the horizontal motions is generally low. These two characteristics combined with the frequencies of the slowly varying wave drifting force can produce large amplitude low frequency motions in moored vessels. These motions may in some cases dominate the loads imposed on the system as shown in Figure (66).

Under certain conditions, with the ship head onto the waves, it is possible to calculate the mean drifting force in regular waves from strip theory (7).

Based on certain assumptions the mean value and the spectral density of the oscillating part of the low frequency wave drifting forces may be calculated for irregular waves. However in comparison with model tests on moored vessels such calculations do not give an accurate measure of such behaviour.

In reviewing the above it may therefore appear that the absence of this force results in the possible inclusion of considerable inaccuracy.

However, with reference to the submarine hose system, this is not strictly so. The upper limit of the above force which may effectively be applied, via buoy excursion, is restricted by the breaking strain of the mooring hawser. During the extreme operating tests therefore cases of limiting excursion are examined (ie: limit of mooring hawser capability), which cover the exclusion of this phenomena.

BUOY MOTION SIMULATION

Up to now a proper explanation for all phenomena that occur with regard to the behaviour of the system tanker - buoy mooring is not available. Thus the motions calculated are subject to error and finally the motions reproduced are subject to additional error (ie: restriction in degrees of freedom etc).

Estimates of the degree of error involved vary with the type of motion (magnitude and frequency) and the input information available, generally these are considered to be of the order of 10% - 26%.

The above errors may appear rather large, however they must be considered in perspective. Buoy motions measured in model tests may reduce such error - efforts are continually being made, with limited success, to obtain such information.

However, irrespective of how the buoy motion is finally computed/ reproduced the largest single cause of error is probably errors inherent in the environmental input data.

(NB: Data collection in recent years, in the N. Sea has revealed considerable inconsistencies in previously employed data, and continues to expose anomalies within itself. The vast majority of SBM sites are not as well documented!)

CURRENT

The inability of the present facilities to model current is a limiting restriction which is imposed. Generally tests are considered dubious for sites that encounter directional currents of greater than 0.75 knots in magnitude. However the vast majority of current/planned SBM installations come within this restrictive band.

Attempts at reproducing the effects of current, via calculation and the attachment of counterbalance wires, have been made with limited success. Such tests are generally restricted to static studies, (ie: no wave motion).

Alternate methods of inducing current type loads (ie: electro magnetic fields etc) are presently being evaluated.

In conclusion, although the possible sources of error high-lighted by this review might appear to cast considerable doubt on the validity of such model tests. The advantages of, even such limited, model tests beyond that of historical design/evaluation methods is conclusively apparent.

However one must not be tempted to confuse such model tests with reality, they are an indication of reality, (sometimes poor, sometimes good) and should not be treated as definitive criteria.

VI.10 FINANCIAL EFFECTS

By their very nature, the justifications for the capital investment outlined, are very difficult to quantify. It will certainly result in the flow of additional business to the Division, as demonstrated by previous experience, particularly if our competitors do not take a similar course of action. Dunlop's standing, as market leaders, will also be enhanced by our ability to compete not only as hose suppliers, but also as system designers/evaluation engineers. To summarise, the investment would result not only in visual evidence of the technical competence of the Division but also in direct financial benefits based upon market place decisions in response to such facilities.

In general, the effects of the investment may be quantified by both financial gains and savings:-

(1) During 1975/76 a total of three major contracts were gained partly because of our ability to conduct model tests, (total value £1,060,000). With the additional tank primarily engaged upon deep water installations it would not be unreasonable to anticipate a minimum contribution in excess of £140,000 as a direct consequence. (ie: One major N. Sea contract/year). However, as it is difficult to be certain that the obtaining of a contract is directly as a result of model test facilities, the values above have not been used in justification calculations.

(2) On the basis of conducting four major test series in the large tank and ten similar tests, (of less complexity) in the smaller

tank during 1976/77 a total turnover contribution of approximately £100,000, (ie: Test fees), would not be unreasonable. Of this figure £30,000 has been assumed to result directly from the proposals outlined.

(3) In addition to the above positive aspects, a saving, as a result of significant diver redeployed for some 75% of his time, would amount to approximately £2,000 in a full year. As this is only a redeployment, no use has been made of the value of calculations.

(4) The direct beneficial effects of such a facility, in relation to the Division's R & D effort in new and existing products, must also be considered: (eg: Salm hose development). This however, is extremely difficult to measure in financial terms.

FINANCIAL SUMMARY

(1) CAPITAL EXPENDITURE

Capital expenditure as detailed below will be incurred between the end of May and the beginning of December 1976.

	£
Enlarged tank and carriage assembly	13,000
Water supply; pumps, piping etc	1,545
Water filtration/circulation plant	1,813
Water tempering; heat exchanger, pump etc	2,303
Electrical supply, lighting equipment	995
Screening of work areas; heating; lighting	1,500
Hydraulic buoy motion simulator	8,123
Torsional load measurement equipment	4,150
	33,444

(2) A & R EXPENDITURE

	£
Transfer of existing tank	3,150
Transfer of heating/filtration equipment plus shower etc.	1,380
Making good existing units	385
	4,915

(3) TIMING OF EXPENDITURE

On the basis of the lead time of components the capital and A & R would be spent before the end of 1976. (This assumes sanction approval by the end of June).

(4) DEPRECIATION

Depreciation has been calculated as follows:-

ITEMS	VALUE	LIFE	DEPRECIATION
Hydraulic Equipment	£ 8123	4 years	£2031
Electronic Equipment	£ 4150	4 years	£1037
Other items	£21171	10 years	£2117
	£33444		£5185

It has been assumed that only half this charge would be borne in the first sanction year.

(5) RUNNING COST

The additional costs of running the proposed equipment will be as below:-

ITEM	ANNUAL COST	FIRST YEAR COST
Water Usage	£ 200	£ 100
Electricity	£ 500	£ 250
Maintenance	£ 1800	-
Model hoses	£ 6500	£6500
Labour	£ 4500	£2250
	£13500	£9100

(6) INCOME

The sale of services is estimated as giving rise to income of £100,000. For the purpose of the calculations £30,000 has been assumed as arising directly from the new facility as in times of strong competition the ability to sell the service, as opposed to offering it free of charge, may be affected. It has been assumed that only a quarter of the savings would be achieved in the first year.

(7) PAY BACK

The above figures give a pay back in 2.75 years.

(8) DCF

DCF will be in excess of 25%.

YEAR	EXP.	TAX ALL.	NET SAVING BEFORE DEPN.	TAX	CASH FLOW
	(4915)				
1	(33444)	17391	(1600)	-	(22568)
2	-	-	16500	832	17332
3	-	-	16500	(8580)	7920
4	-	-	16500	(8580)	7920
5	-	-	16500	(8580)	7920
6	-	-	16500	(8580)	.... 25%+
7	-	-	16500	(8580)	
8	-	-	16500	(8580)	
9	-	-	16500	(8580)	
10	1000	-	-	(8580)	

TABLE 9

OPERATING CAPACITY SMALL TANK (AT 1/20)	
MAX DEPTH	140 ft
MAX BUOY HEAVE	± 15 ft
MAX EXCURSION	50 ft
MIN HEAVE PERIOD	5 sec

TABLE 10

OPERATING CAPACITY LARGE TANK (AT 1/20)	
MAX DEPTH	300 ft
STATIC DISP Z	± 60 ft
STATIC DISP Y	± 40 ft
STATIC DISP X	± 140 ft
MAX HEAVE	± 30 ft
MAX HEAVE FREQ	± 1/5 HZ
MAX SWAY	± 40 ft
MAX SWAY FREQ	1/5 HZ

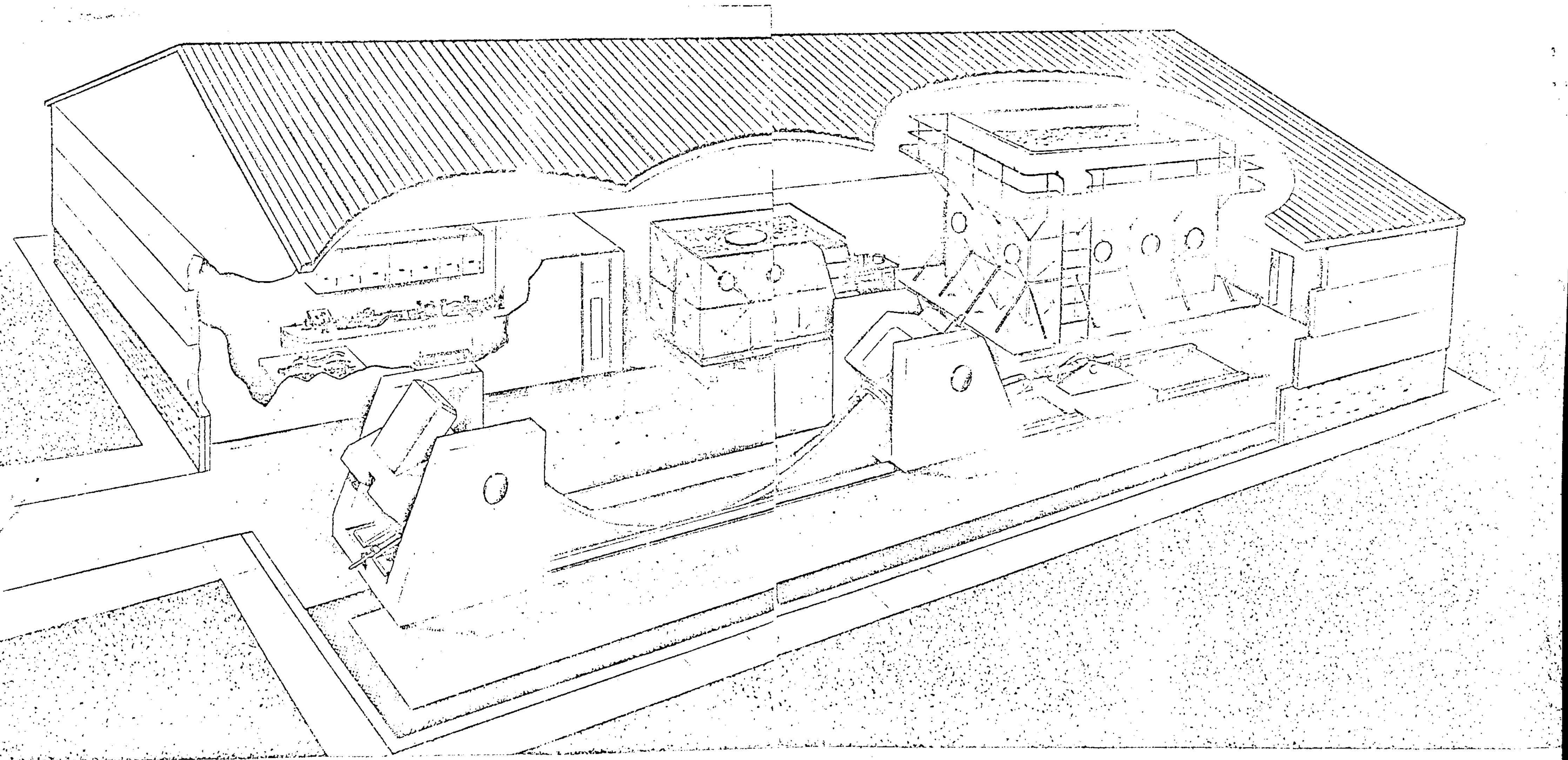


FIGURE (36)

TECHNICAL COMPLEX

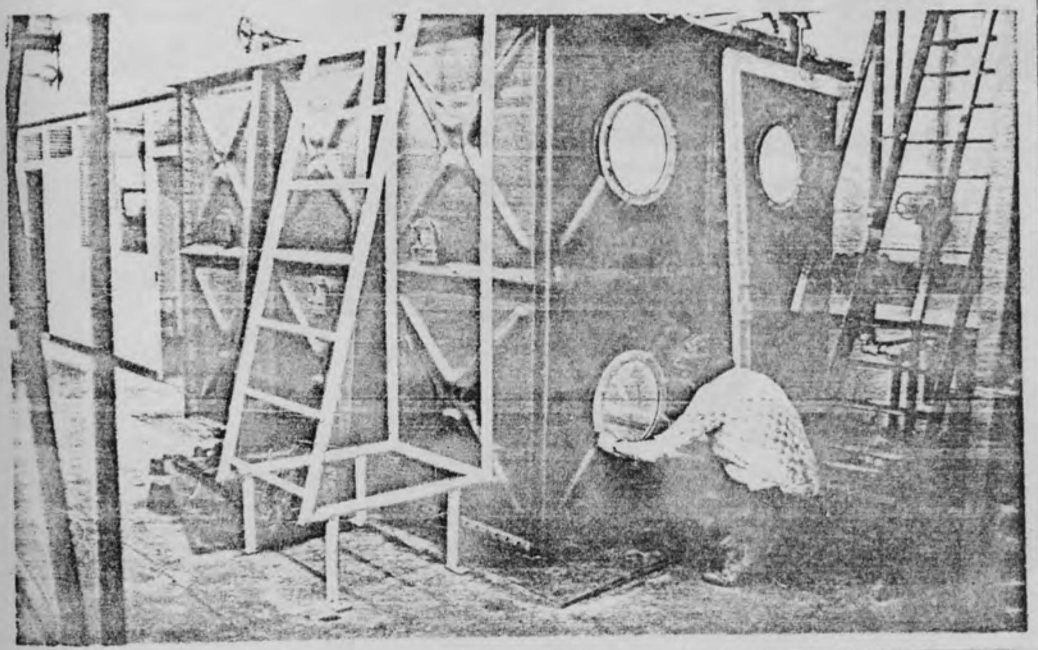


FIGURE (37)

SMALL TEST TANK

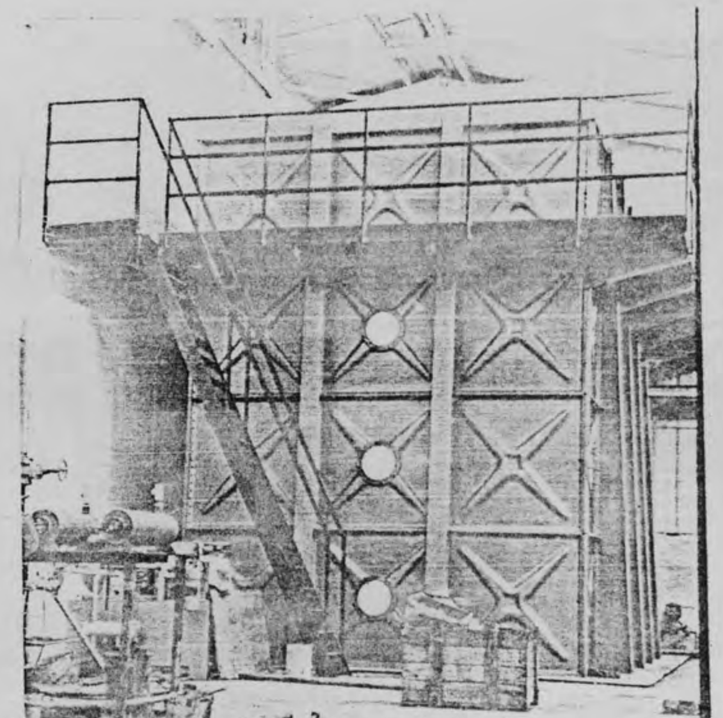


FIGURE (38)

LARGE TEST TANK

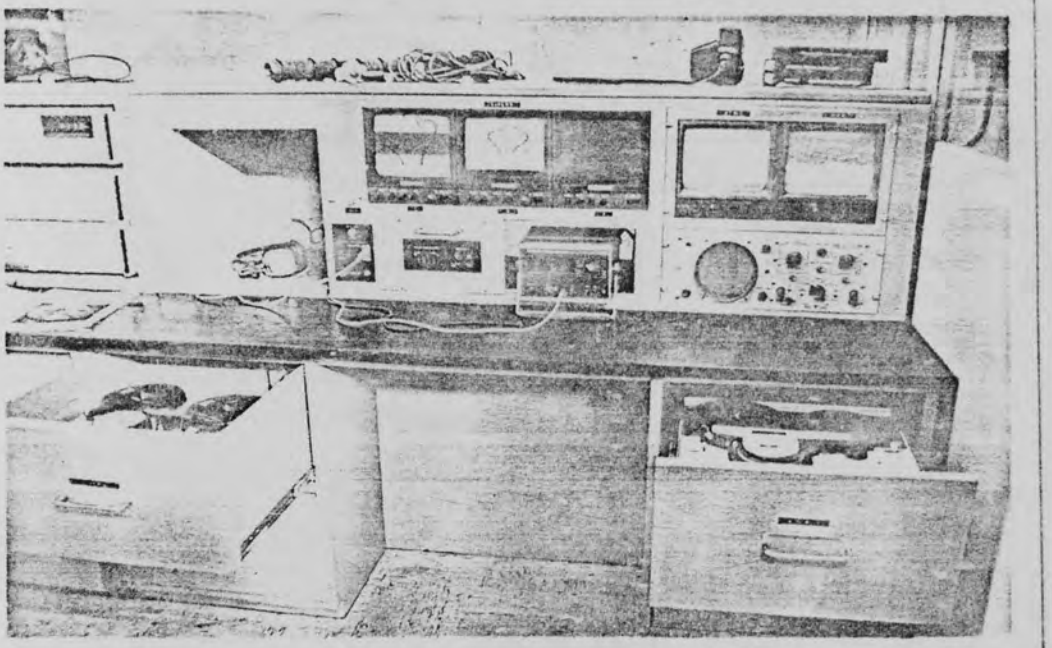


FIGURE (39)
TANK CONTROL ROOM

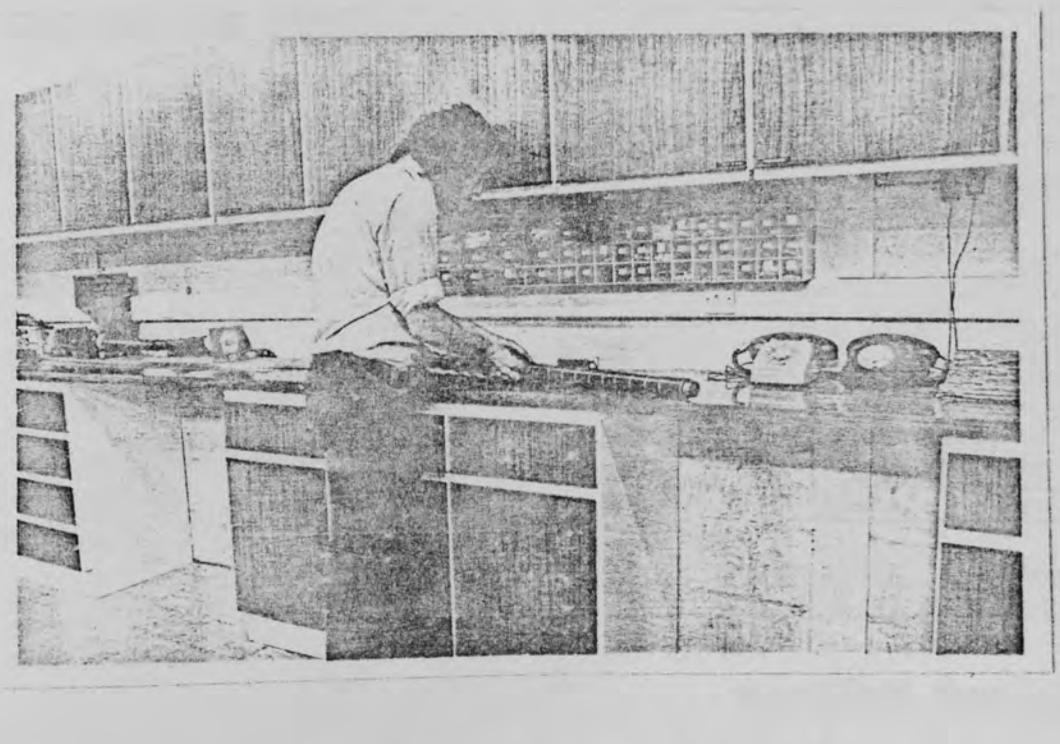


FIGURE (40)
MODEL ASSEMBLY LABORATORY

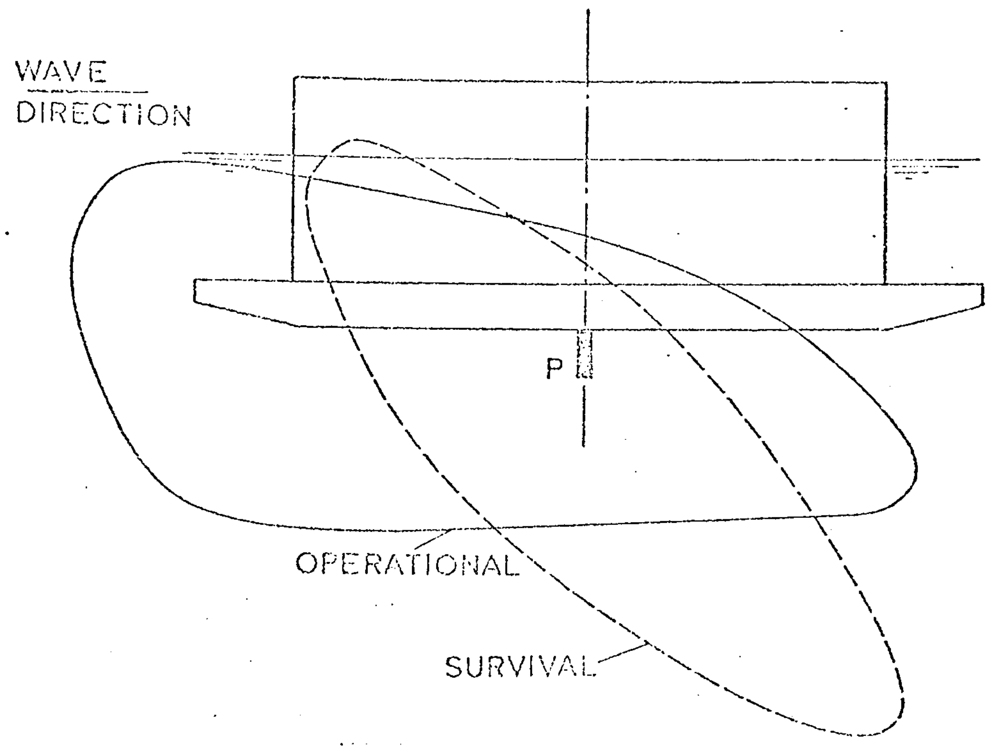


FIGURE (41)

MOTIONS OF THE SUSPENSION POINT OF THE UNDERBUOY HOSES OF A CALM BUOY

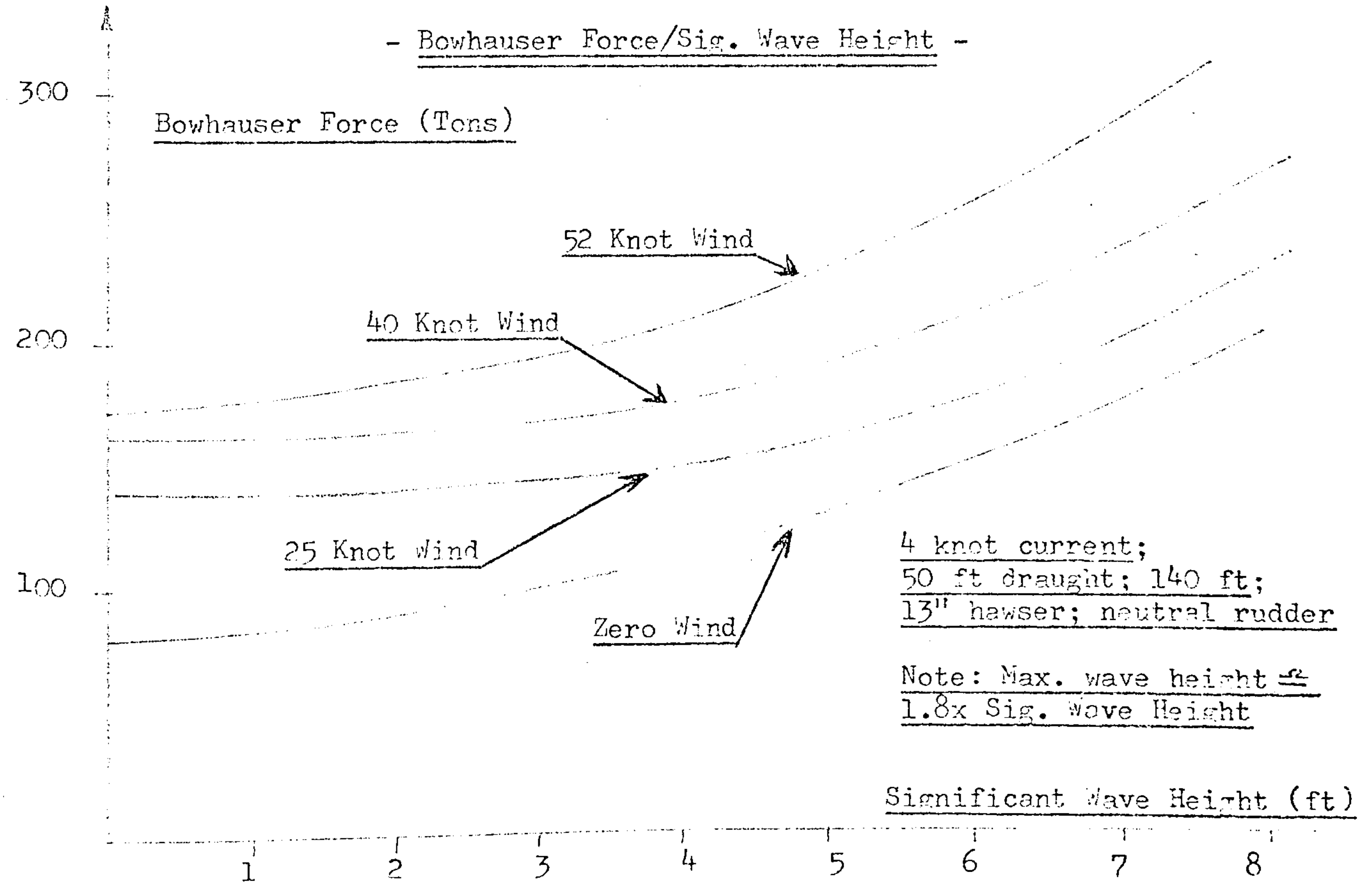


FIGURE (42)

CALCULATED TANKER MOORING LOADS

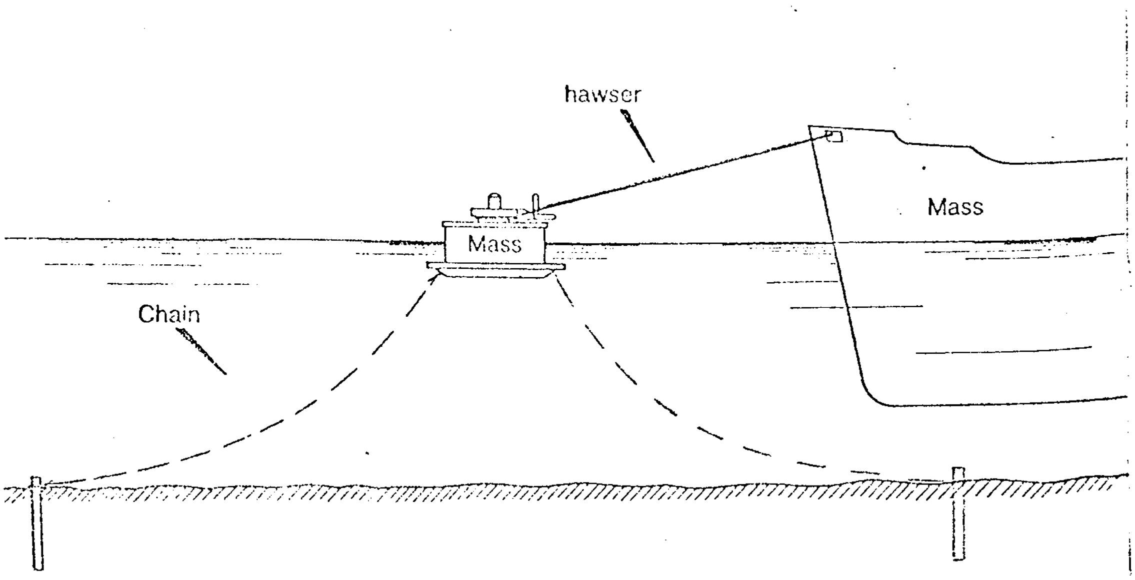
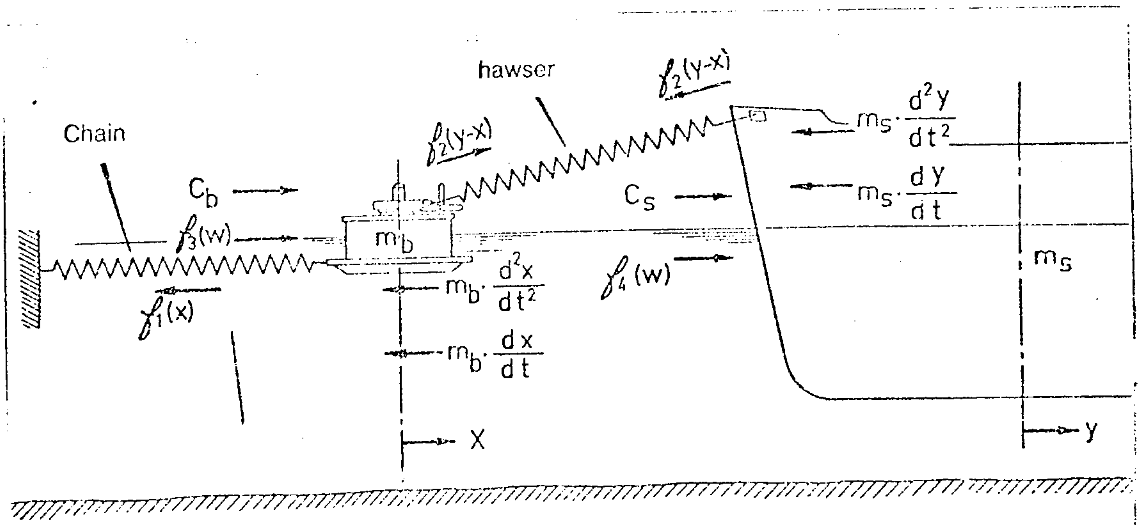


FIGURE (43)

CATENARY ANCHOR SYSTEM



BUOY: $m_b \cdot \frac{d^2x}{dt^2} + D_b \cdot \frac{dx}{dt} + f_1(x) = C_b + f_3(w) + f_2(y-x)$

SHIP: $m_s \cdot \frac{d^2y}{dt^2} + D_s \cdot \frac{dy}{dt} + f_2(y-x) = C_s + f_4(w)$

FIGURE (44)

SPRING MASS SYSTEM APPROXIMATION

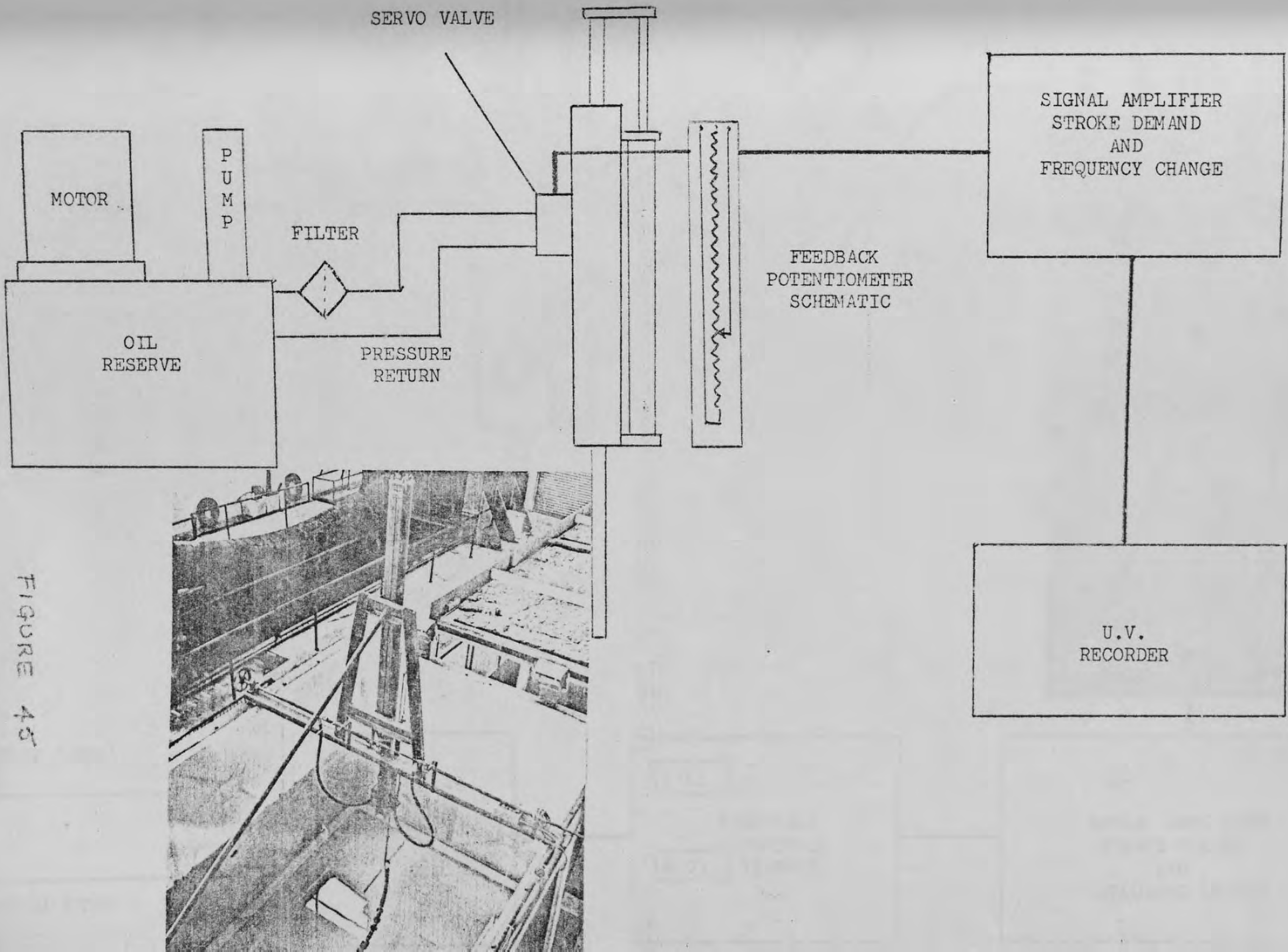

SERVO VALVE
MOTOR
PUMP
FILTER
OIL RESERVE
PRESSURE RETURN
FEEDBACK POTENTIOMETER SCHEMATIC
SIGNAL AMPLIFIER STROKE DEMAND AND FREQUENCY CHANGE
U.V. RECORDER

FIGURE 45

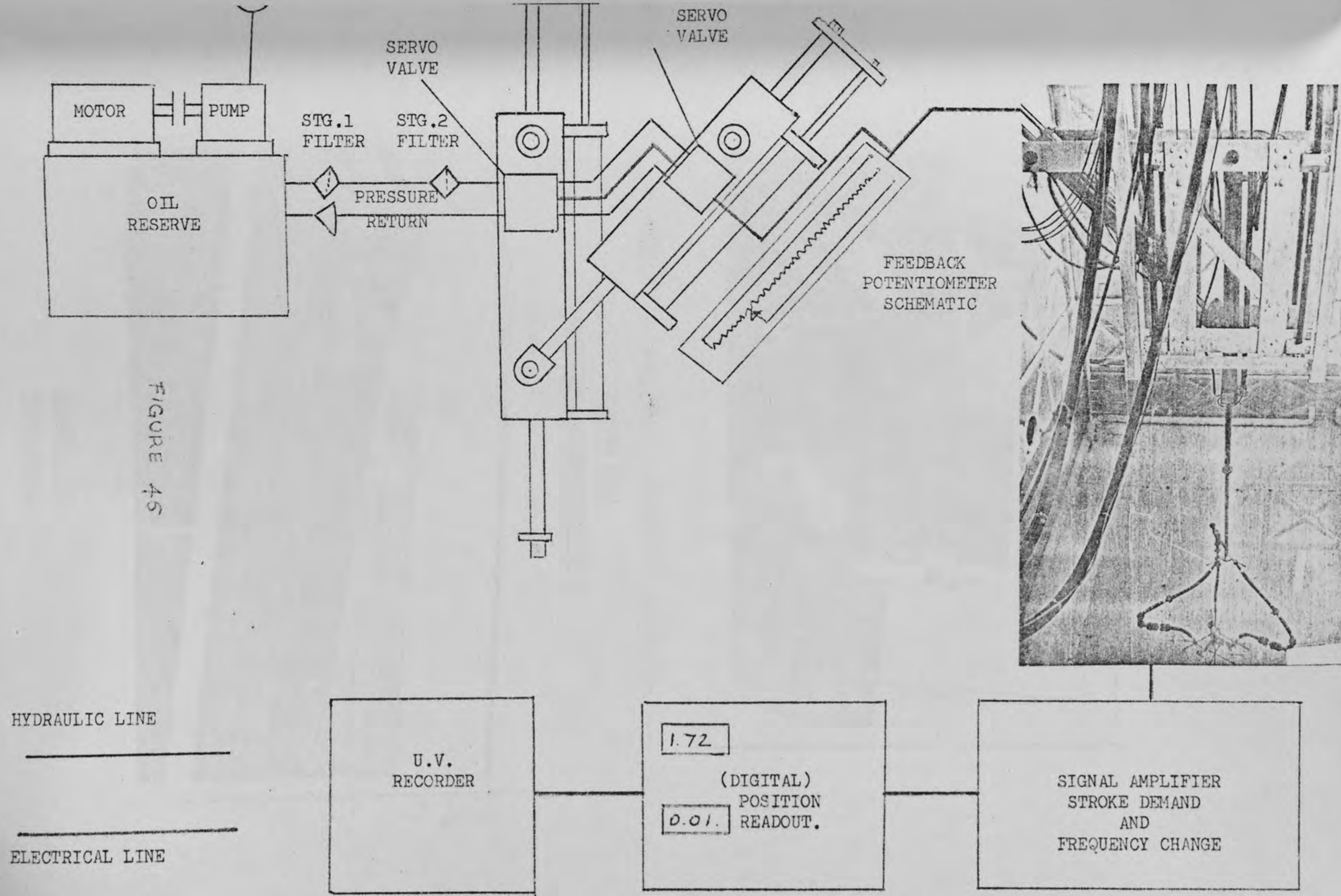

MOTOR
PUMP
OIL
RESERVE
STG.1
FILTER
STG.2
FILTER
PRESSURE
RETURN
SERVO
VALVE
SERVO
VALVE
FEEDBACK
POTENTIOMETER
SCHEMATIC
HYDRAULIC LINE
ELECTRICAL LINE
U.V.
RECORDER
1.72
(DIGITAL)
POSITION
READOUT.
0.01.
SIGNAL AMPLIFIER
STROKE DEMAND
AND
FREQUENCY CHANGE

FIGURE 45

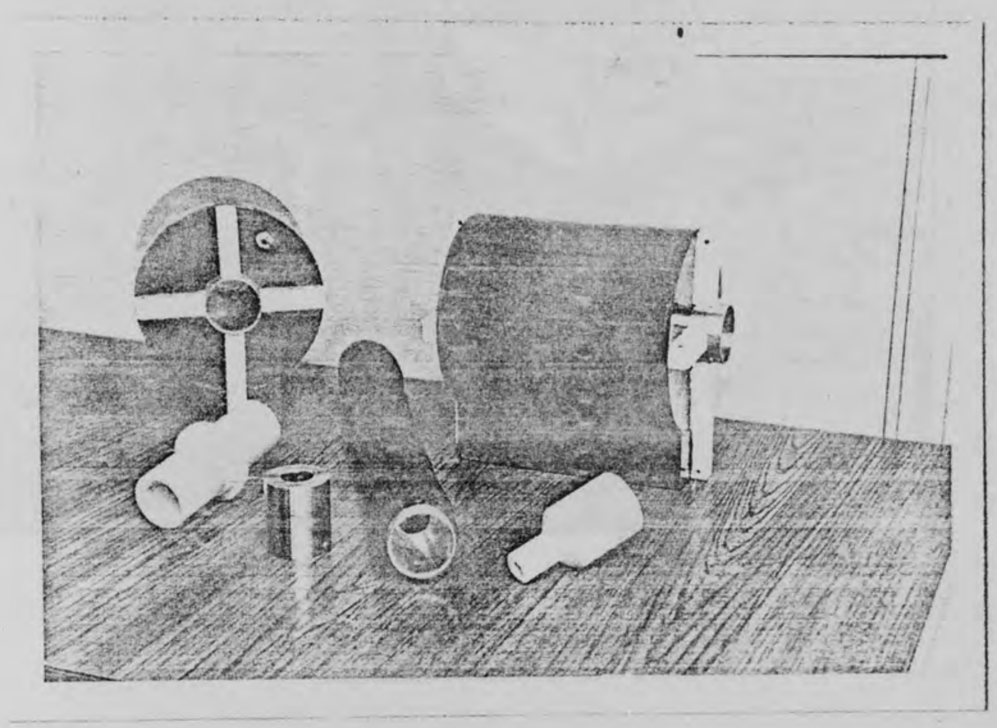


FIGURE (47)

ANCILLARY EQUIPMENT MODELS

FIGURE (48)

MODEL HOSE.

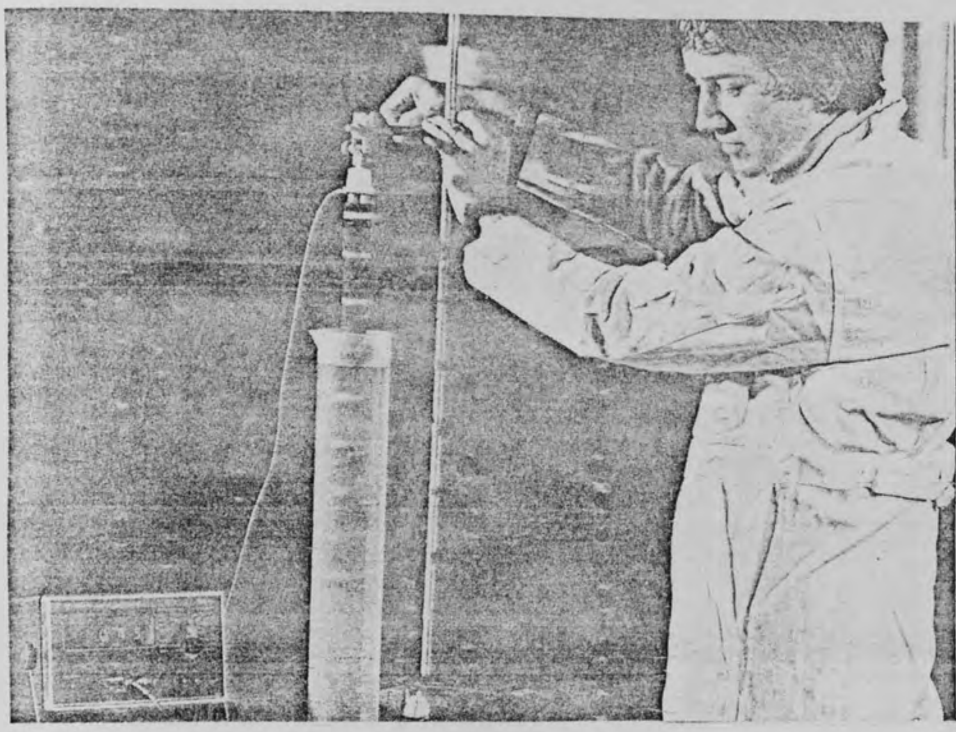
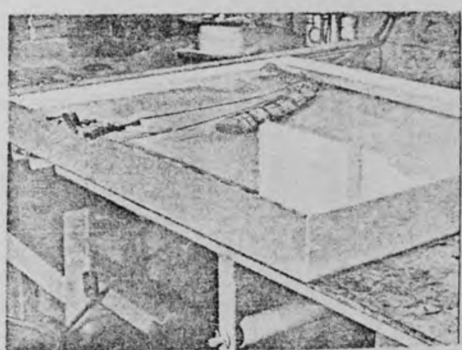


FIGURE (49)

UNDERWATER WEIGHT CHECK



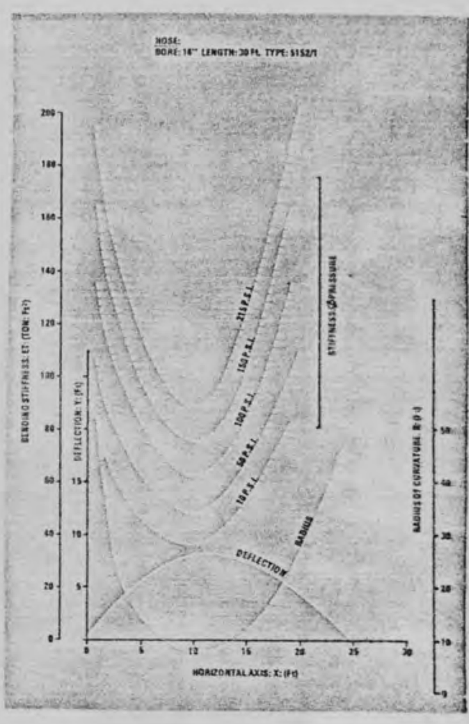


FIGURE (50)

MODEL BENDING CHARACTERISTIC TEST

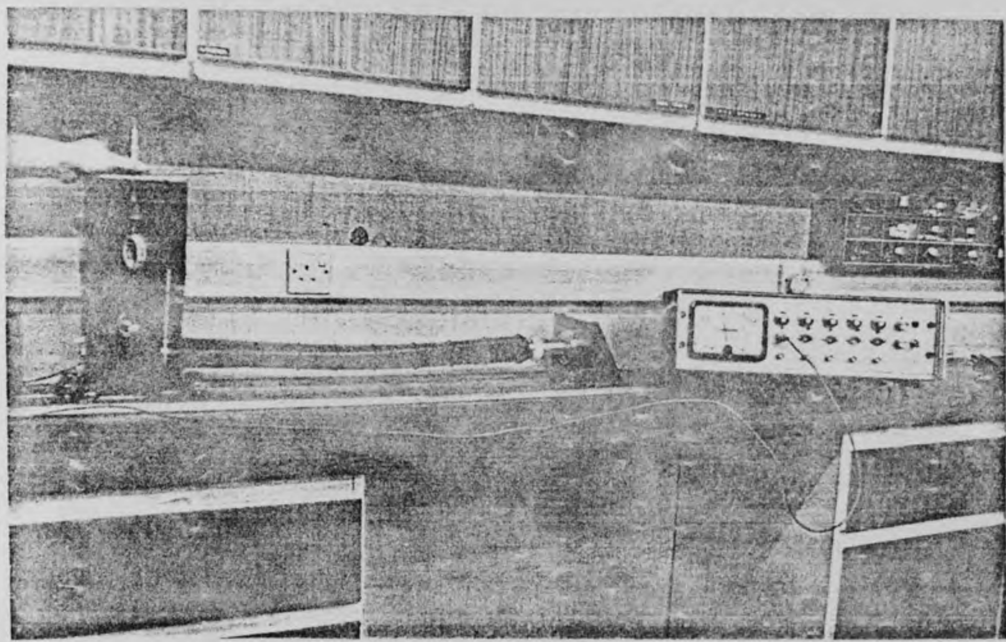


FIGURE (51)

TORSION TEST (MODEL HOSE)

FIGURE (52)

TENSION TEST (MODEL HOSE)

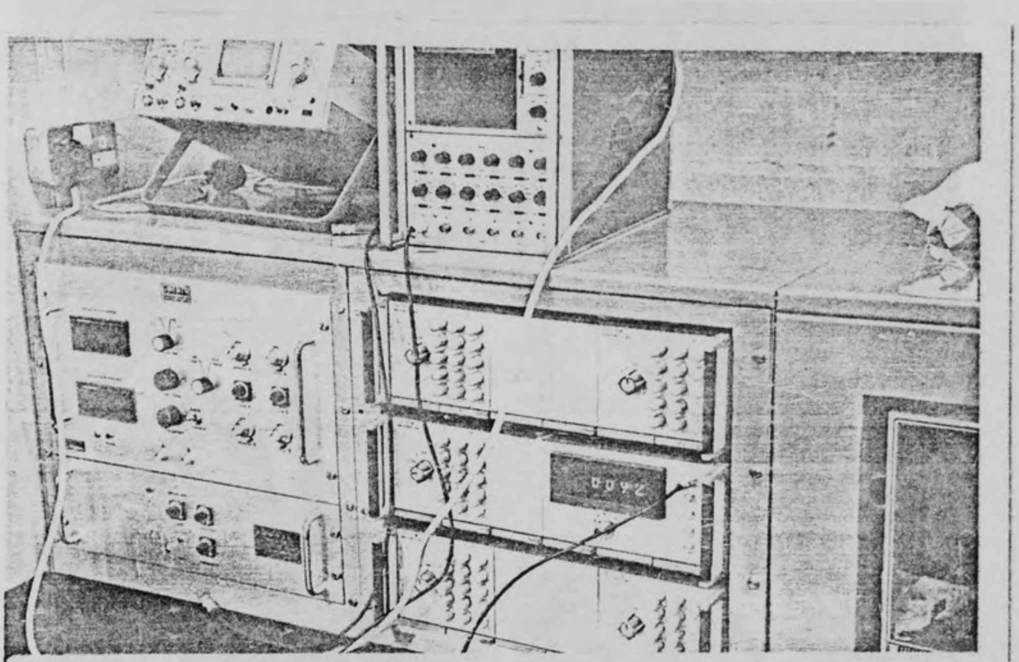


FIGURE (53)

TEST CONTROL CONSULE

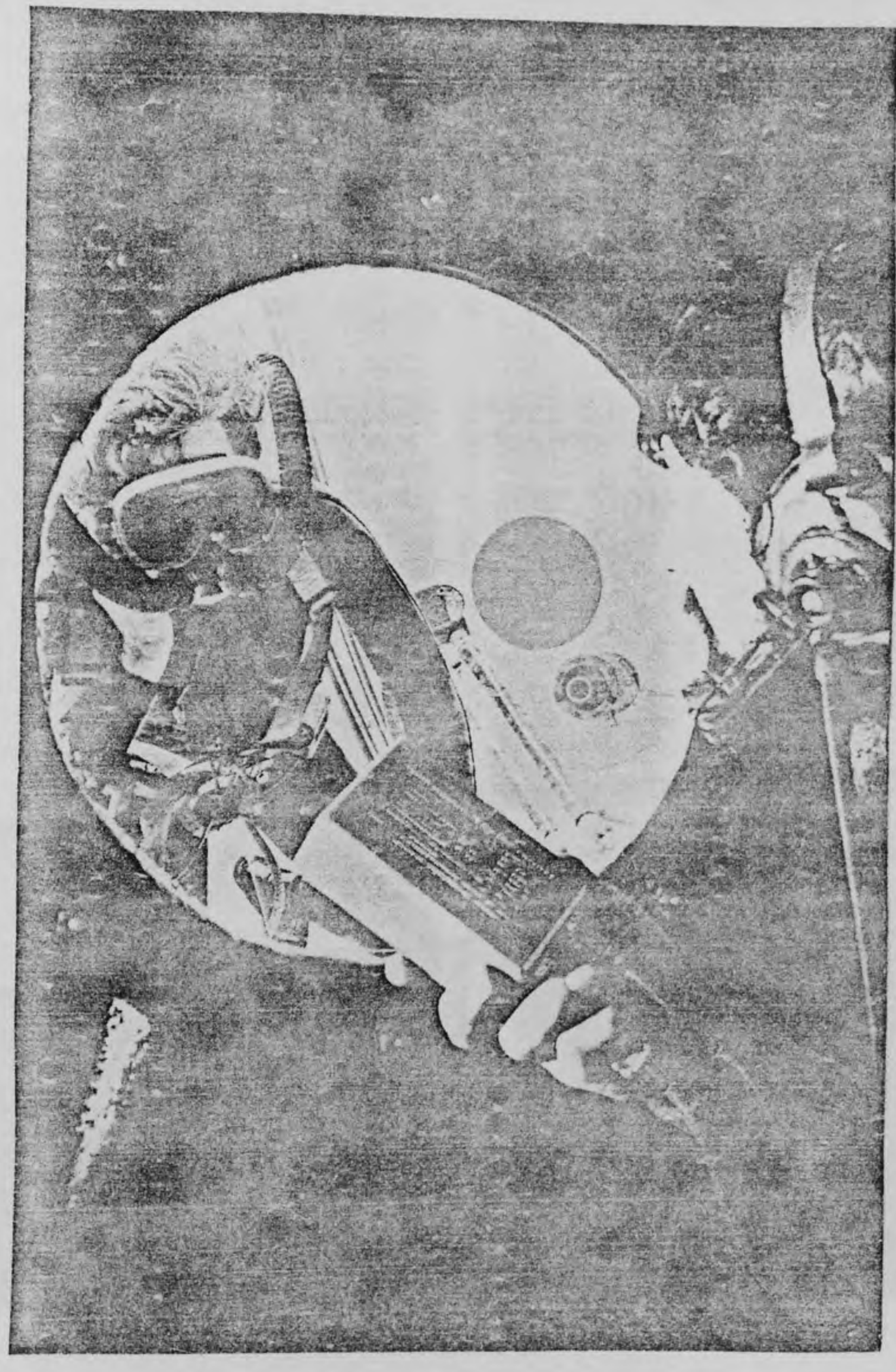


FIGURE (54)

TEST COMMUNICATION SYSTEM

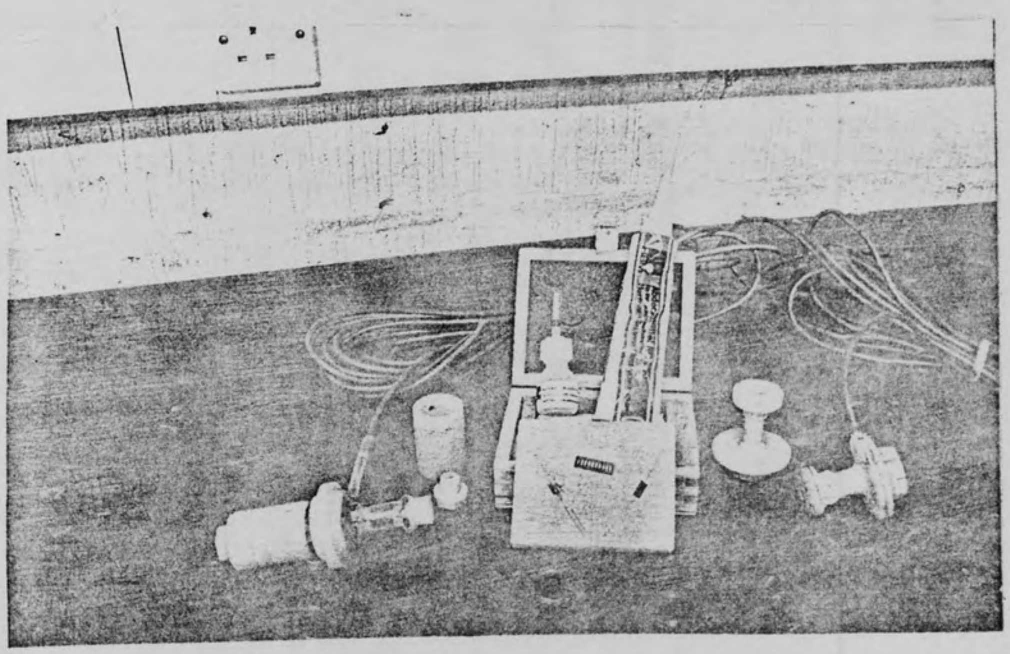


FIGURE (55)/(56)

1/15/(1/10) SCALE LOAD TRANSDUCER/COMPONENTS

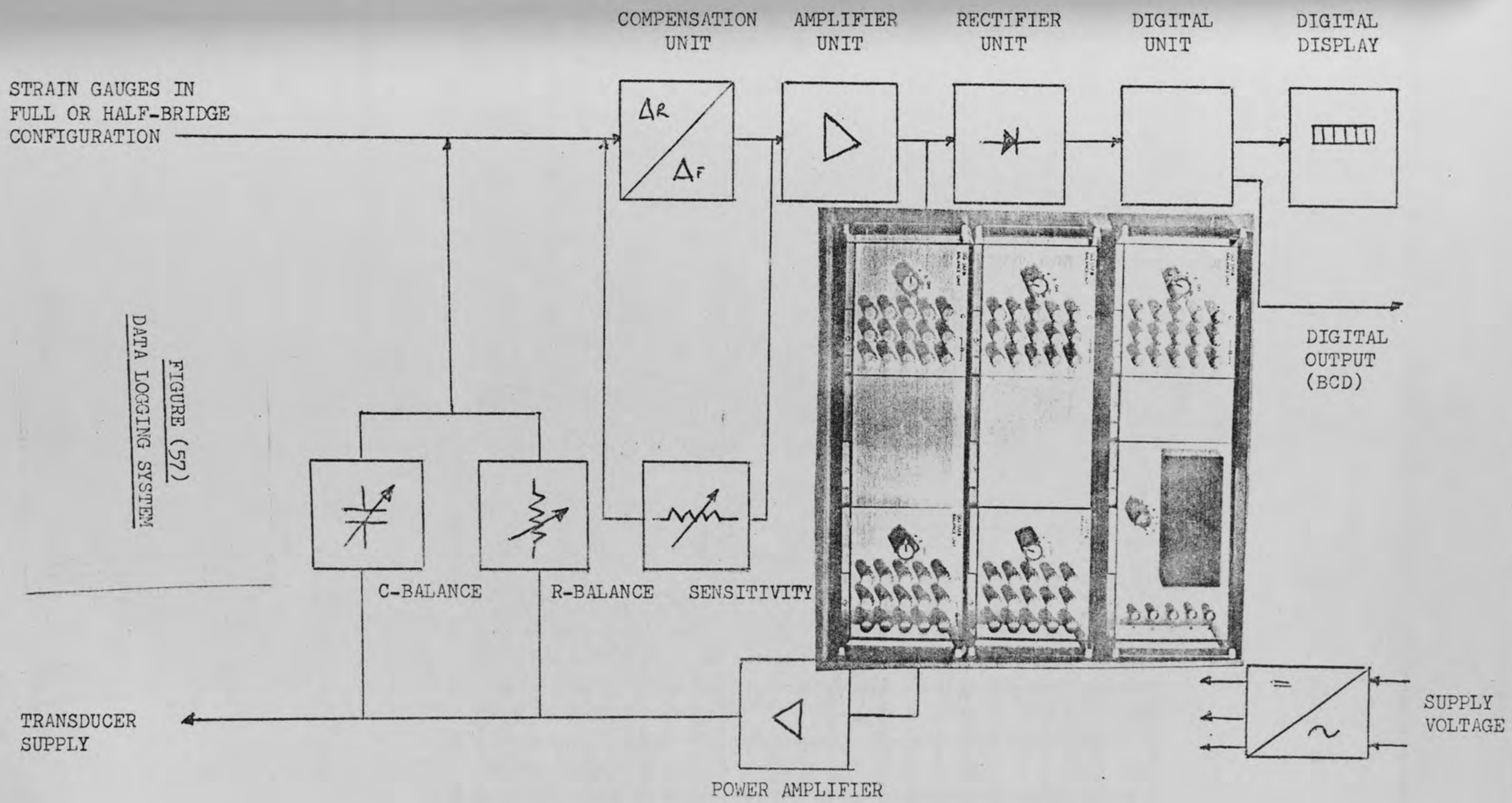


FIGURE (57)

DATA LOGGING SYSTEM

NUMBER OF WAVE OCCURRENCES:– H vs. T MATRIX

N.B. Numbers are rounded off

H \ T	0–1.0	1.0–2.0	2.0–3.0	3.0–4.0	4.0–5.0	5.0–6.0	6.0–7.0	7.0–8.0	8.0–9.0	9.0–10.0	10.0–11.0	11.0–12.0	12.0–13.0	13.0–14.0	14.0–15.0	Totals
Totals	241	121,639	604,150	991,635	867,061	849,218	920,311	779,152	472,379	192,610	49,351	7,520	706	56	5	5,855,034
11.5–12.0																
11.0–11.5								1								1
10.5–11.0								1								1
10.0–10.5							1	1								2
9.5–10.0							2	2	1							5
9.0–9.5							3	3	2	1						9
8.5–9.0						2	7	6	3	1						19
8.0–8.5						5	13	11	6	3	1					38
7.5–8.0						13	24	21	12	5	1					[illegible]
7.0–7.5						33	47	39	24	10	2					155
6.5–7.0						77	90	76	46	19	5	1				313
6.0–6.5					18	153	172	146	88	36	9	1				624
5.5–6.0					76	295	331	281	179	69	17	3				1,242
5.0–5.5					234	568	637	540	325	132	33	5				2,477
4.5–5.0					633	1,594	1,226	1,036	628	255	64	9	1			4,948
4.0–4.5				11	1,583	2,107	2,357	1,995	1,208	490	124	18	1			9,901
3.5–4.0				583	3,131	4,060	4,533	3,838	2,323	943	239	35	3			19,698
3.0–3.5				2,486	6,204	7,827	8,718	7,381	4,469	1,816	461	68	6			39,436
2.5–3.0				8,200	12,378	15,100	16,766	14,197	8,597	3,495	898	132	11	1		79,768
2.0–2.5			1,287	22,410	24,836	29,157	32,255	27,309	16,541	6,729	1,713	256	22	2		162,566
1.5–2.0			13,231	49,076	50,482	55,360	62,049	52,535	31,829	12,956	3,305	496	44	3		332,369
1.0–1.5			57,681	108,877	103,444	109,085	119,376	101,071	61,254	24,953	6,378	964	88	7	1	693,178
0.5–1.0		14,744	159,613	244,529	214,359	211,478	229,692	194,465	117,897	48,070	12,315	1,876	176	14	1	1,449,229
0–0.5	241	106,894	372,339	555,462	449,627	410,804	442,011	374,198	226,956	92,627	23,794	3,654	352	29	3	3,058,991

WAVE PERIOD (seconds)

Figure 5 – Texaco Nigeria Buoy – Average Year

FIGURE 59

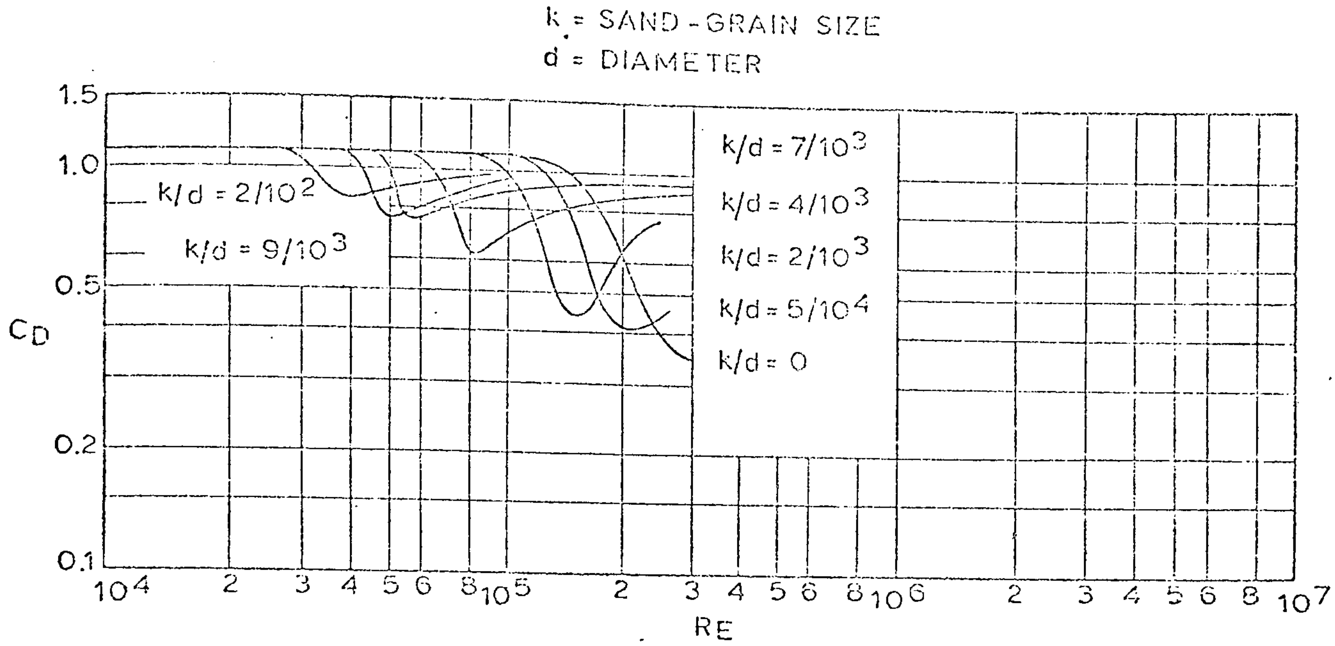


FIGURE (60)

INFLUENCE OF SURFACE ROUGHNESS ON THE DRAG COEFFICIENTS OF CYLINDERS (REF 6)

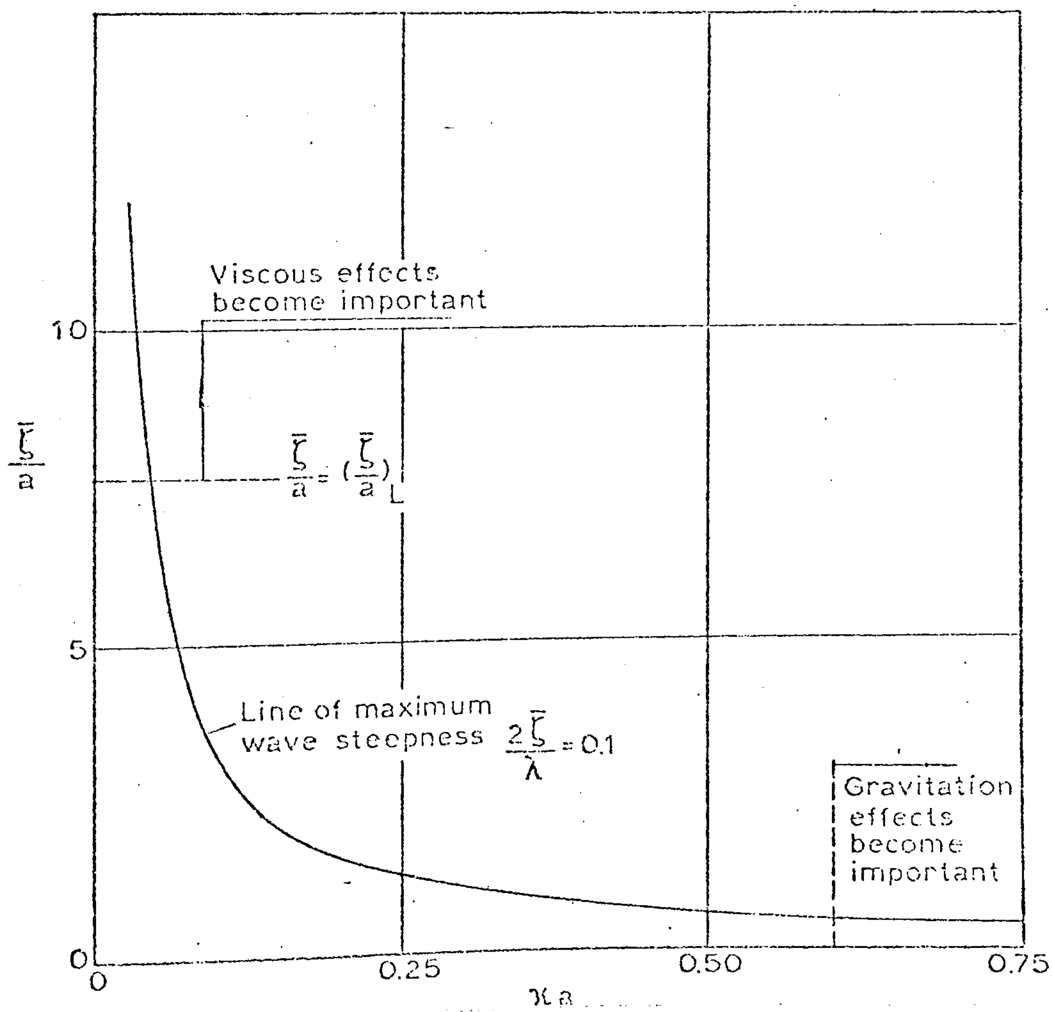


FIGURE (64)

WAVE FORCES OR CYLINDRICAL BODIES (REF 5)

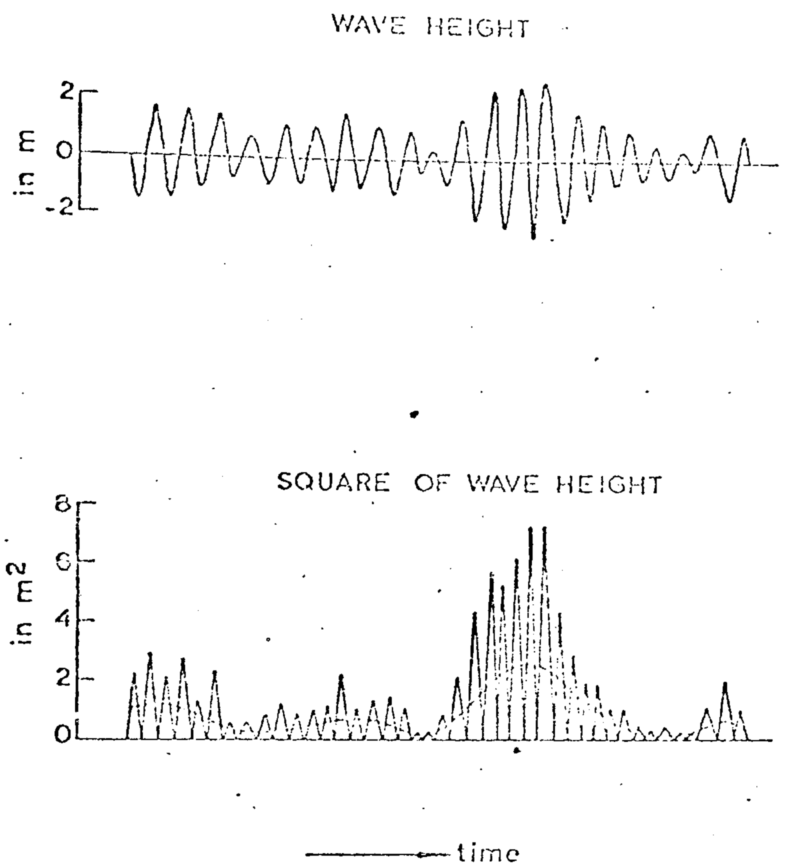


FIGURE (65)

IRREGULAR WAVE AND SQUARE OF WAVE HEIGHT

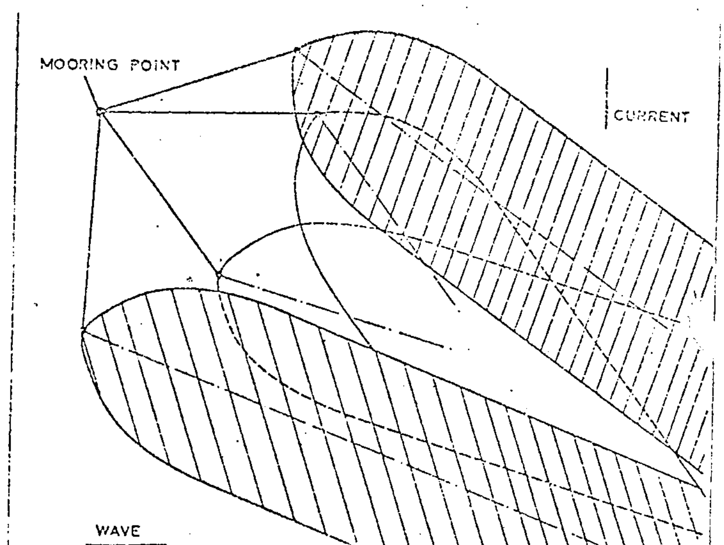
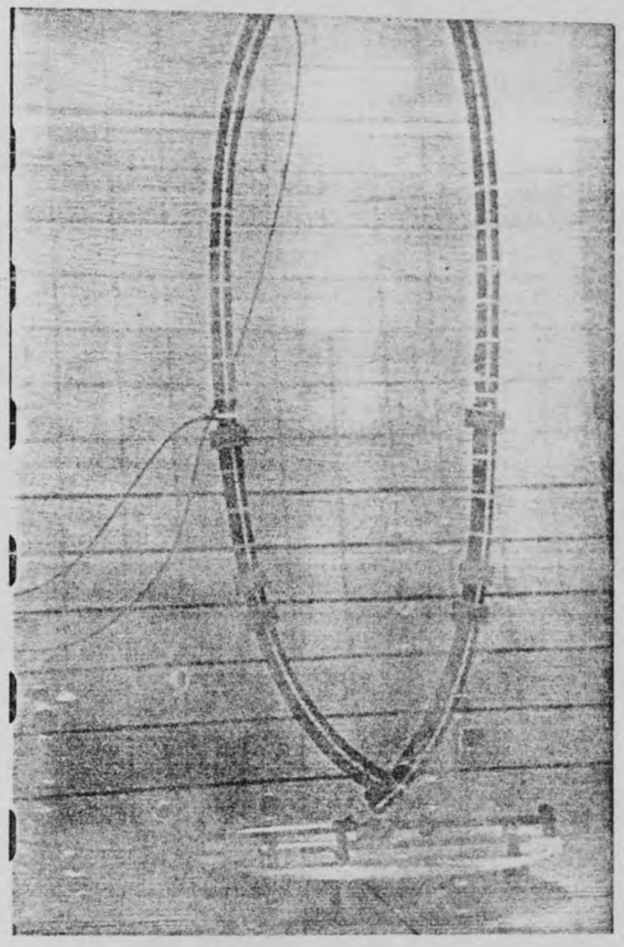


FIGURE (66)

MOORED TANKER YAWING MOTIONS



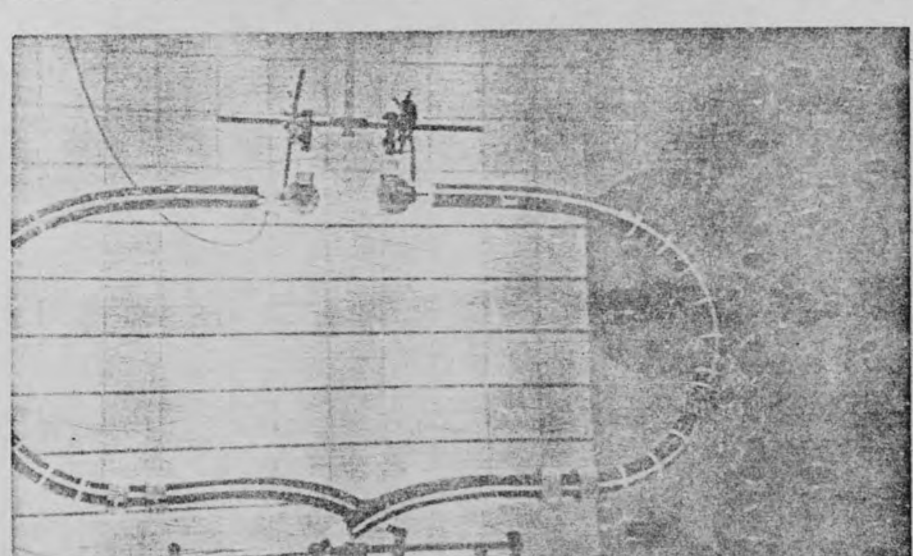
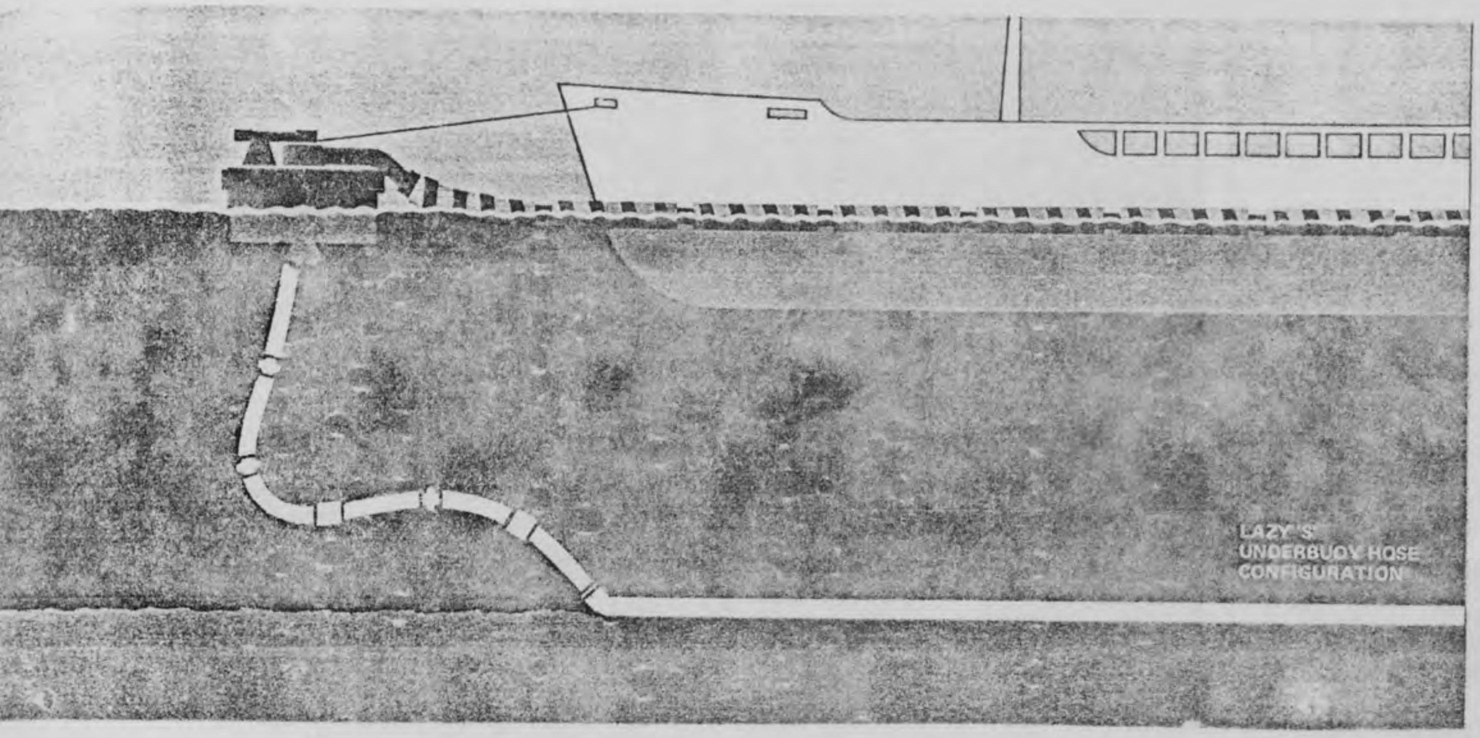


FIGURE (130) & (131)



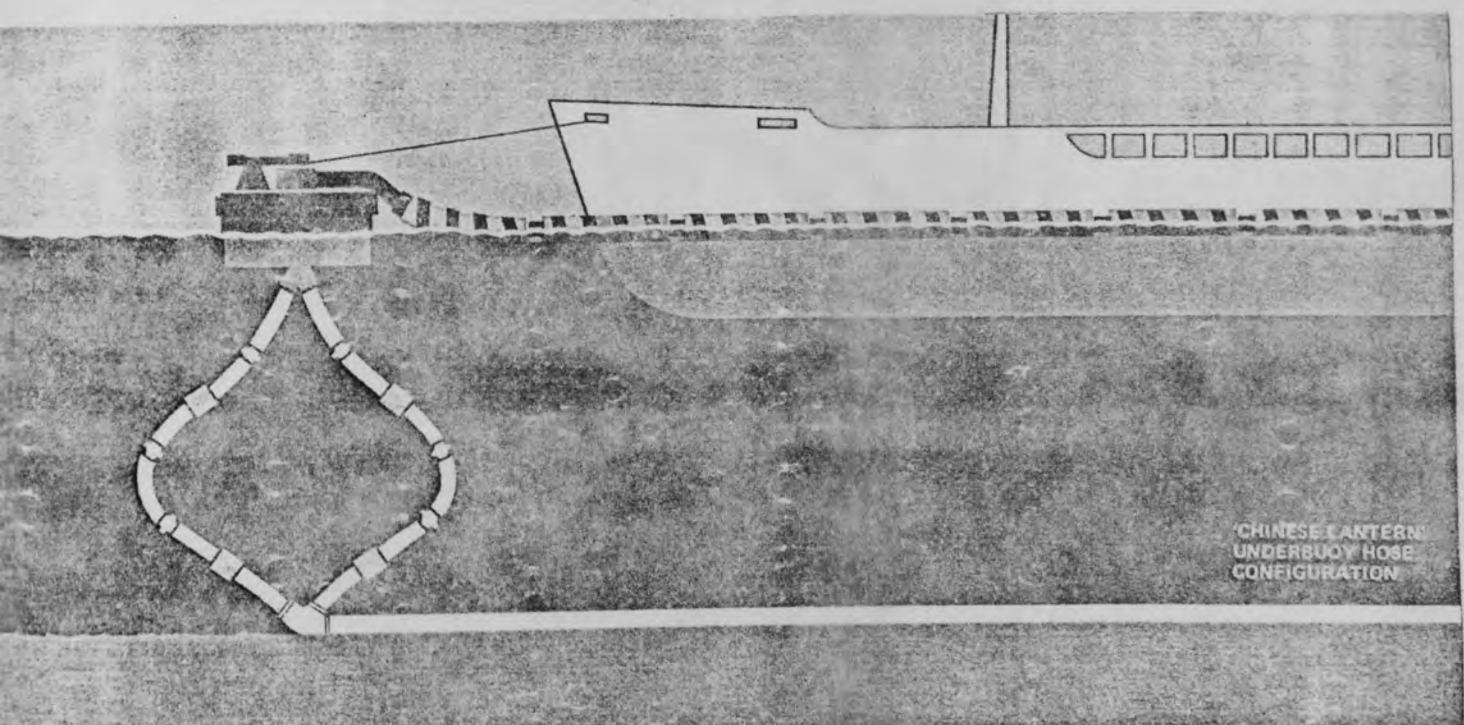
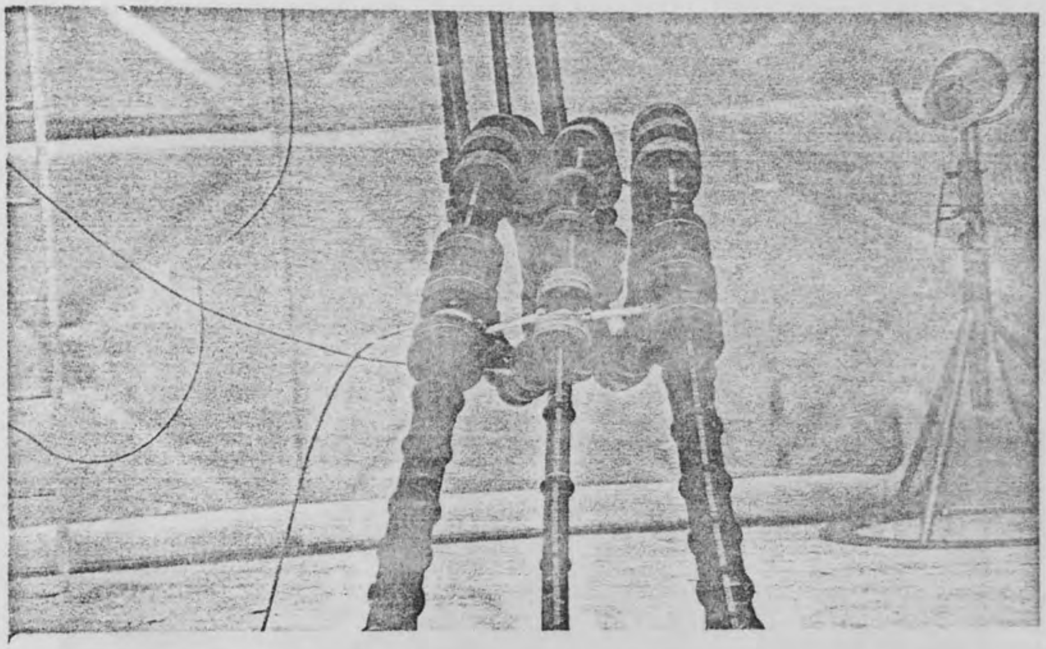


FIGURE (132) & (133)



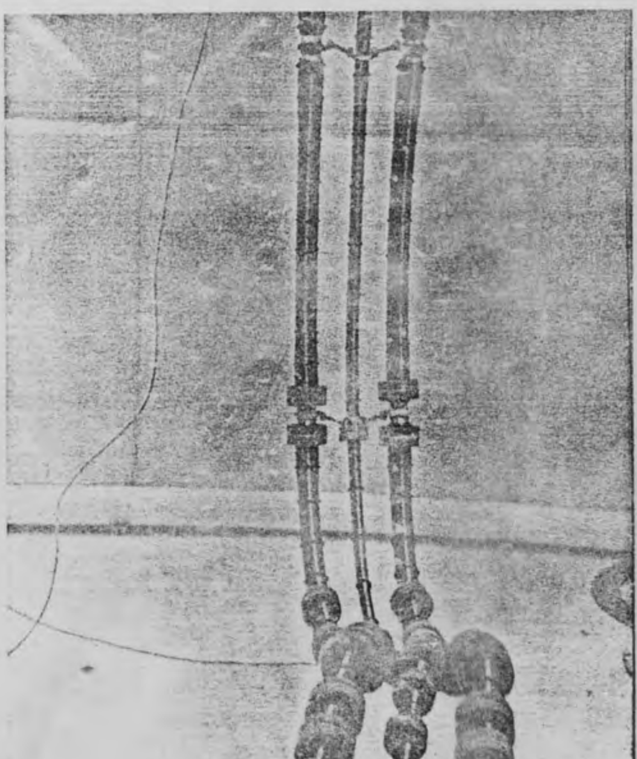


FIGURE (134)

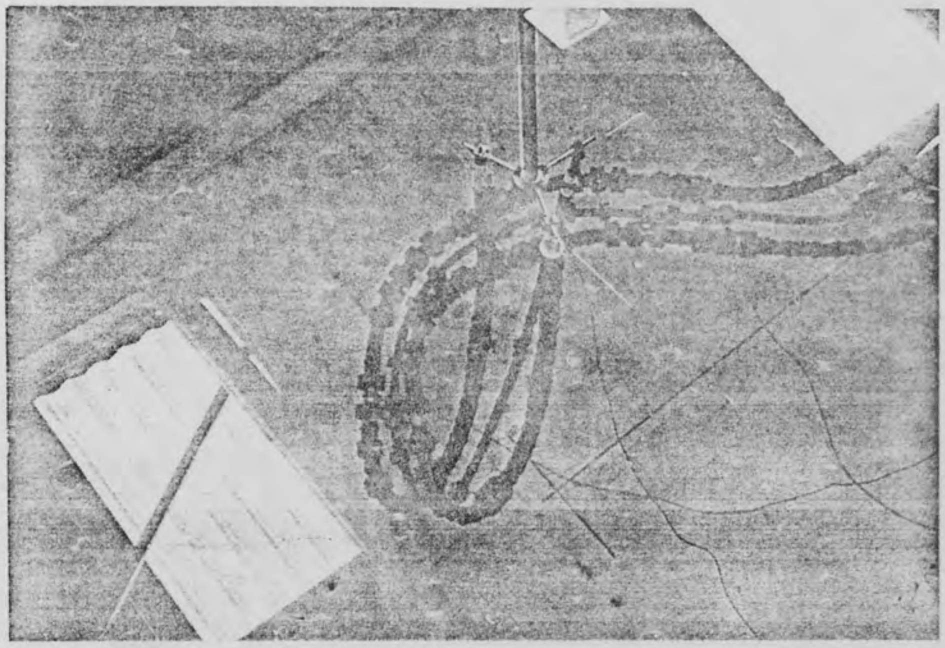


FIGURE (135)

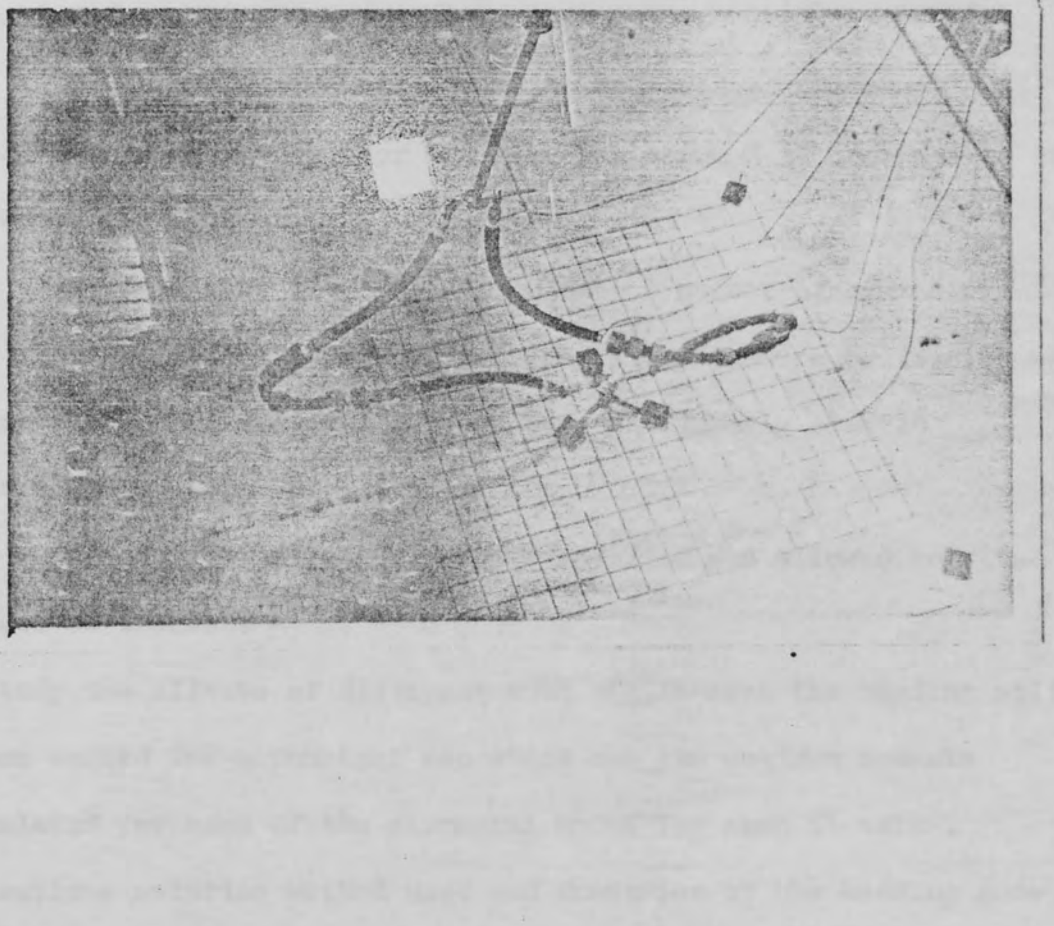


FIGURE (136)

VII MATHEMATICAL ANALYSIS OF SERVICE LOADINGS

VII.1 FLOATING HOSE BEHAVIOUR PREDICTION MODEL

The purpose of this investigation was to predict the dynamic behaviour of a floating hose string for a single point mooring system subjected to regular wave action.

The method of solution adopted was to model the hose string by a number of elements of known weight, displacement and stiffness and to integrate the equations of motion until a steady state solution was obtained. Linear wave and beam theories were used however the nonlinearity of damping and buoyancy forces were accounted for.

A number of basic assumptions were made:

1) The system was assumed to be in deep water such that linear wave theory was adequate. Further the hose was assumed to be close to the surface.

2) In idealizing the hose string into a number of beam elements, only vertical deflections were considered, no tension or torsion was defined in the hose, small deflections and linearly elastic materials were assumed.

3) Buoy motion is considerably simplified and allowed only two degrees of freedom.

To study the effects of different hose stiffnesses the bending stiffness EI was varied for a constant sea state and the maximum moments calculated for each of the elemental nodes for each EI value.

The outline solution method used and summaries of the bending moments versus hose stiffness are contained in Appendix (2).

Of particular interest however is the increase in bending moment at the buoy flange as the hose stiffness is increased.
Figure (67) indicates the maximum moment versus hose stiffness (at the buoy manifold) for the example used (ie: 8ft, 8 sec regular wave); shown circled is an experimental point taken from the Forcados field study.
Whilst no definitive conclusions may be drawn the programme assumptions appear valid for the simplistic approach taken. Many more data points are needed, and to this end it is intended to conduct further runs of the programme for various sea states. In order to do so a number of wave probability histogrammes have been calculated (see Figure (21) and (21)) and will be used to obtain similar load histogrammes for the various sites available. Initially Forcados Nigeria will be used and the results compared with those obtained from the field study.
However before such work is performed the programme requires refinement. Initially to allow for greater stiffness variation, but also to allow for the incorporation of tension and torsional generated loads.

VII.2 BUOY EXCURSION UNDER THE INFLUENCE OF LATERAL FORCES

Data acquired from a variety of sources in order to satisfy the needs of the various model tests which have been undertaken has seemed to indicate the presence of a rather confusing picture for buoy excursions. In some cases the excursions for the operational condition have exceeded those for the non-operational condition and in others the reverse has occurred. In some cases the excursion has been said to increase with depth whilst more frequently the excursion has reduced with increasing depth. It is clear then that an approach from basic principles is needed in order, firstly, to increase our understanding of buoy systems and the effect of their design upon hose behaviour and, secondly, to be able to provide a more complete service to our customers when our advice for system design is sought.

It has been found possible to treat the problem entirely from a theoretical standpoint in order to derive excursion predictions under known loadings which can be regarded with confidence. The work performed has been based on the relationships relating to the catenary and a buoy system of six anchor legs has been considered.

The known parameters for any system are:

L - the total length of each chain

b - the depth to be considered

W - the Wt/unit length of chain

and α - the chain angle at a known depth and position

From these parameters it is necessary to find a relationship between the excursion and the horizontal force causing the excursion. The mathematics used to derive this relationship appears in Appendix (10). Having obtained this relationship for a single chain the problem of sharing the total horizontal force between all six chains still remains and it has been found easier to specify the excursion and from that evaluate the horizontal component of the tension in all the chains and then sum the results. In this way each chain can be treated independently rather than solving a set of simultaneous equations for the entire system.

For these calculations the buoy position must be related to some fixed point and it is most convenient to consider this fixed point as being the piling at the very end of each anchor chain. The "displacement" is then measured from this point. In the equilibrium position, at which the known value of α was measured, the "displacement" figures for all six chains will be the same. If the buoy is now displaced a distance x then, generally, three of the six "displacement" figures will increase and three will decrease. In one particular case only four of the six chains will be affected as the other two will be in the plane perpendicular to direction of motion. The amount of change will depend upon the chain under consideration. For the two chains in the plane nearest that of the translating force (at angle θ to it) the change will be $\pm x\cos\theta$ and for the other four chains the change will be approximately $\pm x\cos(60 \pm \theta)$. This change is approximate because the angle which is initially 60° does in fact change slightly as the buoy moves. The differences from 60° are so small however that for all practical purposes the approximation is fully justified. The small errors introduced by accepting this approximation are explained in Appendix (10).

The problem of relating excursion to translating force is treated numerically as no analytical approach has been found suitable. The procedure is outlined in the following steps.

(i) Find the maximum chain angle at a specified depth at which all of one chain will just lift from the sea bed.

(ii) Find the maximum chain angle at the same depth at which the chain will be drawn taut.

(iii) Find the relationship between displacement (horizontal distance from buoy to anchor pile) at the depth at which the known chain angle was measured for all angles (1° interval) up to the maximum evaluated in (ii). The approach adopted is finding this relationship is different if the angle is between 1° and the maximum found in (i) compared with the case when the angle is between this maximum and the absolute maximum found in (ii).

(iv) With a new depth (if it is to be changed) at which the excursion is to be considered repeat steps (i), (ii) and (iii).

(v) By comparing the relationships obtained in the first and second runs through step (iii) obtain the equilibrium angle at the new depth.

(vi) Add - or subtract according to the chain under consideration - the appropriate excursion figure to the equilibrium value of displacement.

(vii) Find what value of horizontal force corresponds with this value of displacement by referring to the table formed in the second run through step (iii).

(viii) Repeat steps (vi) and (vii) for all the chains with the appropriate displacement for each chain related to the buoy.

(ix) The forces obtained can be added to give the total horizontal force causing the excursion.

This process could obviously be very tedious as a number of the stages have to be repeated several times and each stage itself requires usually about 70 or 80 similar calculations to be repeated. This kind of problem is ideal for solution by computer and an interactive programme for its solution has been written.

Figures (68) and (69) show graphs typical of the output from the computer programme written. Fuller sets of graphs, tables etc are included in appendix (13).

From a study of the results of the above claculations, figures etc. it is possible to list a number of characteristics common to all multi catenary chain anchor buoy systems:-

(i) The force required to cause a particular excursion increases with increasing depth but is not in direct proportion to the depth. A typical relationship is shown by Figure (68).

(ii) The difference in maximum excursion at widely separated depths is, perhaps surprisingly, relatively small.

(iii) Looking at the buoy system in plan, the form of the excursion caused for any particular translating force acting in any direction around the full 360° is virtually hexagonal in shape, the corners of the hexagon corresponding with the mid points between each of the chains. See Figure (69).

(iv) Providing the entire length of the chains opposed to the translating force is not lifted from the sea bed the additional force required to cause equal increments in excursion rises only very slowly and could almost be approximated to a straight line.

(v) The gradient of this part of the curve (before the chain entirely lifts from the sea bed) increases with increasing depth.

(vi) When one or more chains opposed to the translating force has extended beyond the point at which some of its length still lies on the sea bed the translating force required to cause further excursion begins to rise very sharply. The shape of the curve during this phase cannot be approximated to a straight line as the gradient very rapidly rises to infinity.

VII.3 MATHEMATICAL MODEL OF SUBMARINE HOSE CONFIGURATION

In the practical situation the submarine hose is a complex structure with many factors involved in causing internal loads. The loading of the hose will be a direct consequence of the geometry and physical properties of both the hose and its environment. The following factors affect the stress pattern.

(i) Size of hose, both length and cross-section.

(ii) Elastic properties of the complex material from which the hose is made.

(iii) Weight of hose.

(iv) Depth of the sea; this is variable and the effect of buoy movement in a vertical direction is important certainly because of variation of depth and from a dynamical point of view.

(v) Conditions at the manifolds at either end of the hose.

(vi) Drag loads exerted by the water on the hose.

Any mathematical model which includes all the parameters given in (i) - (vi), would be a complex model making the analysis of the problem very difficult. A simplification of the problem can be obtained by noting that the length of the hose is large compared to the diameter so that as a first approximation the hose can be considered to be a thin strut fixed between two points (the bucy and the sea bed manifold). The construction of the hose is complex and elastic properties are difficult to predict so that it seems reasonable to consider the structure as uniform and isotropic with a bending stiffness that can be measured. With these two simplifications the problem can be written down in mathematical terms. The quations for the problem are well known - they are the Euler equations for the buckling of a thin elastic rod.

Proceeding a stage further, the next approximation is that of a thin rod constrained to lie between two fixed points but with the lower end pivoted rather than clamped, so that the bending moment at the lower end vanishes. The equations are the same as those of the first approximation, the difference being in the boundary conditions.

It may be possible to obtain numerical solutions to these two problems, but before any attempts are made to find such numerical solutions some attempt at analytical solutions should be made.
In the following analysis an attempt is made to find an analytical solution to determine the forces and bending moments in a uniform thin elastic rod fixed between two points in the same vertical line.
This model is subject to the following restrictions:-

(a) At each end the rod is clamped in a vertical direction. (This implies that the manifolds on the sea bed and the buoy are vertical, which is not the practical situation).

(b) The loads in the rod are caused solely by the fact that it is longer than the distance between the fixed points. (This neglects the effect of drag forces and assumes that the hose is neutrally buoyant).

(c) The problem is a statical one. (Vertical movement of the buoy is not considered although the solution is found for different sea depths).

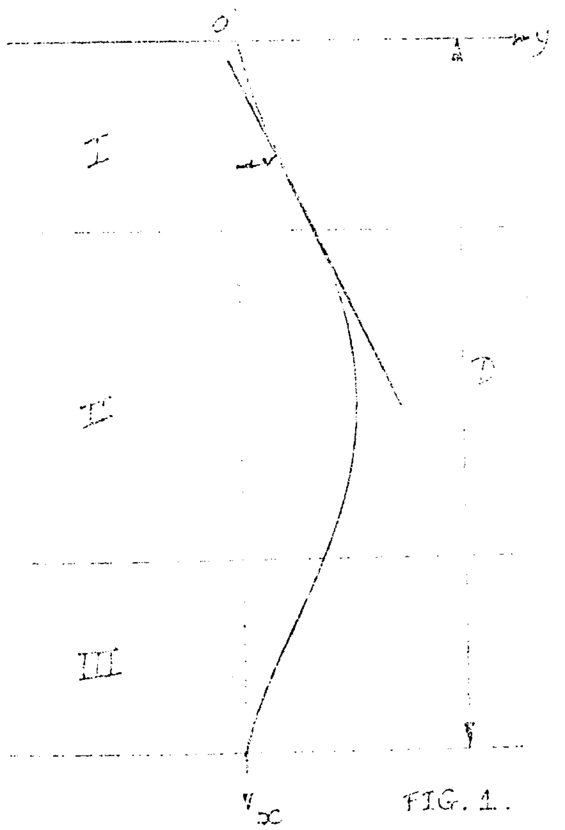


FIG. I.

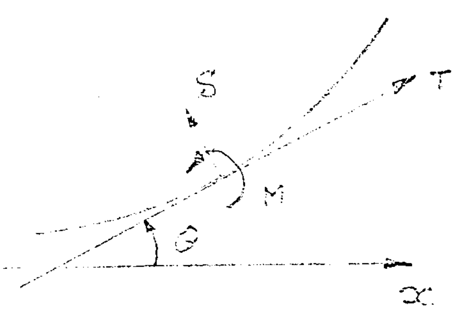


FIG. II.

T^*	Tension
S^*	Shearing force
M^*	Bending moment
s^*	Arc length measured along the curve
L	Total length of hose
θ^*	Angle between the tangent to curve and x^* axis
D	Distance between the fixed points.
B	= EI where E is Young's modulus and I the moment of inertia of the cross-section.

The equilibrium equations are (Mathematical Treatise on Elasticity - A E H Love)

$$\frac{dT^*}{d\theta^*} = S^*$$

$$\frac{dS^*}{d\theta^*} = -T^*$$

$$\frac{dM^*}{ds^*} = -S^*$$

$$M^* = B\frac{d\theta^*}{ds^*}$$

For the problem where the ends are clamped vertically the boundary conditions can be written as

$$x^* = 0,\ Y^* = 0,\ \theta^* = 0 \text{ when } s^* = 0$$

$$x^* = D,\ Y^* = 0,\ \theta^* = 0 \text{ when } s^* = L$$

These equations can be written in dimensionless form by the introduction of new variables.

$$S = \frac{L^2}{B} S^*,\ T = \frac{L^2}{B} T^*\ ,\ M = \frac{L}{B} M^*$$

$$s = \frac{s^*}{L},\ x = \frac{x^*}{D},\ \theta = \theta^*$$

Under this transformation the equations and boundary conditions become

$$\frac{dT}{d\theta} = S \qquad 1.1$$

$$\frac{dS}{d\theta} = -T \qquad 1.2$$

$$\frac{dM}{ds} = -S \qquad 1.3$$

$$M = \frac{d\theta}{ds} \qquad 1.4$$

and

$$\left.\begin{array}{l} x = 0, \quad \theta = 0 \quad \text{when } s = 0 \\ x = 1, \quad \theta = 0 \quad \text{when } s = 1 \end{array}\right. \qquad 1.5$$

Equations (1.1) and (1.2) may be integrated and since the problem is symmetrical the shearing forces at the ends must vanish which implies S = 0 when θ = 0. Hence (1.1) and (1.2) gives

$$S = W^2 \sin\theta \ , \ T = -W^2 \cos\theta \qquad 1.6$$

where W is a constant.

Equations (1.2), (1.4) and (1.6) give

$$\frac{d^2\theta}{ds^2} = -W^2 \sin\theta$$

which on integration yields

$$\left(\frac{d\theta}{ds}\right)^2 = 2W^2 \left[\cos\theta + c\right], \quad c \text{ constant}$$

By writing $c = -\cos\alpha$ this result becomes

$$\left(\frac{d\theta}{ds}\right)^2 = 4W^2 \left[\sin^2 \tfrac{1}{2}\alpha - \sin^2 \tfrac{1}{2}\theta\right]$$

ie:

$$M = \frac{d\theta}{ds} = \pm 2W \left[\sin^2 \tfrac{1}{2}\alpha - \sin^2 \tfrac{1}{2}\theta\right]^{\frac{1}{2}} \qquad 1.7$$

where the choice of sign depends on the region, as indicated on the diagram

$$\text{I} \qquad 0 \leq x \leq \tfrac{1}{4}, \qquad \frac{d\theta}{ds} = 2W\left[\sin^2\tfrac{1}{2}\alpha - \sin^2\tfrac{1}{2}\theta\right]^{\frac{1}{2}}$$

$$\text{II} \qquad \tfrac{1}{4} \leq x \leq \tfrac{3}{4}, \qquad \frac{d\theta}{ds} = -2W\left[\sin^2\tfrac{1}{2}\alpha - \sin^2\tfrac{1}{2}\theta\right]^{\frac{1}{2}}$$

$$\text{III} \qquad \tfrac{3}{4} \leq x \leq 1, \qquad \frac{d\theta}{ds} = 2W\left[\sin^2\tfrac{1}{2}\alpha - \sin^2\tfrac{1}{2}\theta\right]^{\frac{1}{2}}$$

The symmetry of the solution enables the boundary conditions to be modified so that

$$x = \tfrac{1}{4}, \qquad s = \tfrac{1}{4} \qquad \text{when } \theta = \alpha$$

Using (1.7) it is now possible to obtain two equations which determine W and α, namely

$$\tfrac{1}{2}W = \int_0^\alpha \frac{d\theta}{\left[\sin^2\tfrac{1}{2}\alpha - \sin^2\tfrac{1}{2}\theta\right]^{\frac{1}{2}}}$$

and

$$\frac{WD}{2L} = \int_0^\alpha \frac{\cos\theta\, d\theta}{\left[\sin^2\tfrac{1}{2}\alpha - \sin^2\tfrac{1}{2}\theta\right]^{\frac{1}{2}}}$$

The integrals in these equations are elliptic integrals and can be evaluated by using tables. An approximate solution can be found by expanding the integrals in power series involving $\sin\tfrac{1}{2}\alpha$ which is less than unity. Such a method gives

$$\frac{W}{2\pi} = 1 + \tfrac{1}{4}\beta + \tfrac{9}{64}\beta^2$$

$$\frac{WD}{2\pi L} = 1 - \tfrac{3}{4}\beta - \tfrac{15}{64}\beta^2 \qquad \text{where } \sin^2\tfrac{1}{2}\alpha = \beta \ll 1 \qquad 1.8$$

A first approximation can be found by neglecting β^2 and solving (1.8) to give

$$W = \frac{8\pi}{(3 + \frac{D}{L})} \qquad 1.9$$

$$\beta = \sin^2\tfrac{1}{2}\alpha = 4\left(\frac{1 - \frac{D}{L}}{3 + \frac{D}{L}}\right)$$

Now that W and $\sin\frac{1}{2}\alpha$ have been calculated we can transform back to physical quantities S^*, T^*, M^*.

$$S^* = \frac{BW^2}{L^2}\sin\theta^*$$

$$T^* = \frac{-BW^2}{L^2}\cos\theta^*$$

$$M^* = \pm\frac{2WB}{L}\left[\sin^2\frac{1}{2}\alpha - \sin^2\frac{1}{2}\theta^*\right]^{\frac{1}{2}} \qquad 1.10$$

The extreme values of these loads can now be examined. For convenience the *'s will now be dropped.

Shearing force S:-

This varies from Zero at the two end points and the mid point to a maximum value when $\theta = \pm\alpha$

$$S_{max} = \frac{BW^2}{L^2}\sin\alpha$$

Using (1.9) this can be written

$$S_{max} = \frac{256\pi^2 B}{L^2}\;\frac{\left\{(1 - \frac{D}{L})(5\frac{D}{L} - 1)\right\}^{\frac{1}{2}}}{(3 + \frac{D}{L})^3} \qquad 1.11$$

Tension T:-

From (1.10) the tension is always negative so the hose is in compression at each point along its length. The extreme values of this tension are

$$T_{max} = -\frac{BW^2}{L^2}, \qquad T_{min} = -\frac{BW^2}{L^2}\cos\alpha$$

Using (1.9) these are

$$T_{max} = \frac{-64\pi^2 B}{L^2(3 + \frac{D}{L})^2}, \qquad T_{min} = -\frac{64\pi^2 B(9\frac{D}{L} - 5)}{L^2(\frac{D}{L} + 3)^3} \qquad 1.12$$

Bending moment M:-

The bending moment M varies from a minimum of zero at the points where $\theta = \pm \alpha$, to a maximum value at the end points and the mid point.

$$M_{max} = \frac{2WB}{L} \sin \frac{1}{2}\alpha$$

Using (1.9)

$$M_{max} = \frac{32\pi B(1 - \frac{D}{L})^{\frac{1}{2}}}{L\,(3 + \frac{D}{L})^{\frac{3}{2}}} \qquad 1.13$$

These extreme values can be found to a greater degree of accuracy by using second approximations for W and sin α which are obtainable from (1.8).

While this model is a very simple one from the point of view of practical hose installations, it does give some indications of the features that may occur. In the simple situation where the hose is loaded because of its position alone there are bending moments which could cause problems. In this situation the tension is of little significance because it is predicted that the hose would in fact be in compression and failure due to tension is probably unlikely.

This mathematical model is far too simple for the practical situation but a more suitable model is outlined below.

There would be obvious advantages in making M vanish at the sea-bed manifold by adjustment of the end conditions. A prediction of the angle of clamping at the sea-bed to achieve this situation would be of importance.

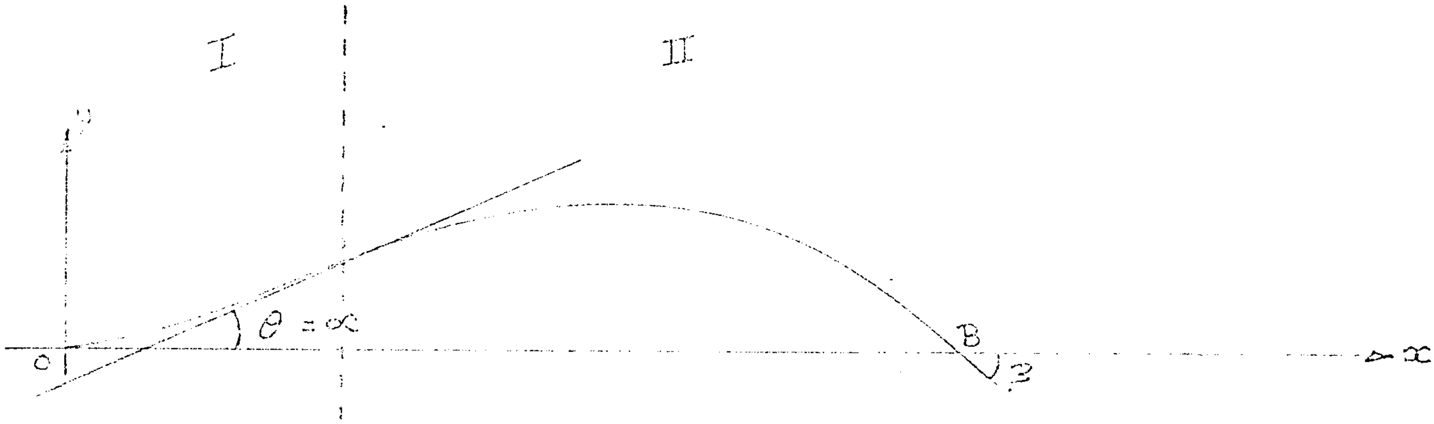


FIGURE 3

Region I	$\frac{d\theta}{ds} > 0.$	At A	$\theta = \alpha,\ \frac{d\theta}{ds} = 0$
Region II	$\frac{d\theta}{ds} < 0.$	at B	$\theta = \beta,\ \frac{d\theta}{ds} = 0$

For this problem the equations remain the same as in the previous analysis but the boundary conditions become:-

when $s = 0,\ x = 0,\ y = 0,\ \theta = 0$

when $s = L,\ x = D,\ y = 0,\ \frac{d\theta}{ds} = 0\ \ (\triangleright\theta = \beta)$

However, the changing of the second boundary condition makes the analysis extremely complicated and no solution can be obtained without some numerical work. An easier approach would be to solve this problem by numerical techniques only, this method is shown in Appendix (9).

From appendix (9) it has been shown that the simplified problem outlined can be solved using a numerical technique. Further from the results it has been shown that:-

a) A suitable fixation angle at the sea bed (Plem manifold angle) can be established corresponding to zero bending moment at that point.

b) The configuration resulting and angle above varies only slightly with "body force". The latter being a dimensionless quantity $F = \frac{WL^3}{B}$ where W = Total nett hose weight

L = total nett hose length

B = hose bending stiffness

(NB: The latter has been demonstrated via model tests whereby the addition of substance amounts of buoyancy (ie addition of submarine floats) has produced little change in configuration of the system under study).

c) Changes in both the neutral fixation angle and magnitude of bending moments were found to be greatly dependent upon variations in the ratio of sea depth to hose length. This obviously is important in considering the extreme survival motion states of a buoy system, (maximum wave heights).

d) Maximum bending moments (in opposing senses) were found to occur at the buoy manifold and at a point approximately two thirds of the distance down the hose string.

Whilst the mathematical model constructed is obviously a simplification of a real system it does serve as a useful tool in the initial design of a system. That is a study of a proposed system at mean water level and other "assumed static" states in order to initially evaluate ideal hose length, manifold angle(s), buoyancy distribution etc.

Comparisons with measurements taken from model tests, ie: loads and configuration, have given favourable results.

Further work currently being undertaken includes the introduction of:-

1) Buoy Excursion

2) Drag (current) forces

3) Three dimensional solution

4) Dynamic solution

The latter two areas are considered to be extremely complex if any meaningful results are to be obtained. It may therefore prove to be uneconomic to pursue such a complex study at this stage and to concentrate on physical model testing.

Figure (70) shows a predicted submarine configuration, superimposed upon which is a measured configuration (model tests) for similar conditions.

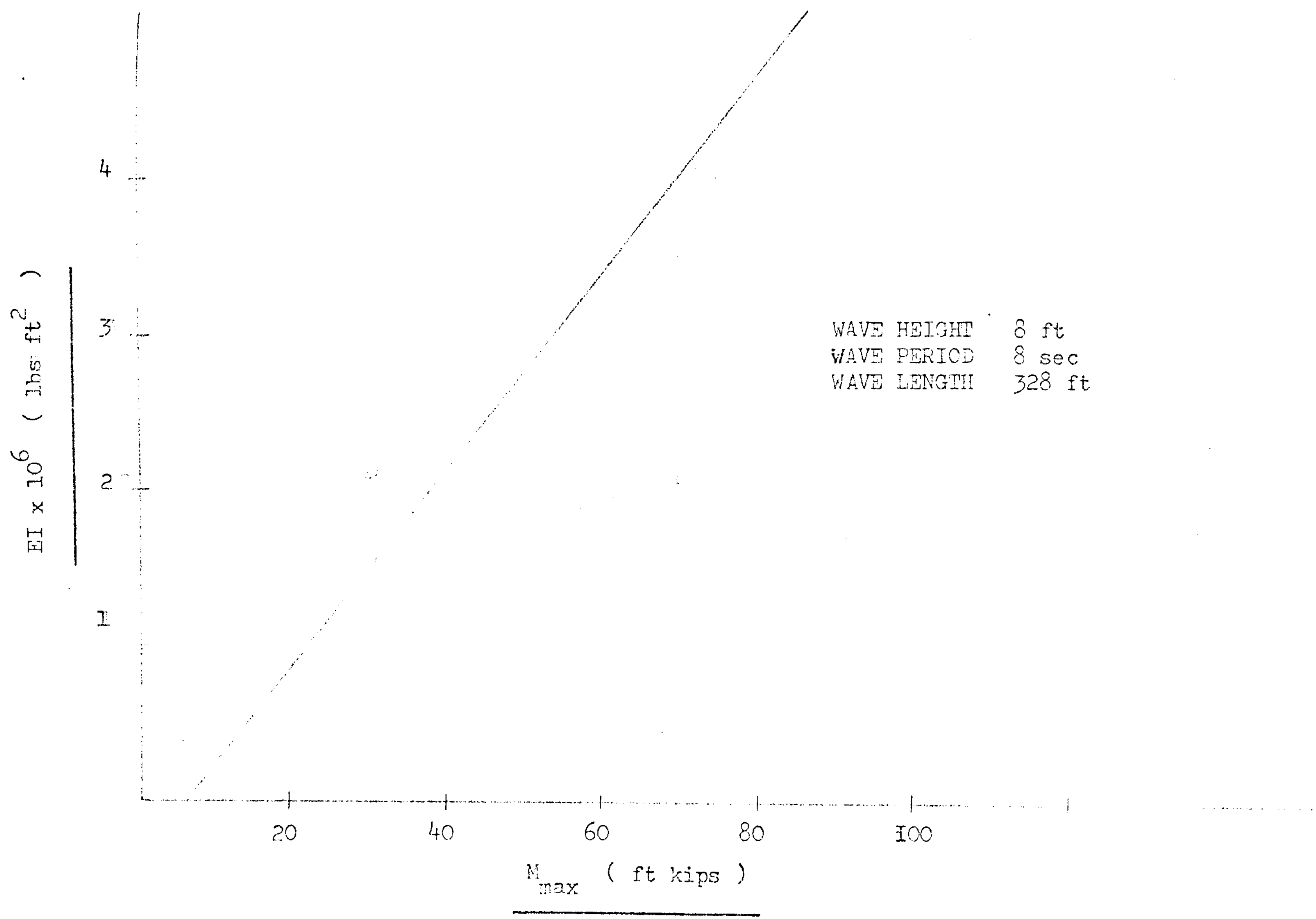


FIGURE 67 - MAXIMUM MOMENT VERSUS STIFFNESS (FIRST OFF THE BUOY HOSE)

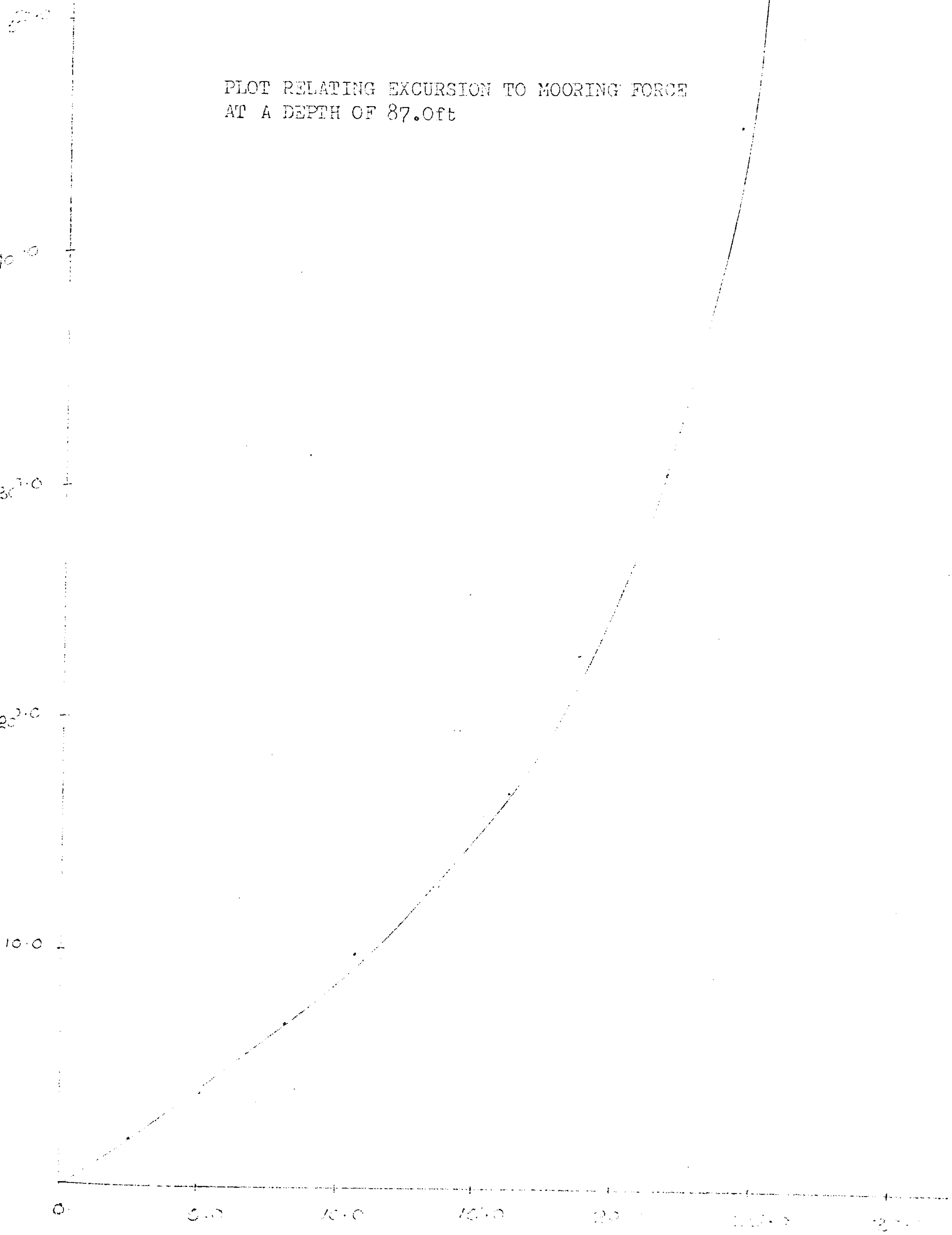

PLOT RELATING EXCURSION TO MOORING FORCE
AT A DEPTH OF 87.0ft
10.0
0

FIGURE 68

PLOT OF MAXIMUM EXCURSION AS THE MOORING

FORCE CIRCLES THE BUOY :

(WATER DEPTH 99.0 ft.)

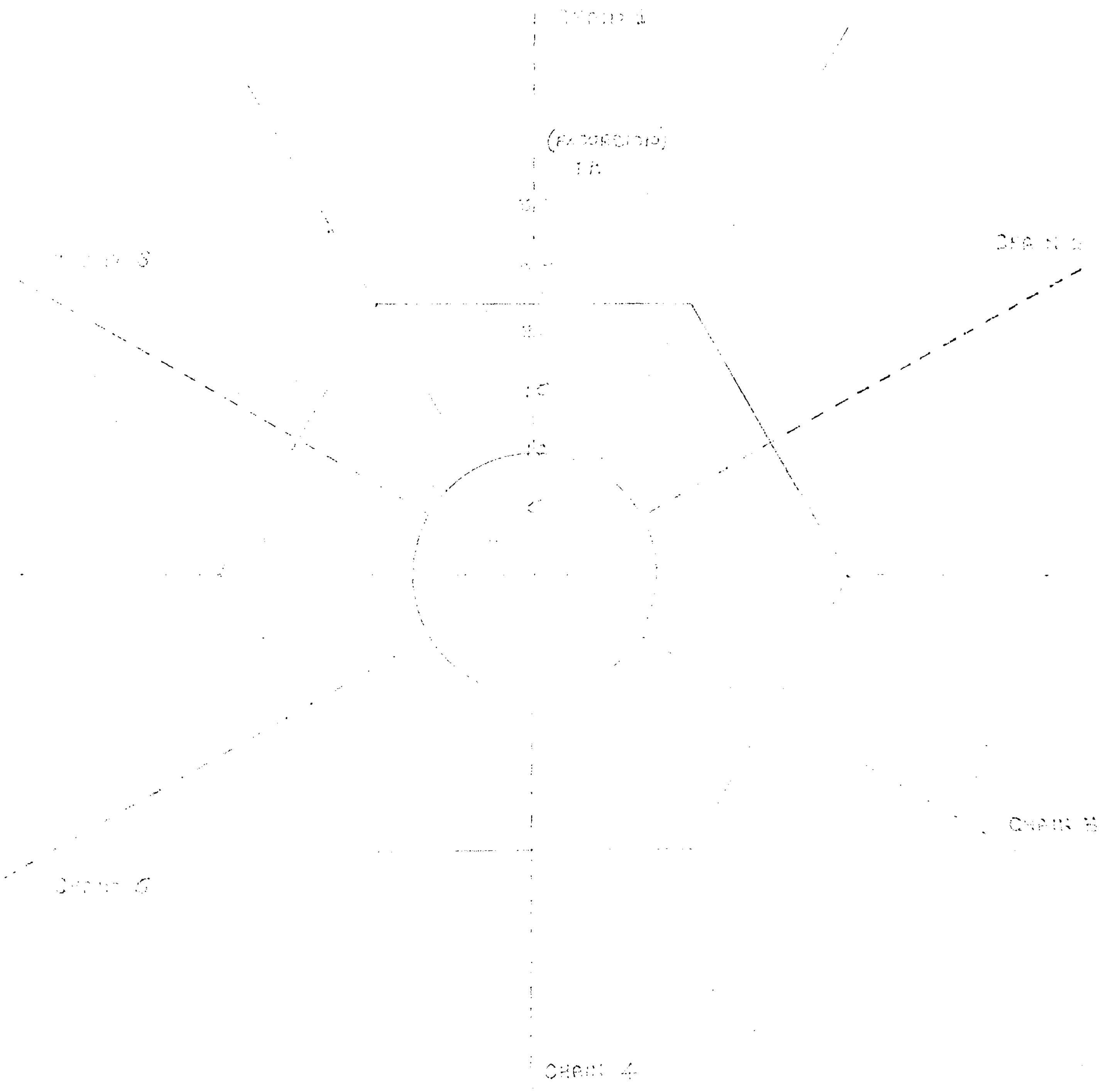


FIGURE (69)

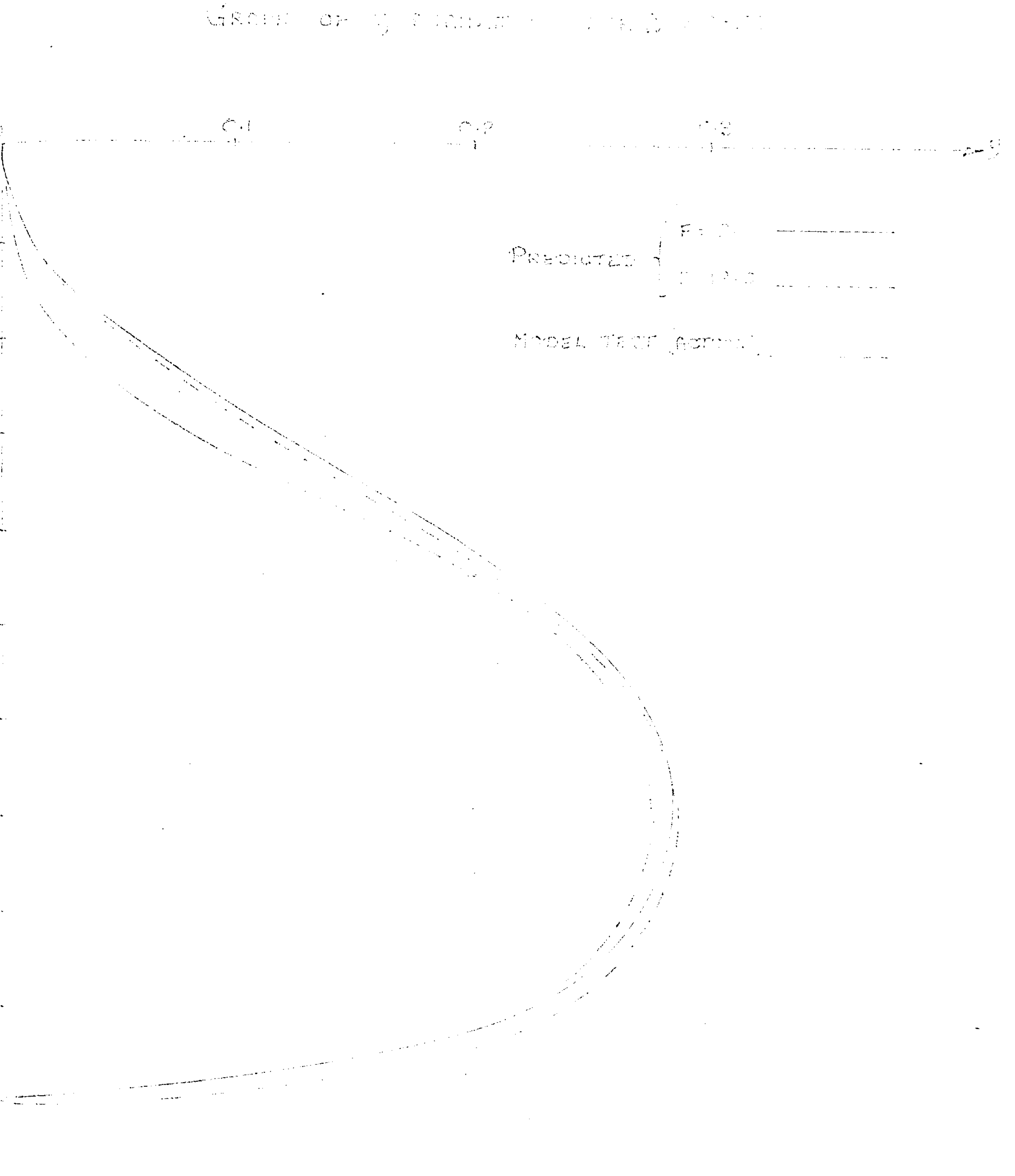


FIGURE 70

VIII CHARACTERISTIC PROPERTY TESTING

INTRODUCTION

In essence, a hose consists of no more than four principle components in one permutation or another; the lining, the reinforcing plies, some form of hoop reinforcement and a final outer cover. It is,however, this very "simplicity" which makes the design of a hose very complex for the only two components which affect its physical behaviour or properties (the reinforcing plies and hoop reinforcement) must account for:-

(1)	Burst strength		
(2)	Crush/Kink resistance		
(3)	Bending Properties	)	Bending stiffness
		)	Minimum Bending Radius
(4)	Axial Properties	)	Axial Stiffness
		)	Tensile strength
(5)	Torsional Properties	)	Torsional stiffness
		)	Torsional strength

Traditionally hose has been designed on the basis of the first two properties listed, ie: burst strength and the ability to withstand external loading. Little or no attention was paid to the latter properties, not even those values resulting from the primary design criteria let alone attempting to design for these latter properties. It was thus considered appropriate to attempt to define and hopefully classify some of these properties in order to gain a fuller understanding of hose design. Additionally the historical method of designing hose purely as a pressure container clearly was not adequate for products intended to function as "structural elements" in a marine environment.

VIII.1 BENDING CHARACTERISTICS

Bending stiffness is one of the most variable and imperfectly understood properties of hose behaviour. Such properties are however extremely important, particularly so in the design of underwater systems, where the hose layout or configuration is mainly defined by such stiffness properties.

Initial test work in this area, (prior to the involvement of the writer) was very "basic" and of little value. Figure (121) illustrates the test method employed. - Basically the hose was clamped at one end and a pulling force applied at the other, this end being lifted clear from the ground by means of a rolling crane. The pulling force was measured as was the deflected shape by means of triangulation between two fixed points on the floor and marked intervals on the hose surface. - Based on the test results a graphical method was employed in order to calculate the radius of curvature and bending moment, stiffness was then calculated, (usually a single value), using the relationship :- $M/I = E/R$.

The above method however proved inadequate in a number of ways:-

(a) Method of test/evaluation were very crude and resulted in considerable inaccuracies.

(b) Test method involved the use of "offsite" mobile cranes/personnel.

(c) No tests were performed with the hose in a pressurised state.

A revised test/evaluation method was thus evolved by the writer to overcome these problems and to extend the scope of the tests conducted.

The general bending principle employed is that of an eccentrically loaded strut (refer to Figure (122)).

(a) Both ends of the hose under test are sealed with blind flanges and the hose filled with fresh water.

(b) A structural member lever arm is bolted to each blind flange.

(c) A steel cable is attached to each lever arm attachment point with a load measuring device (load cell) and tensioning device inserted between attachment points.

(d) The hose is supported, with the hose axis and lever arms in a horizontal position, on a number of roller bearing trolleys. (These trolleys are equipped with smooth rimmed steel wheel castors and run on a smooth concrete surface in order to minimise frictional effects).

(e) The steel cable is tensioned until the hose bends into a segment arc. At this stage load cell readings, P, are recorded for various internal pressures ranging from 0 lb/in^2 to 275 lb/in^2. In addition the distance, Y_o, between the steel cable and the initial hose centre line is determined, and the deflected curve defined by a series of deflection measurements, Y, at equal intervals along the wire axis.

(f) The test described by (c) is repeated for various segment arc deflections until a minimum bend radius is achieved.
(NB: Load readings are recorded in both increasing and decreasing curvature directions and the results averaged in order to eliminate any residual frictional effects of the support trolleys).
In order to calculate the bending moment, radius of curvature and hence bending stiffness from the results obtained it is first necessary to specify various properties of the deflection curve mathematically. The data obtained from the test consists of a set of co-ordinates (X,Y) giving the shape of the bent hose at a set of N points equally spaced along the axis, so that the n^{th} co-ordinate is:-

$$X_n = n^h \qquad o \leq n \leq N$$

h being the distance between the points, and the corresponding Y value is:-

$$Y = Y_n$$

then the derivatives can be calculated by a simple finite difference formula as follows:-

$$Y_n^1 = \frac{Y_{n+1} - Y_{n-1}}{2h} \qquad 1 \leq n \leq N-1$$

and

$$Y_n^{11} = Y_{n+1}^1 - Y_{n-1}^1 \qquad 2 \leq n \leq N-2$$

or alternatively:-

$$Y_n^{11} = \frac{Y_{n+2} - 2Y_n + Y_{n-2}}{4h^2}$$

Thus the radius of curvature may be defined:-

$$R = \frac{(1 + Y^{12})^{3/2}}{Y^{11}}$$

(NB: It is not sufficient to define R by $R = \frac{1}{Y^{11}}$ as 'large' deflections are involved).

Naturally the accuracy of the above depends on the accuracy of the measured values Y_n and the size of interval h.

The bending moment M is calculated directly by:-

$M_n = T_{(p)}Y_n$ where $T_{(p)}$ is the wire tension which is a function of internal pressure.

Thus the bending stiffness may be calculated from the relationship:-

$$EI = MR$$

Typical results of such tests are shown by Figures (123) and (124). Test results show the stiffness to vary not merely along the hose length (ie: varying stiffness resulting from additional reinforcement at points of discontinuity such as fittings) but also with varying with internal pressure and degree of curvature.

ie: $EI \propto (p), (R)$

All tests conducted for differing hose sizes and constructions exhibit such relationships and clearly illustrate the fatuity of quoting single value stiffness values for hose as previously done.

Further work has shown that the multiplicity of curves (Figure 124) relating to varying pressures may be approximated to a single straight line of intercept A and slope B if stiffness is plotted against the quantity $\frac{R}{p}^{\frac{1}{2}}$ where R = radius of curvature

p = internal pressure

(Figure (125)).

The constants A and B are unique and indicative of the hose in question.

In addition to the previous relationships it was also determined that hose stiffness varies significantly with age, by factors as great as two or three. (Figure 124). Work investigating this phenomenon is as yet far from complete, however initial results would suggest that the rate of decline in stiffness is most rapid in the early stages of use and thereafter decays exponentially.

It is not surprising, in view of the complex behaviour pattern determined, that it has not proved possible to develop a satisfactory theory to explain or predict such behaviour. In particular, the reasons for the very significant changes which occur during the initial stages of working life are not at all clear. Nevertheless, it is possible to identify some general proportionalities and suggest the causes behind some of the observations.

It would not be unreasonable to assume that stiffness is generally proportional to a combined modulii for the hose construction. If we simplify this complex problem to a situation whereby two modulii are considered as representative of the main reinforcing plies: E_1 and E_2 representing the cords and rubber matrix respectively. The latter is extremely difficult to control or specify as it depends upon:-

CHARACTERISTICS OF THE COMPOUND

CURING PROCESS

SHAPE ADOPTED

EFFECTIVENESS WITH WHICH IT IS CONTAINED.

The latter two being stress dependent, thus variable. Additionally, the value of E_1 for the modulus of the cords is not a constant, but variable dependent upon cord elongation.

The effect of pressure in relation to hose stiffness is dealt with elsewhere in this thesis, (Chapter IX.2), however let us briefly consider the stiffness/curvature relationships observed. Most cords, whether wire or fabric used in hose constructions have variable properties which, as previously stated, depend upon elongation. A typical graph of which might be as shown below:-

+ Tensile

- Compressive

It may readily be seen that the compressive capability of such cords is very limited before buckling occurs. It is this feature coupled with the geometry of the hose that may be used in order to explain decreasing stiffness with increasing curvature.

"Under low levels of curvature, cords under both compressive and tensile loading exhibit similar load bearing properties resulting in a symmetrical stress distribution and the achievement of maximum stiffness. However as the curvature is increased the cords under compression progressively collapse (depend on position) thereby transferring the compressive stress to the matrix material and the remaining plies, thereby radically altering the "effective section" of the hose together with the position of the neutral axis. Eventually a situation is reached whereby the total compressive stress is carried by the matrix material rather than the cords".

Naturally pressurisation of the hose decreases the magnitude of the compressive stresses for any given degree of curvature, hence delays the onset of localised buckling of the cords and thereby increases the effective stiffness.

Work is continuing in this area, both theoretically and practically, in order to gain a greater understanding of these effects. In the latter case, cords (Kevlar) are currently being evaluated which have a greater compressive loading capability, than those currently employed.

VIII.2 TENSILE PROPERTIES

Tensile or axial stiffness is a property which has not historically been considered either as a design parameter or even as a resultant property. As such no information was initially available to the writer. An initial appraisal of the property, and how it was achieved by a hose type construction immediately suggested that it was an extremely complex property dependent upon a wide variety of variables. These variables were considered to include:-

HOSE DIAMETER

MAIN REINFORCING PLIES	(Material (properties), distribution, ((Angle of application, etc.
RUBBER MATRIX	(Material (properties), shape factors, ((adhesion levels, effectiveness of ((containment.
HELICAL REINFORCEMENT	(Size, number, spacing etc. ((of helical wires or alternate members.

INTERNAL PRESSURISATION

It may readily be seen that already so many combinations are available that in order to fully understand the property considerable test work would need to be conducted. A further complication arose, in that such tests would be expensive to conduct, in both time and materials. In order to overcome this latter problem, whilst a small number of tests were conducted over a range of hose sizes, (in order to gain an assessment of the orders of magnitude involved), tests designed to assess the contribution of the many variables were limited to short length, small bore samples. (Generally the samples used were 8in Ø x 15ft, NB: The seemingly long lengths were found necessary in order to overcome complications resulting from end effects due to the fittings and additional reinforcement of this area.

Axial stiffness is defined as:-

$$EA = N/\Delta L/_{L}$$

where N = Tensile load

L = Original Section length

ΔL = change in length

Figure (120) is a typical set of test results whereby the effect of varying the main reinforcement application angle (α) and the size of helical wire (d) were assessed.

N.B. The hoses tested as shown by the above figure consisted of two main plies of X2 @ EPI and two holding plies of 8376075, the variable parameters are shown below:-

Sample	Application angle (α)	helic wire dia (d)
GYP 11/75	45°	1'S
/76	45°	3'S
/77	35°	1'S
/78	35°	3'S
/79	50°	3'S
/80	50°	1'S

A number of features may be observed from this graph:-

1) Axial Stiffness increases with decreasing ply angle, i.e. GYP 11/77 (78) α = 35 are stiffer than other samples.

2) As the ply angle decreases; the effect of hoop reinforcement, (helical wire), on hose stiffness diminishes. It may be seen that at a ply angle of 35°, changes in the hoop reinforcement are little or no effect upon stiffness.

3) Conversely to (2) as the ply angle is increased, the hoop reinforcement takes on a more direct role in contributing to axial stiffness. N.B. It may be seen that for a ply angle of 45°, two stiffnesses are exhibited relating to helical wire diameter).

4) At low values of application angle (i.e. $\alpha \leq 40$) a single stiffness is exhibited, i.e. load deflection curve is of constant slope. For values of greater than 40 deg. there is an increasing tendancy of the similar curve to exhibit dual slopes, stiffness remaining at a relatively low value for initial extension thence achieving a higher constant valuefor greater elongations.

Similar test were conducted in order to evaluate the effect of hose diameter, various ply materials etc.

On the basis of such tests the following basic initial conclusions are drawn:-

1) The effect upon stiffness of hoop or helical reinforcement increases with increasing ply application angle, This may be explained by the relationship between application angle, pitch and wound diameter of the reinforcing components:-

$$P = \frac{\pi D}{\operatorname{Tan} \alpha}$$

For a given elongation under load there results a change in effective pitch, thus in order to restore equilibrium D and α must also change. By virtue of the tangent curve it may readily be assessed that the greater the angle α the greater is the change in D for a constant change in P.

In other words the tendancy for greater radical dilation exists for increasing ply angles. Since such movement is restricted by the helical or hoop reinforcement the latter naturally plays a significant role in restraining axial elongation, hence increasing tensile stiffness in such cases.

2. It has been found that for ply application angles of less than 40 deg. (i.e. $\alpha \leq 35$) stiffness is effectively independent of both helical reinforcement and rubber matrix stiffness, (for extensions considered, i.e. $\leq 10\%$)

In these cases it has been determined that stiffness is proportional to the hose diameter, ply properties and the ply application angle. This dependance may be expressed by the relationship:-

$$EA = f.D \cos^2 \emptyset \ F.n.N.$$

where

EA - Axial stiffness

D - hose diameter (Mean wound ply diameter)

$\emptyset$ - cord application angle

F - Force/unit extension of individual cords

n - N° of cords/ply

N - N° plies

f - empirical factor relating to the type of reinforcement material employed.

The above formula has resulted in reasonable correlation with test results.

3) For cord application angle of greater than 35 deg., rubber matrix stiffness and helical reinforcement properties predominate. (In fact the properties of the ply material have little or no effect assuming they are of sufficient strength to transfer the resulting "strangulation" or collapsing stress onto the helical reinforcement). In this case, initial deformation occurs, in the main, within the rubber matrix. Since the "around diameter" of the plies is restrained by the hoop reinforcement initial hose elongation is accommodated by an alteration of the ply angle. Under these conditions which exhibit a shallow slope on the load/deflection curve, the following relationship exists:-

$$EA = \frac{Pf\ D^2}{\mathrm{Tan}^2 \emptyset}$$

where - Pf - Stiffness property of the rubber matrix. (NB This is extremely difficult to predict because of its dependance on stress, shape factors, fatigue ageing etc).

other symbols as previously defined.

After initial extensions as defined by the above, the properties of the helix predominate resulting in an approximate relationship:-

$$EA = \frac{f_H\ S_H\ D^2}{D_H^3\ P_H\ \mathrm{Tan}^2 \emptyset}$$

where S_H - Hoop or helix stiffness under crushing

D_H - Hoop or helix diameter

P_H - Hoop or helix pitch

f_H - Efficiency factor (const) depending on hoop material/construction.

other symbols as previous

The latter two formullii hypothesised prove difficult to use since:-

a) Transition from one to the other is difficult to accurately predict, also no method has proved satisfactory in predicting properties during the transition .

b) The factors P_f and f_H often prove difficult to evaluate particularly the former.

Never-the-less predictions obtained do generally correlate reasonably well with empirical results.

The final property that briefly has to be evaluated is the effect of internal pressurisation upon axial stiffness. Figure (127) shows a comparison between two twenty four inch bore hoses, stiffness plotted against internal pressure. (The samples being nominally identical apart from the ply application angle, this being 35 deg. and 50 deg. for curves (a) and (b) respectively).

In the case of (a) $[\alpha = 35°]$, it may be seen that stiffness initially increases with internal pressure but rapidly a constant value, at 50 lbf/in^2 which prevails over the working pressure range evaluated.

For case (b) $[\alpha = 50°]$, stiffness increases; from a lower initial value than the previous case, the increase continuing over a range of pressure values until it again approaches approximately the same constant value as in the previous case.

Whilst information with respect to the above is based on only the one experiment, whose results are presented, the following hypothesis is made:-

Internal pressurisation repidly stabilizes the matrix formation, controlling the structures ability to deform radially. The initial difference between the structures may be attributed to strangulation effect of the higher ply angle, (i.e. Tensile load is in effect translated as a strangulation or crashing pressure on the helix, until internal pressure reaches an equilibrium value the hose stiffness is still influenced by the factors previously described, however once an equilibrium state is achieved the pressure loading predominates.

The above situation may be described by the following:-

Cruching pressure due to end loading may be approximated as:

$$P = 2L \frac{\tan^2 \emptyset}{D^2}$$

Where P = Crushing pressure

L = End load

Other symbols as previously defined.

For the sample hoses in question

Sample (a)	45	35	26
Sample (b)	45	50	26
	L	Ø	D

Thus, P (a) = 46.5 lbf/in^2
P (b) = 134 Ø lbf/in^2

It may be seen from figure (I27) that the above values closely approximate the pressure at wich stiffness becomes a constant. Work is continuing in the above area in order to gain more information and to confirm that already obtained. Naturally progress is slow, because of sample and testing costs, however the information gained to date would certainly seem to warrent the continuation of such evaluation.

VIII.3 VOLUMETRIC DISTENSION

Because of the diverse nature of hose usage, it is frequently necessary to assess the life status of a hose and specify whether or not it remains suitable for further service usage. This is particularly important in situations where planned maintenance systems are employed. In such cases hoses are taken out of service on a twelve monthly or shorter period basis, inspected and then either re-installed for a further similar period or retired. Naturally the consequences of a wrong decision are high; on one hand the hose if returned to service may fail resulting in pollution, production losses, high 'change-out' costs etc, on the other hand the retirement of usable products would rapidly result in high revenue costs for what is an extremely high value item. It is obvious that the inspection criteria/test employed is critical.

The current test criteria, described by the buoy mooring forum hose guide (12), aimed at assessing future potential, (rather than immediate status for which standard pressure/vacuum tests are employed) is as follows:-

> "At test pressure - when the field test temporary or permanent elongation of a hose exceeds the factory test temporary or permanent elongation respectively by 2% of the overall length, the hose should be retired from service".

Over a period of time the above test has been shown to be totally meaningless, although a substantial number of operators still employ the method for want of a better method. By virtue of the various construction, alternatives briefly illustrated in the previous pages, elongation under pressure may be controlled by a number of factors, which respond to hose degradation, also some of which have no bearing on the subject.

A method was thus sought whereby degradation could be more readily assessed. Because of the variation in physical properties in relation to 'time in service' or fatique, it was initially considered that a property test, such as bending stiffness, might be a useful inspection tool. eg Test stiffness 40% Original Stiffness = hose retirement.

contd/..

However, on reflection, it was considered even if one could define the test method and the go/no-go value, (arbitrarily quoted as 40% above), the test itself would be difficult and time consuming to conduct. This would particularly be so under site conditions where equipment and trained personnel are not as readily available as might be considered. Thus the above test although suitable as a long term sampling technique and useful for R & D purposes, was not considered appropriate. Similarly other property testing methods were considered and likewise rejected.

It was almost by accident that whilst conducting tests in relation to pressure surge analysis a possible method came to light.

For the former work it was deemed necessary to establish typical hose pressure/volumetric distension curves for hose such that these values could be used in relation to surge pressure calculations.

It was whilst conducting such tests, that observations were made relating to the relationship between % elongation change and the similar volumetric change for a given pressure. (NB The tests were conducted by pressurising a hose to a given pressure, then releasing the pressure in defined increments by bleeding off fluid which could then be measured in terms of its volume).

Figure (128) indicates results for two hoses tested in the manner above. It may be seen that the relationship between % volume change and similar elongation change is non linear in the case of each hose, and that relationships do not apparently exist between the two hoses, (although they are of the same bore size).

From figure (114) - chapter XI.8 it has been shown that temporary elongation results do not reflect hose degradation. However, when a similar graph is plotted relating to temporary volumetric change, Figure (129), it may be seen that correlation does exist.

Whilst no firm conclusions may be based upon the results of a single test, it would initially appear that such measurements might well be of value in assessing the life status of hose. Naturally much work remains to be conducted, in conjunction with the test rig etc, in order to confirm the initial test results. Also to establish criteria correlating such a measurement to degree of degradation. The latter is only likely to result empirically following the testing of a significant sample of hoses.

However, bearing the above limitations in mind the technique would appear to offer advantages in terms of its ease of conduction, simplicity of method and the use of non specialist readily available equipment.

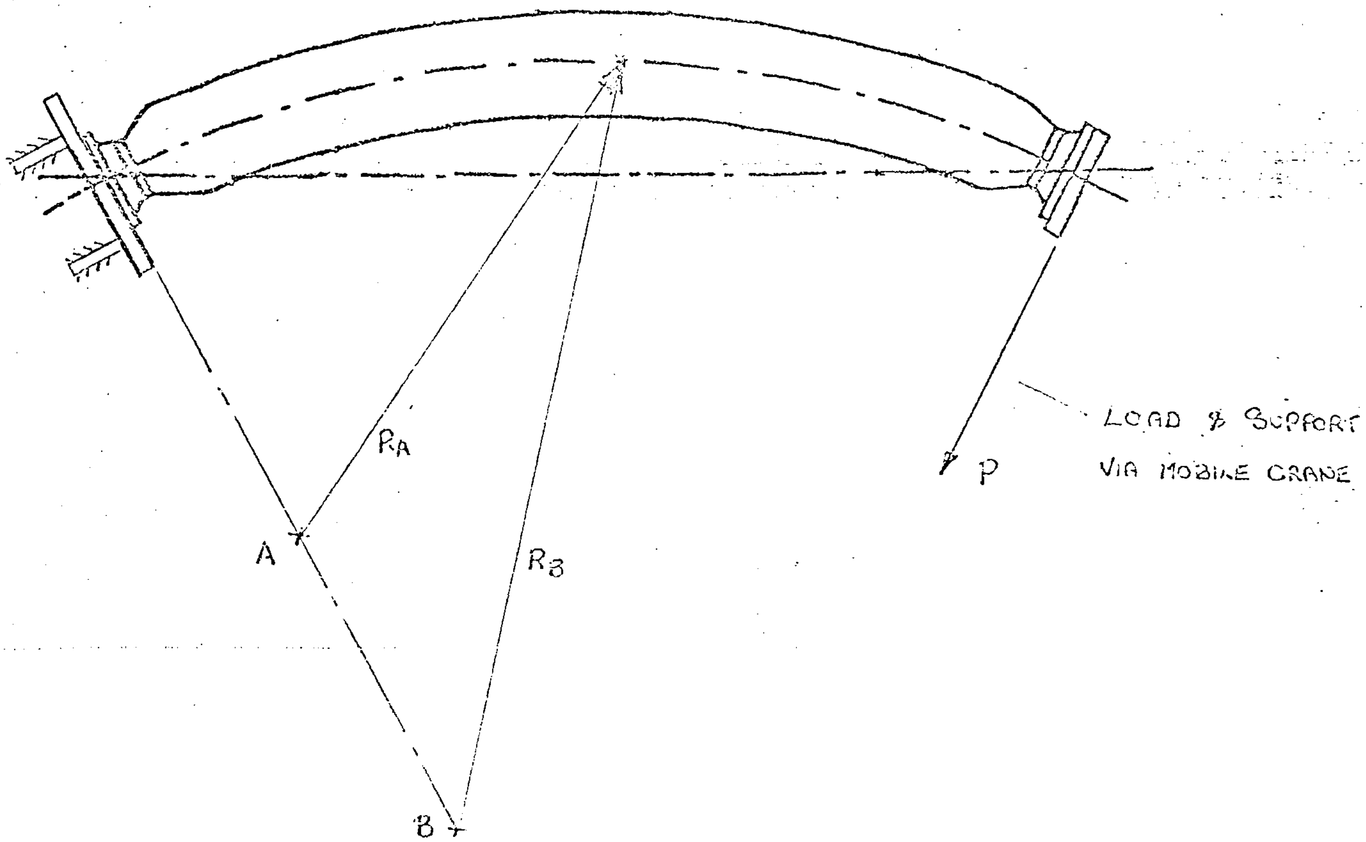


FIGURE (121) BEND TESTING (ORIGINAL METHOD)

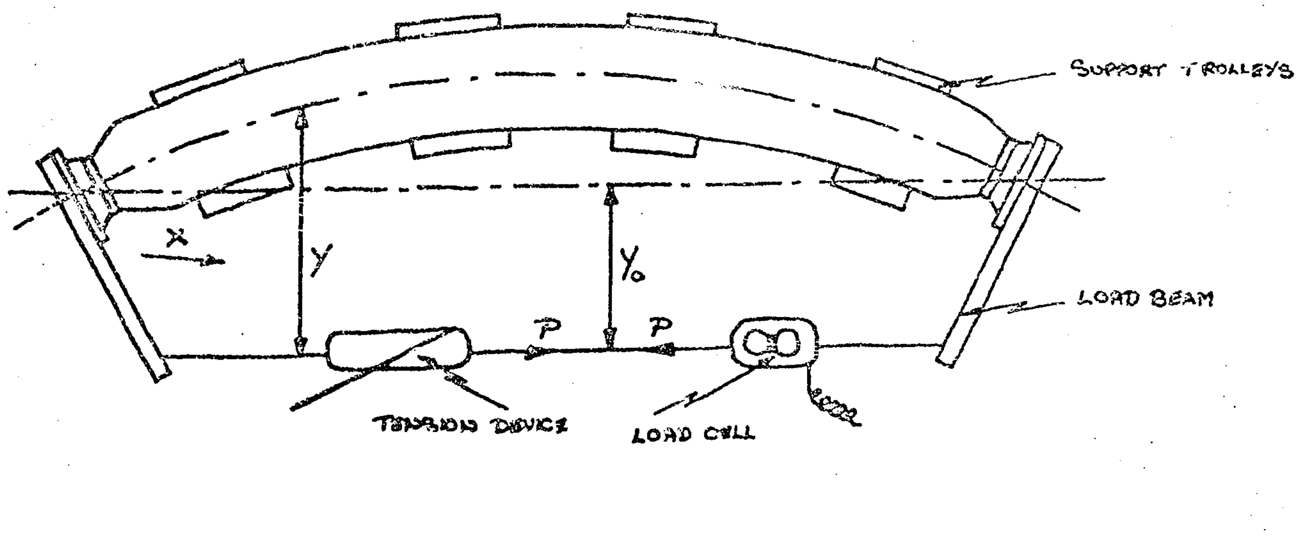


FIGURE (122) BEND TESTING (REVISED METHOD)

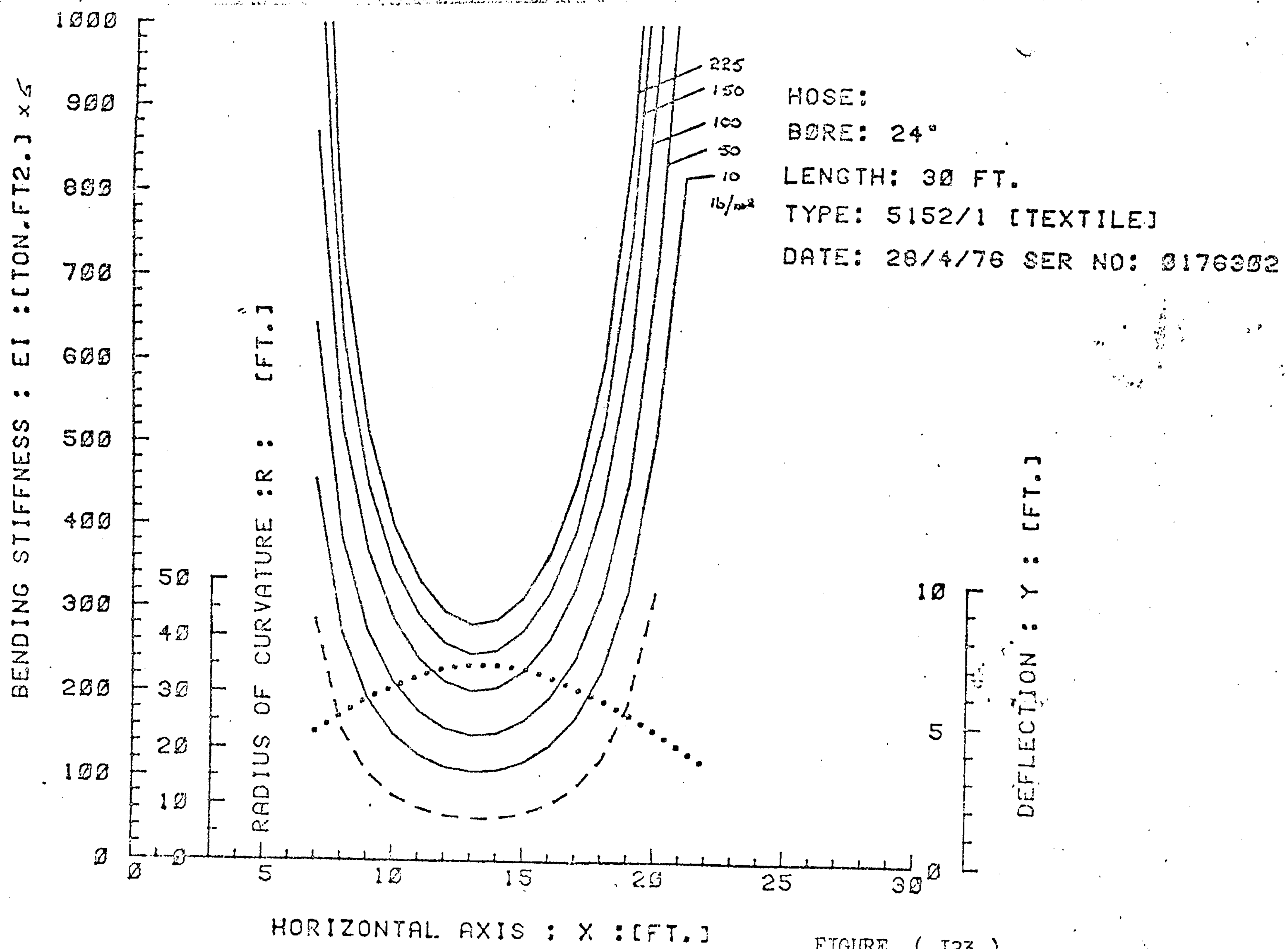

HOSE:
BØRE: 24"
LENGTH: 30 FT.
TYPE: 5152/1 [TEXTILE]
DATE: 28/4/76 SER NO: 0176302
225
150
100
50
10
lb/in²
BENDING STIFFNESS : EI :[TON.FT2.] x5
RADIUS OF CURVATURE :R : [FT.]
DEFLECTION : Y : [FT.]
HORIZONTAL AXIS : X :[FT.]
1000
900
800
700
600
500
400
300
200
100
0
50
40
30
20
10
0
5
10
15
20
25
30

FIGURE (I23)

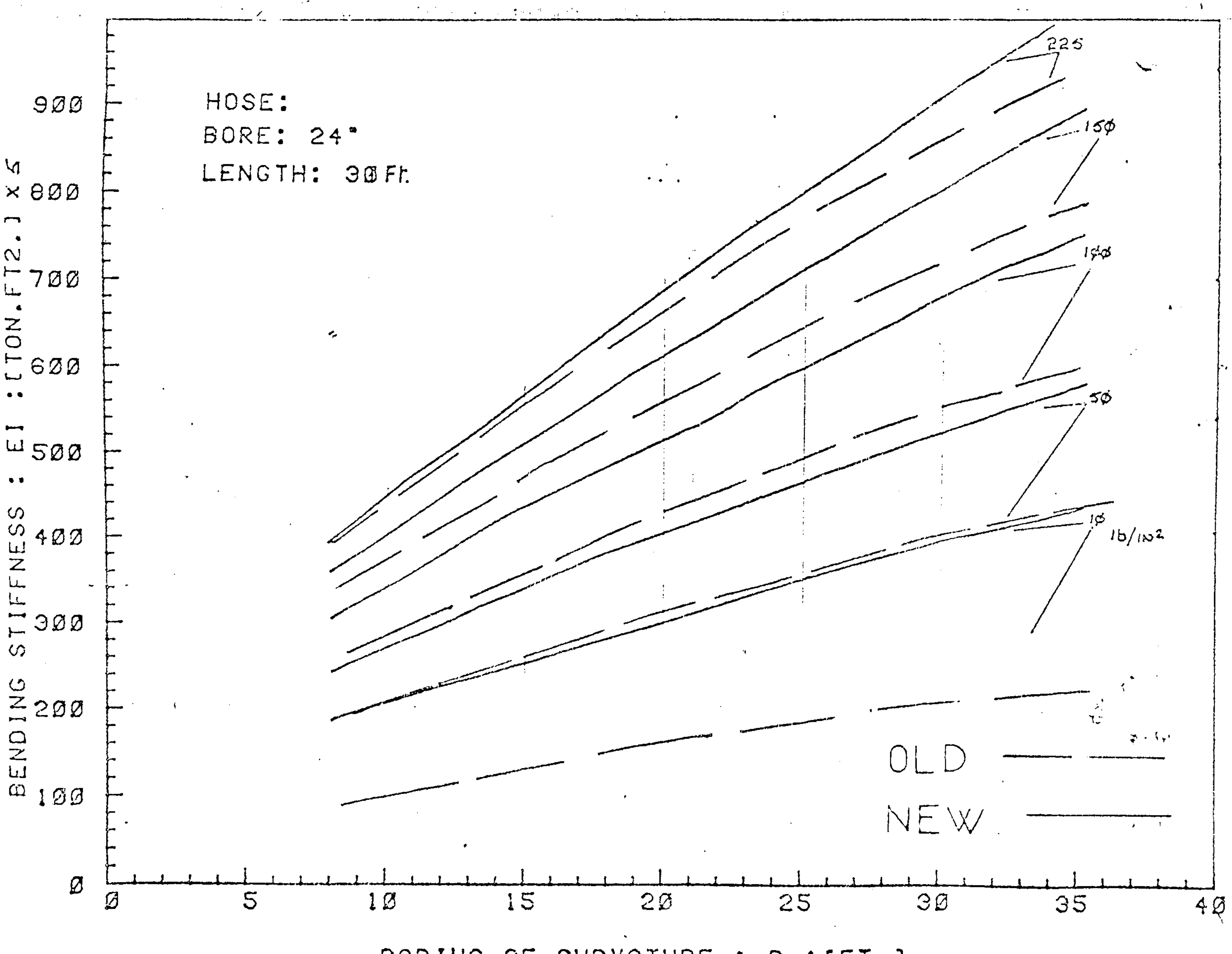

HOSE:
BORE: 24"
LENGTH: 30 Ft.
225
150
100
50
10
lb/in2
OLD
NEW
BENDING STIFFNESS : EI :[TON.FT2.] x 5
RADIUS OF CURVATURE : R :[FT.]
0
100
200
300
400
500
600
700
800
900
5
10
15
20
25
30
35
40

FIGURE (124)

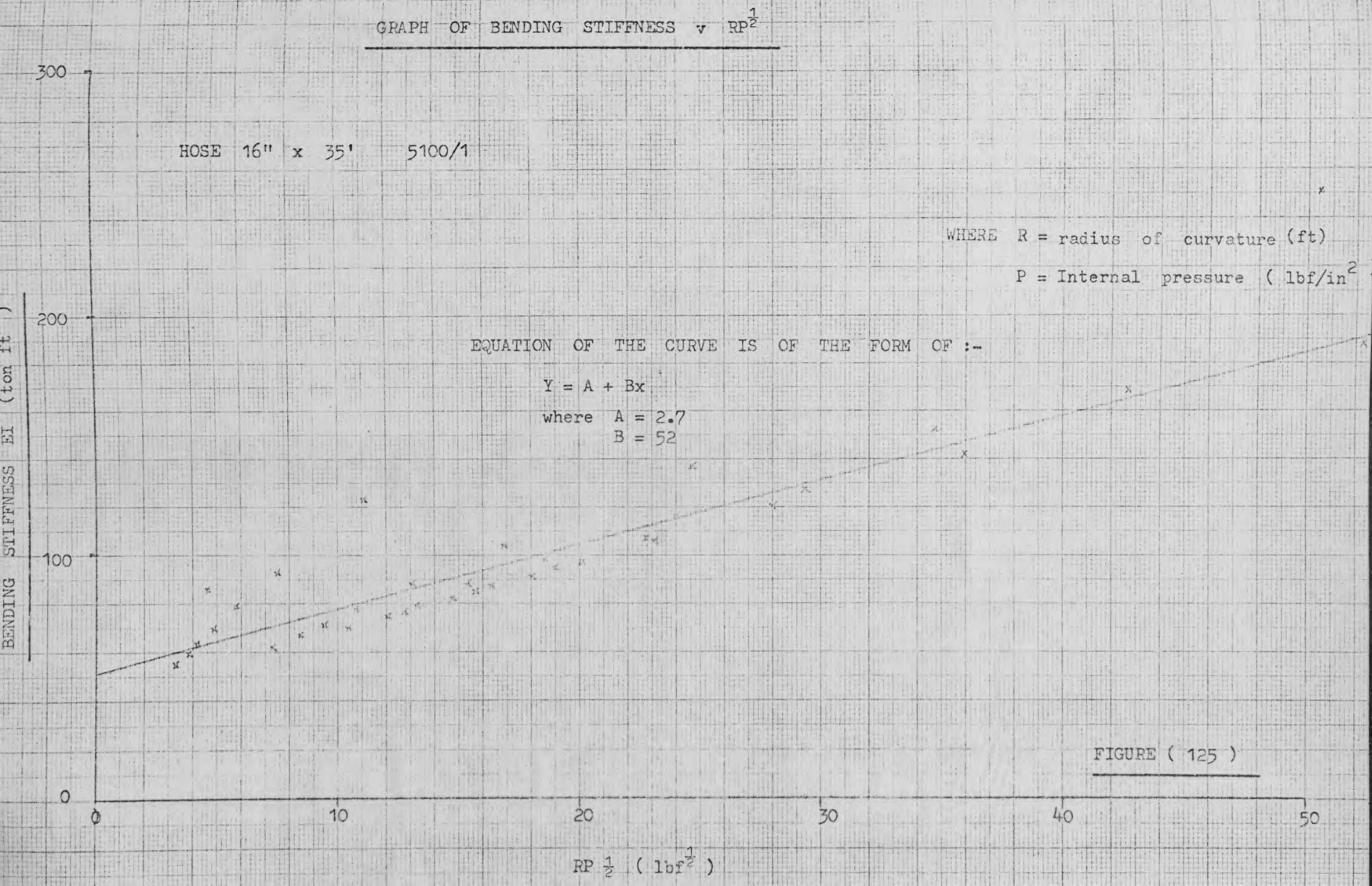

GRAPH OF BENDING STIFFNESS v $RP^{\frac{1}{2}}$
HOSE 16" x 35' 5100/1
WHERE R = radius of curvature (ft)
P = Internal pressure (lbf/in^2
EQUATION OF THE CURVE IS OF THE FORM OF :-
Y = A + Bx
where A = 2.7
B = 52
BENDING STIFFNESS EI (ton ft)
300
200
100
0
10
20
30
40
50
$RP^{\frac{1}{2}}$ ($lbf^{\frac{1}{2}}$)

FIGURE (125)

FIGURE (126) AXIAL STIFFNESS (E_A) v PLY ANGLE (α) & HELICAL WIRE DIA (d)

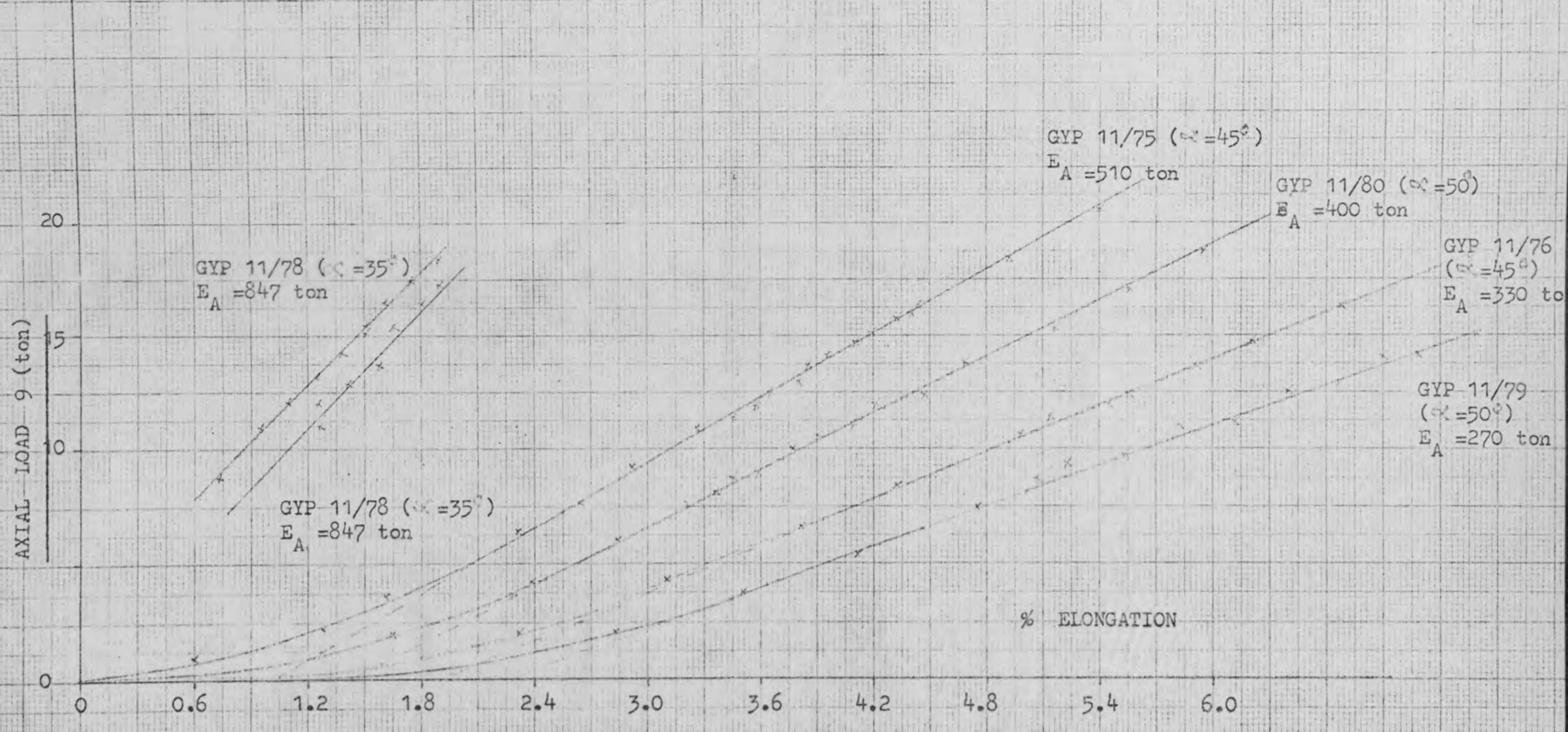


FIGURE (127) AXIAL STIFFNESS v INTERNAL PRESSURE

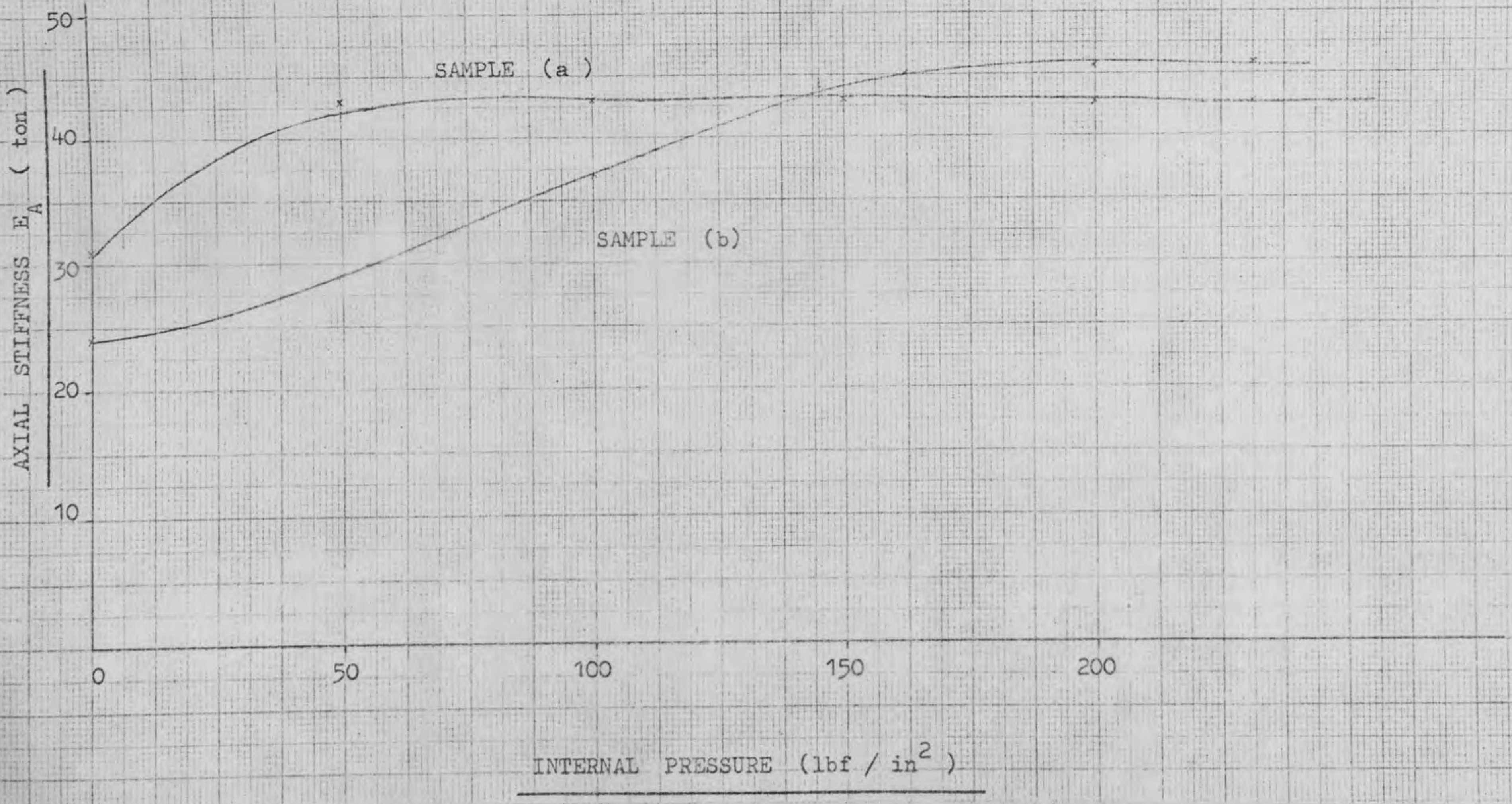


FIGURE (128) VOLUMETRIC DISTENSION & ELONGATION CURVES

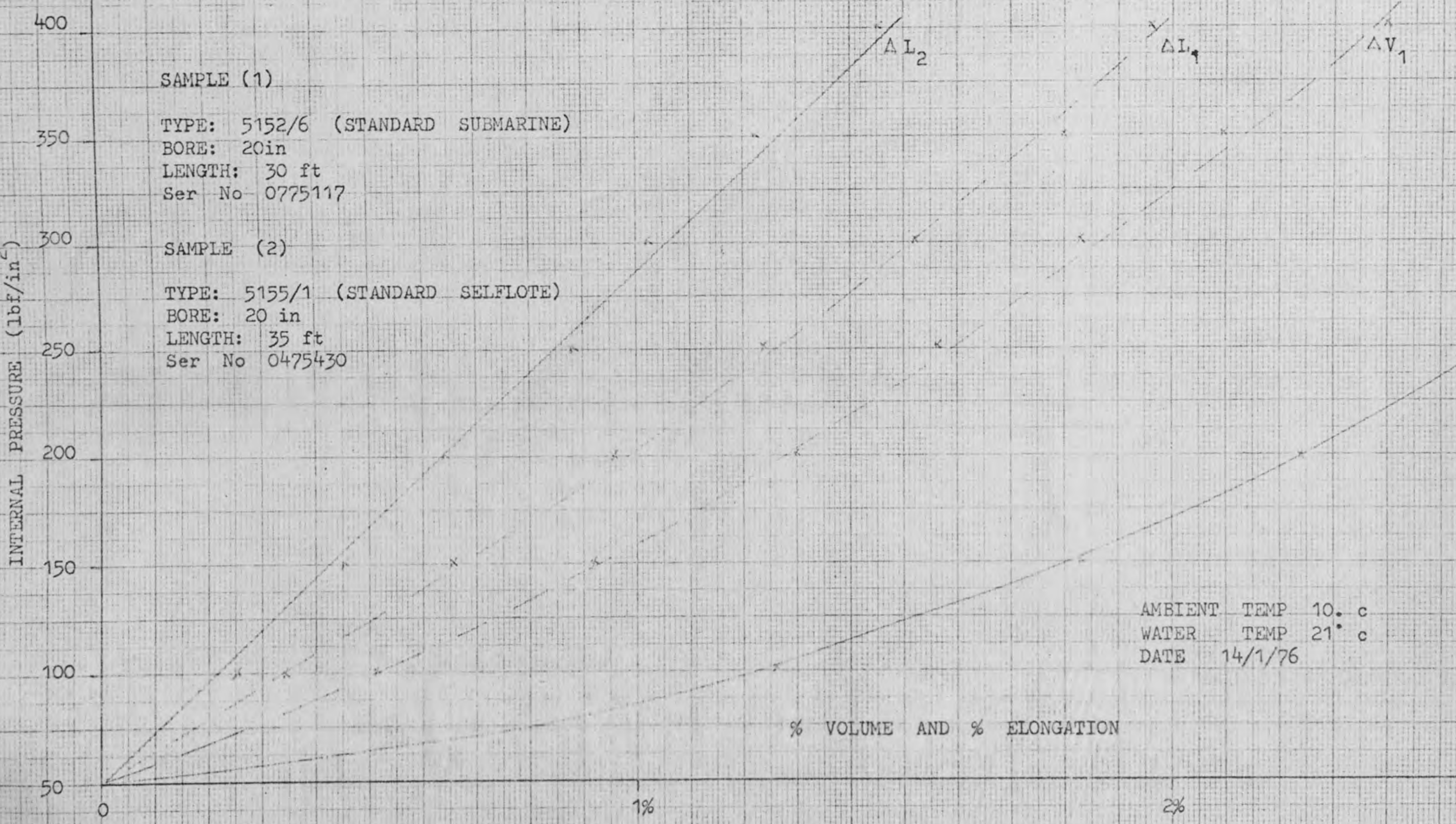
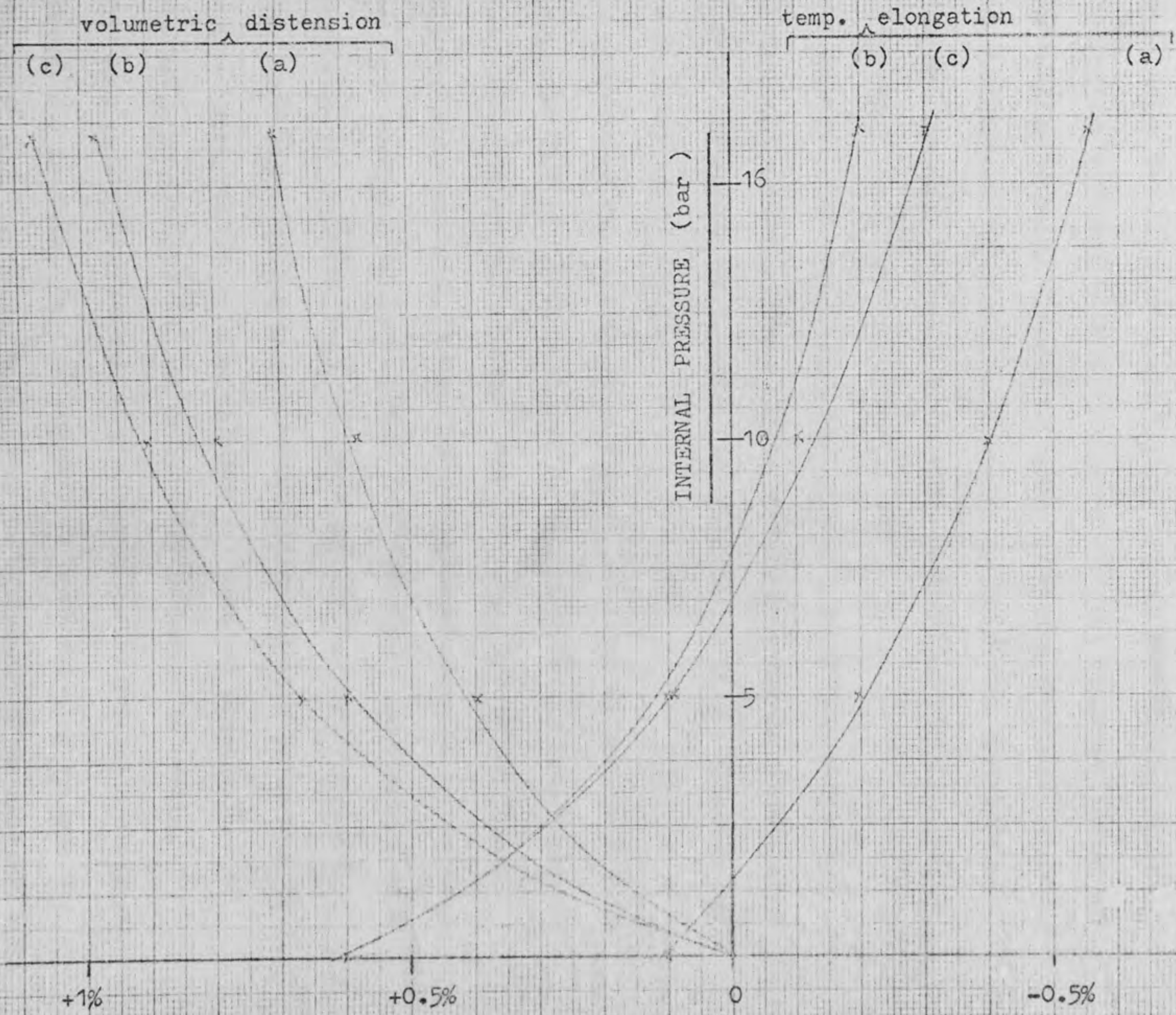


FIGURE (129) ELONGATION / VOLUMETRIC CHANGE v PRESSURE

(AFTER TESTING ON TEST RI



(a) AFTER 5000 CYCLES

(b) " 10000 CYCLES

(c) " 15000 CYCLES

% CHANGE IN LENGTH / VO

-1%

IX THEORETICAL ANALYSIS

IX.1 LARGE DEFLECTIONS OF BEAMS

During the conduction of characteristic bend testing, calculations in relation to design criteria for the Dynamic Test Rig and other theoretical studies; it became apparent that the classical theory of the bending of beams (limited to cases of small deflection), might be too restrictive. An alternate procedure was therefore developed to consider the equilibrium of beams whose deflections under the action of external forces are large.

The initial ideal case studied was that of a light beam supported on unrestrained frictionless pivots subject to a single concentrated load at its centre. It is shown, in appendix (5) that a solution can be obtained fairly readily.

In contrasting the results of both small and large deflection methods (appendix (5)) it may be seen that at low loads/deflections there is good agreement between theories, but under more severe conditions divergence is apparent. Further as greater divergence is encountered between theories, the large strain theory shows the development of a phenomena which can be termed geometrical instability, Figure (61). (NB: This behaviour has been demonstrated practically (Reference (2)), and is unrelated to the elastic limit and onset of plastic flow of the material of the beam, but is entirely geometric in origin.

The next extension attempted to the above, was to develop a similar theory for a uniformily distributed load, also detailed by Appendix (5). This case was chosen because of its relevance to the selected mode of operation in relation to the Dynamic Test Rig, (Refer to Chapter XI).

However, immediate difficulties were encountered in relation to defining the deflection curve at the correct stage of the calculation. The method finally developed was based upon dividing the beam into short lengths or elements and assuming that each was bent to a uniform radius of curvature.

The basic method for this solution and later solutions relating to more complex load combinations required an estimation of the inclination of the beam at the pivot, as input data. This is later modified by an iterative/interactive computer technique until a final solution is obtained.

In order to check the accuracy of the elemental method adopted it was applied to the case of the single concentrated load previously solved analytically. Agreement was found to be good.

It should also be noted that beyond the point of geometric instability, the element method does not converge to some value of α . (Inclination angle at the pivot), this too is taken to indicate the same instability thus indicating that the element method also represents reality.

To further test the method, both small and large theory were applied to documented test on a full size hose. The results of this comparison are indicated by Figure (62).

IX.2 BENDING STIFFNESS DUE TO INTERNAL PRESSURE

The bending stiffness of a hose depends upon the intrinsic stiffness of its structure and the internal pressure, (see test results VIII (a)). It was considered instructive to utilise the theory of inflated corded structures to calculate the effect of the latter, as a contribution, to understanding the stiffness properties of offshore hose.

Generally it was assumed that when a hose is bent its sectional shape remains unchanged, ie: circular. However it may readily be shown that such an assumption is invalid. For equilibrium the applied bending moment must be opposed by changes in the axial tension around the hose. If the circular section of a hose remains unchanged it may readily be shown that this tension remains unchanged and is independent of the radius of curvature.

The basis for this work has been to assume that when a hose is bent to some defined radius of curvature that its cross-section rather than remaining circular takes the form of an ellipse.

Based upon the theory of corded structures and assuming that the cords forming the hose construction are inextensible, changes in the bias angle of the cords about the hose section and the properties of the assumed elliptical cross-section may be defined.

Details of the theory used and calculations performed may be found in Appendix (4); however the method may briefly be outlined as follows:-

(1) From a consideration of the geometry of the cross-biased cord ply construction of the hose the changes in bias angle around the hose as it is bent may be defined by:-

$$\sin \alpha / \sin \alpha' = 1 + (y - \delta)/R$$

(NB: Terms defined in Appendix (4)).

(2) The parameters of the ellipse forming the hose cross-section are then determined using two criteria:-

First the cord length per wrap remains constant (inextensible cord).

Secondly the ratio of the principle surface tensions (ie hoop and axial) must satisfy the general relationship for corded structures:-

$N_a / N_s = \tan^2 \alpha$ at every point around the section.

Because of the number of unknowns, an interactive approach was adopted for this solution.

(3) It was determined that by accepting a limited degree of error in the calculation $\simeq$ 2%, it was possible to set the variable δ (displacement of neutral axis) equal to zero, thereby considerably simplifying the previous calculations.

(4) Utilising the above it was determined that the axial tension force N_a could be related to Y (distance of element from neutral axis) by a relationship which could be approximated to a straight line described by:-

$$N_a = A + B_Y$$

A being a constant relating to the hose size and

B being a constant relating to hose size and radius of curvature. Both being pressure dependent.

(5) The stiffness characteristic EI then may be calculated from:-

$EI = B\Pi r^3 R$ (B being the constant from the above equation).

Typical results of the above calculations, the values of EI (due solely to the effect of internal pressure) are plotted, against radius of curvature for three different hose sizes; Figure (63). Also superimposed on this graph are results derived from practical tests. Whilst it may be seen from Figure (63) that the plots of theoretical and actual stiffness increase generally relate in so far as they are of the same order of magnitude, they differ in a number of other respects; Namely:-

(a) All theoretical curves indicate a similar pattern of the stiffness increasing from a low value, when the hose is only slightly bent, to reach a substantially constant value for most levels of curvature. Whilst the practical examples indicate similar stiffness levels at maximum curvature, (ie: small radii), they rapidly diverge from the theoretical lines, and infact, stiffness increases with decreasing curvature. (This may have been expected - refer to Chapter VIII.1).

(b) It may be seen from the curves representative of a 24 in. hose that constructional differences also affect this property significantly. This is not directly accounted for by the theory.

How can these discrepancies be explained:-

Firstly the theoretical analysis has been based upon the principle of the Neutral Angle (refer to Reference 8), that is the reinforcing plies are applied at such an angle to the hose axis that the hose neither extends or contracts in length or inversely diameter when pressurised. However in actual hose construction the above is rarely adhered to. Most frequently, plies are applied at an angle such that the hose would tend to expand in diameter when pressurised. This effect

is however negated by the application of a circumferential helical wire. (Refer to Chapter X and Figure (38)). Although the helical wire is incorporated as a component enabling the hose to withstand external pressure without collapse it is also secondarily used to assist hose stressing during internal pressurisation. In this form of construction therefore the main reinforcing plies expand against the helical wire which accommodated additional circumferential stresses and simultaneously prevents radial expansion of the hose.
Secondly it has been assumed that changes in axial surface tension around the hose circumference relate to an isotropic material. This is certainly not the case, under tensile stressing the cords are naturally very effective in accommodating a high proportion of the applied stress, however the compressive stress capability of such cords is extremely limited, local buckling of the cords occurring and a greater proportion of the stress is carried by the matrix material, (ie: rubber). Both of the above are naturally dependent upon the cord application angle and the specific properties of both cords and rubbers.
Immediately it may be seen that apart from changes in the position of the neutral axis, due to geometrical changes during bending; similar changes, probably of greater proportions occur as a result of the effective stress capability of the hose materials in the modes described.
Some of the hypothesised effects may be seen in the case of the 24 inch hose curves shown in Figure (63), these may be summarised as:-

(1) Standard "wire cord" construction - cord angle marginally below neutral angle (ie: 51 deg : 54.4 deg), imparting partial circumferential loading on the helical wire.

(2) "Fabric cord" construction - Cord angle considerably below neutral angle (ie: 45 deg : 54.4 deg), the helical wire contributing approximately 40-50% to the circumferential stress capability.

(3) "Wire-Free" construction - Helical wire replaced by radical design incorporated a variety of cord types and application angles.

It may readily be seen that any theoretical model incorporating the above variants would be extremely complex. Work is proceeding in this area, rather slowly, in an attempt to improve the model.

Whilst it may be readily observed that the theoretical model currently generated is not adequate to describe the phenomenon it has served as an initial estimate and also promoted discussion of the various inter-relationships now apparent in the study of hose stiffness.

Once again (refer also to Chapter VIII.1) it is abundantly clear that a single or even a series of values in relation to describing the flexural stiffness of a hose is completely inadequate.

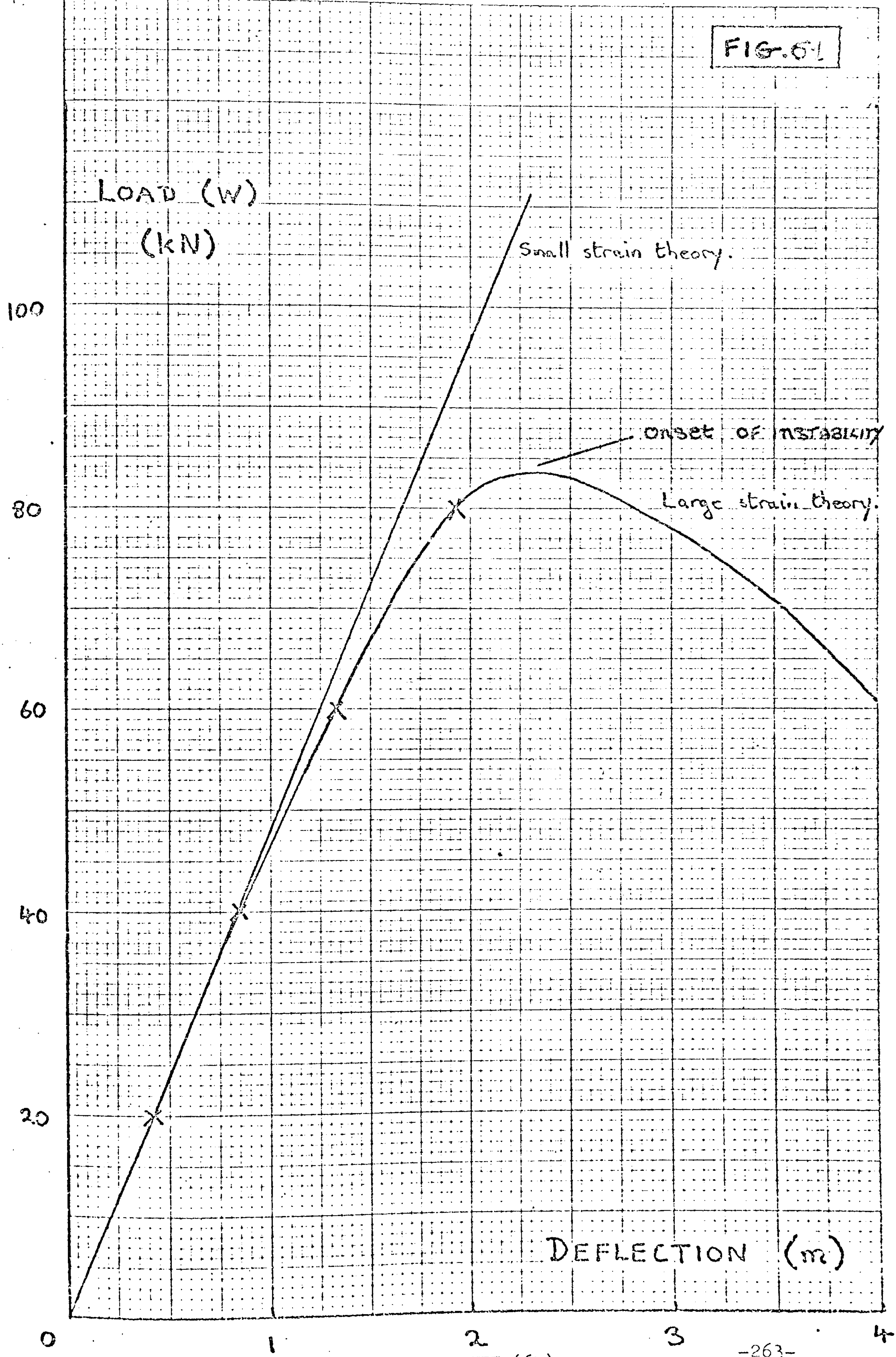


FIGURE (61)
DEFLECTION CURVE SHOWING ONSET OF GEOMETRIC INSTABILITY

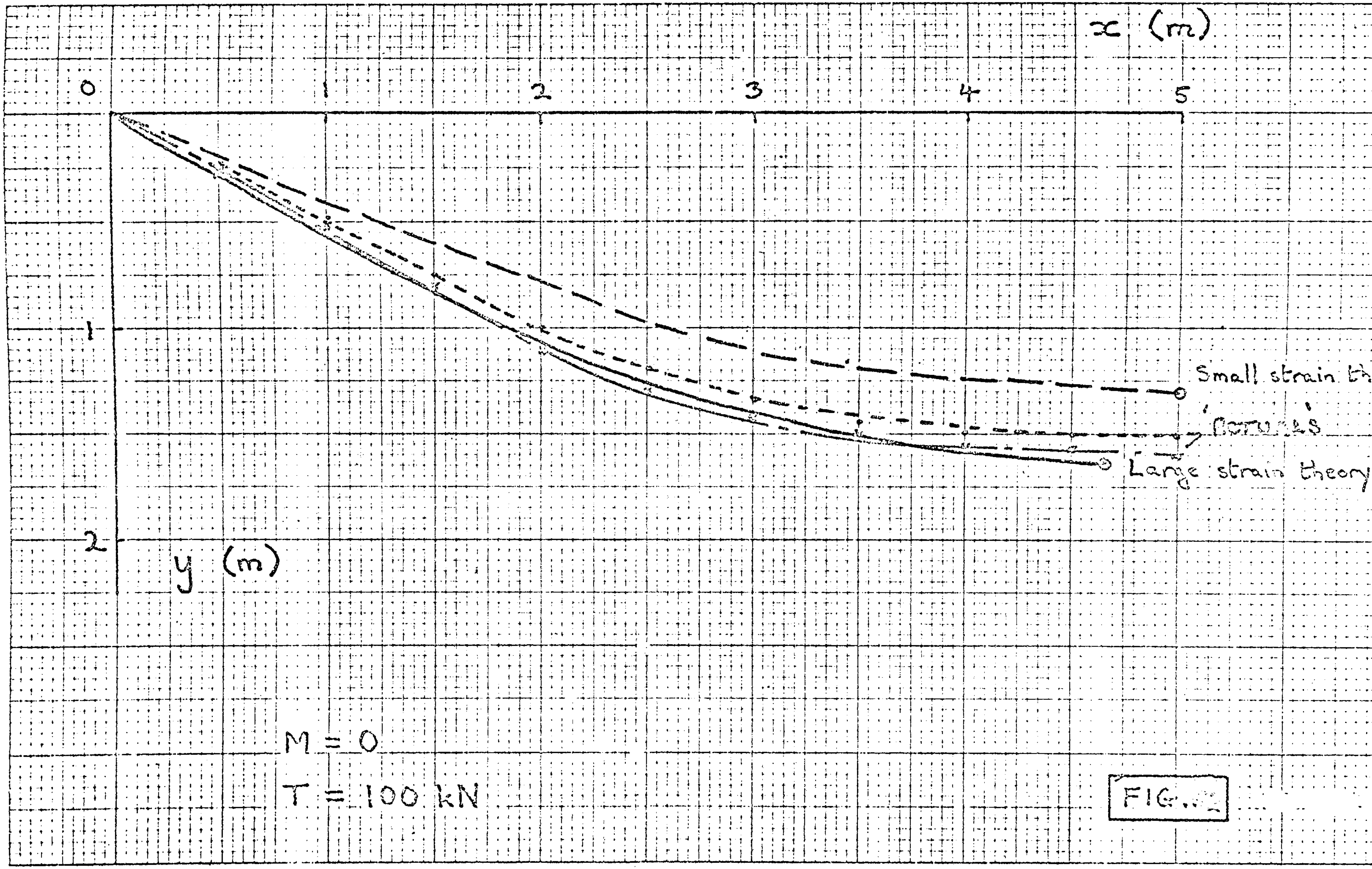


FIGURE (62)
COMPARISON OF BENDING DEFLECTIONS.

Plot of nett increased stiffness, resulti
from an increase in pressure of 225 lbf/i

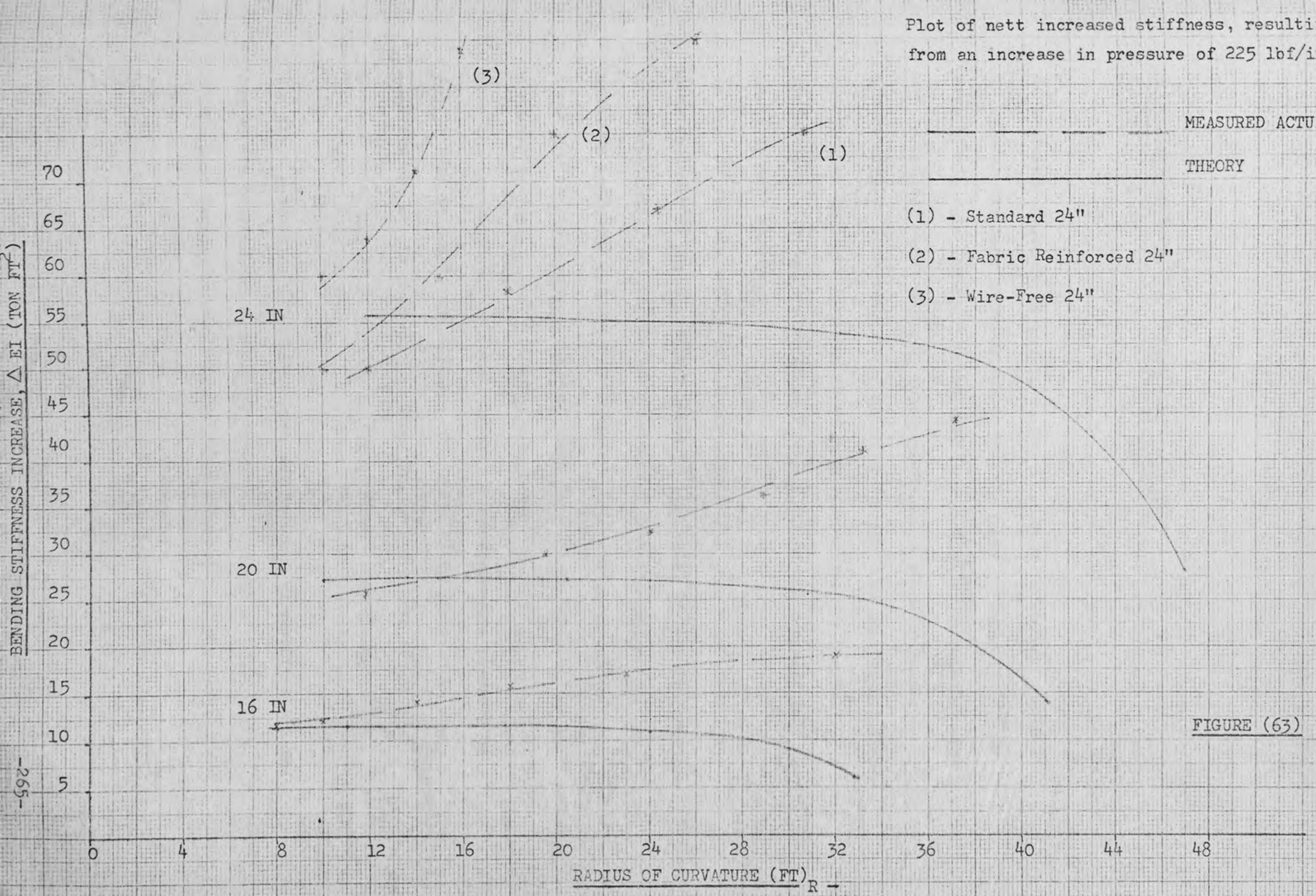


FIGURE (63)

X ANALYSIS OF HOSE CONSTRUCTION/GEOMETRY

It appeared obviously desirable during some stage of the micro study to examine the hose construction in greater depth in order to assess each item's role (if known), how well it forefilled this role and what its redundancy might mean. By doing so one was also afforded the opportunity to form basic opinions about current hose design, hypothesis changes, or modifications and store such information for use in conjunction with failure analysis with respect of the proposed full scale dynamic test rig.

In view of its history it is not surprising that when hose eventually took to water it was ultimately bound to fail until further improvements in understanding and improving its physical characteristics could be made. On land, the principal stresses to which a hose is subject are those due to its working pressure; apart from slight movement due to pressure variations most hoses are essentially immobile or, at worst, move through well defined paths. At sea, whether carrying pressure or not, a hose is constantly working, constantly being stressed in every possible way at the whim of the waves. The offshore hose cannot rest between brief spells of use; it is being tested every minute of its life.

The first generation of offshore hose was designed in exactly the same way as its traditional counterpart on-shore. In spite of this, early trials were largely successful. The first commercial single point mooring buoy, manufactured by SPM, and installed by NIRI for Shell in 1958 was equipped with 8" bore hoses.

The SPM facility was quickly recognised for its potential for providing low capital cost port facilities in remote corners of the world and also, with the growth of tankers, for providing deep water anchorages where this would normally be impossible. With the mushrooming of SPM terminals, offshore hose was being produced in greater and greater quantities - though still no traditional designs. Improvements were effected, however. Buoyancy for the floating hoses, which had originally been provided merely by lashed-on oil cans and then strapped on plastic floats was, by 1963 integral to the hose through the use of closed cell sponge. This concept was first patented by Dunlop in 1962 although improvements to the original idea have naturally altered the appearance of the system now. Hoses also increased in size from the original 8" through 10", 12" and then 16".

And this was the situation when hose problems, clearly related to its use offshore, began to be identified. The 16" bore hoses had held sway as being the largest afloat for some 3 years (a delay in the advance to larger sizes owing to the limitations of tanker derricks for lifting heavier hoses) and hoses were beginning to fail through fatigue induced by the action of the sea. These problems were particularly concentrated where the hose was fixed to a rigid structure, the first hose off the buoy for both the floating and submarine strings, the connections to the pipeline end manifold on the seabed and connection to any submarine buoyancy tanks. These problems had always existed of course but had been aggravated by the continuing trend to larger bore sizes and had eventually reached an unacceptable level. The cause of the problems was obvious, the patently different dynamic behaviour of the rigid structures compared with the flexible hoses and the concentration of bending stresses which this created at the points where the two came together. The solution also seemed clear, at least in principle - remove or smooth the discontinuities at the connection points. This was simply an extension of the traditional practice of reinforcing a hose in the area connecting it to its rigid end fittings. In effect, a buoy was treated as an oversized fitting and the extra reinforcing plies over the shoulder area of the hose were increased in number giving birth to the Dunlop Sampson Hose in 1968.

This improvement was one of the first structural differences to set offshore hose apart from its shore based ancestry. Other changes which were also specific to the new generation were the development of lightweight hoses for the tail of the floating strings to ease the load on tanker derricks and, somewhat later, high flowrate linings for these same hoses. This latter improvement came about when the advance to larger sizes had got underway again. 20" and 24" hoses could clearly not be hauled up the ship's side so, in order to fully use the capacity which the larger sizes allowed, the smaller bore tail strings had to be able to cope with higher rates of flow than would normally have been recommended. The usual lining compound was rated to carry flow velocities of up to 40 fps but the new lining, reinforced with multiple plies of nylon fabric to resist the drag forces and cavitation effects was uprated to 70 fps.

The design improvements which had been brought about through experience with smaller bore hoses were applied to the increasingly popular 20" and 24" size but with only limited success as insufficient allowance for the effects of size were made. Two further significant changes were therefore introduced in 1971. The hoses which had been reinforced with a single helically wound wire over the main reinforcing plies had been found, in the larger sizes, to be prone to a delamination at the helix level. In effect, the helical turns were perforating the rubber matrix and bringing about premature separation within the hose carcase leading to failure. This situation could be improved by reducing the "perforation" through opening the wire pitch. This move alone would, however, have reduced the crush resistance of the hose and so a second, smaller, helical wire was added to the construction to run above the first and between its coils. The contact area between adjacent levels was therefore increased and, in addition, the second wire had the effect of locking the first in place over the main plies. This change, admittedly based largely upon an intuitive understanding of the problems, clearly overcame the hose limitation in this respect and extended its lifetime. The second improvement was of a similar nature though for different reasons. The sampson concept was successful but was insufficient for the larger bore hoses and so, in addition to the extra plies in the shoulder area, a further helical wire was run to increase the stiffness of the hose and lessen the stiffness jump to the fitting and whatever rigid structure to which it was attached. This third helical wire increased the resistance of the hose bore to deformation and so avoided the tendency to ovality of cross section under severe bending moments. This change, too, which created the Super Sampson Hose, was most effective in improving hose lifetime in the severest situations.

Apart from basic design changes, production techniques also had to improve to cope effectively with the larger bore sizes. Wire cord plies were introduced as the main reinforcement for the 24" bore hoses, allowing a large reduction in the number of plies due to the difference in strength. The bulk of the hose was thus lowered whilst at the same time rendering the task of binding the plies to the end fittings that much more reliable. Allied to this change was an amendment of the ply application angle from the neutral value of $54^\circ\ 44'$ to 50°. This change required the embedded helical wires to take a larger share of the hoop stress created in the pressurised hose and so further relieved the main plies. Tensile strength of the hose was also improved by this alteration.

Rubber technology was not by any means forgotten amongst these improvements. The internal carcase of the hose, which had traditionally been based upon neoprene rubber compounds, was changed to a nitrile rubber, improving the resistance to aromatic substances and increasing adhesion levels. Adhesion levels were further enhanced by the introduction of calendered cord in which the main ply cord material was sandwiched in a rubber matrix prior to application on the hose. This also improved the accuracy of cord spacing on the hose and increased production efficiency.

There are two major design philosophies for the fabrication of hose, one is to make the hose stiff enough to successfully resist the loads applied, whilst the other is to use flexibility to allow the hose to deflect with the input loads thereby maintaining a low stress level.
To date most hose manufacturers have chosen to adopt the first approach evolved from smaller bore hoses, 6 in. and 8 in. As the demand for larger hoses grew the industry responded by scaling up the construction of the smaller bore hoses and strengthened them with additional steel body plies and helical wires.
It was reported that 12 in. and 16 in. had substantial failure rates when first constructed, but internal modification as based on use or experience eventually gave satisfactory results. Unfortunately failures in 20in. and 24in. hoses represent a much larger loss not only in hose cost, but in pollution and installation costs.

PRESENT HOSE DESIGNS

Hoses intended for use offshore still carry the hereditary pattern of their generic past; lining, main plies, embedded helical wire and cover. These components are, after all, unlikely to disappear but the details of their form and purpose have and will continue to change in keeping with the new environment.
All offshore hoses have certain components which are installed to resist the loading conditions previously described. Figure (18) shows a typical hose cross-section.
The linings of offshore hoses bear direct comparison with land based forms, the only distinction being the natural improvements in

compounding necessary to suit the material to the rigours of its service. High flowrate linings are probably unique to the marine hose but merely extend the adaption to its most extreme form yet. Adhesion of linings to end fittings is usually accomplished in two stages. After immediate degreasing and shot blasting, the fitting spigots are chemically treated and lined externally with the same compound rubber as the hose lining. The lining forming the body of the hose is later laid upon this spigot lining and, when the hose is eventually cured, fuses with it.

End fittings have merely grown with the size of hose and are designed to have a length sufficient to provide an effective chemical and physical bond to hose body. Small bore hose fittings often have serrations to promote retention and larger bore hoses, including those for use onshore, simply use enlarged and more discrete ribs behind which the various hose components are securely bound. A single rib is usually sufficient to retain the hose carcase against the various forces imposed on it but it is often more convenient to add further ribs so that the higher carcase layers can also be locked in place.

As the fitting spigots naturally have to possess sufficient thickness to avoid distortion during service, linings have to climb this low step from the body of the hose. This step could possibly be a weak point in the lining and so is smoothed and reinforced by the addition of further lining material in this region tapering into the hose body.

The entire hose lining, including that applied initially over the spigots, is backed by an open weave fabric called a breaker. This material strengthens the lining in much the way that concrete is reinforced with steel. Adhesion between the lining and the subsequent main reinforcing plies is also promoted.

It is usual for the main reinforcing plies to be added immediately over the lining breaker but, with the development of specialised hoses for the newer breeds of exposed location buoys, the main ply arrangement itself is becoming more diverse. If fabric plies are to be used, these are usually added up to six in number before being bound in over the spigots. If further reinforcement is still required a second grouping of main plies is added after the application of a cushion layer of thin rubber. Wire cord plies are applied in pairs, binding in over the spigots after each pair until the requisite number, normally six, is obtained. Occasionally, however, in cases when high tensile strength is required, some of these main reinforcing plies are added after the hoop or helical reinforcement. This move ensures that the outer group of plies carries the principal tensile loading and so avoids the possibility of delamination as the stressed main plies attempt to relieve their loads by moving radially inwards. Uneven contours immediately over all the binding wires are smoothed by the application of short lengths of rubber compound before further ply material is added. In the case of hoses required to be electrically discontinuous, this filler rubber also serves as a barrier to contact between conductive components in the hose.

Reinforcement to the shoulders of the hose, needed to lessen the stiffness discontinuity between the hose body and fittings and strengthen the carcase to take the loads concentrated in these areas, is added through further short length plies immediately after the main plies. Additional shoulder reinforcement occasionally occurs elsewhere -

especially later in the carcase of a Super Sampson hose - but there are invariably at least some shoulder plies after the main reinforcement. These plies are not bound in, there is little point as they are free in a sense at one end anyway, but adhere to the plies immediately beneath them and so supplement their effects.

Hoop or helical reinforcement is normally the next major component to be added and this is embedded in a rubber matrix. In the case of a wire helix, rubber to a thickness of approximately half the cross sectional diameter of the wire is applied first and then the wire is applied under tension to firmly embed it in the compound. Rubber is then again applied to cover the wire, the gauge of the rubber being chosen so that, upon curing, the upper and lower rubber layers flow together to fully encapsulate the wire.

More plies of reinforcing material - not necessarily of the same type as the earlier reinforcing plies - are now added to lessen the possibility of an inner carcase delamination from the embedded wire or hoops. This possibility is due to the stress relieving tendency of the main plies to move inwards when the hose is under axial tension. This tendency is only resisted by the presence of the embedded helical or hoop reinforcement and the adhesion of the rubber compound to it. Without the plies over the wire the inner plies would eventually strip away and presage hose failure. The stronger the plies over the wire, the stronger will be the hose in resisting tension.

The second wire, when it is used, is applied at the same pitch and in the same manner as the first, embedded in and then covered with rubber. This second wire extends further than the first to pass over its forerunner's final touch pitch turns and move nearer the spigot where it is welded behind one of the spigot bands. The outer "filler" rubber is again strengthened against deformation by the addition of reinforcing plies but just before this stage the basic compounds of the hose carcase change from nitrile to neoprene. Whilst nitrile rubbers resist aromatics more effectively, neoprenes are more abrasion resistant and so the final filler material is a blend of these two compounds to ensure adhesive compatability between the two sections of the carcase.

For a submarine hose, all that remains is to add the cover construction built up in alternate layers of rubber and open weave "breaker" fabric. Location bands for submarine floats are added later if necessary. Hoses destined to become Selflotes have a thin rubber carcase cover added and are then wrapped for cure.

After curing, the hose carcase is chemically treated to promote bond strength to the layers of closed cell sponge which provide the hose with buoyancy. Sponge is added until the diameter is sufficient to give the intended buoyancy reserve. Extra sponge is usually added in the shoulder areas of the hose to support the additional concentration of weight due to the fittings, bindings and shoulder plies. A layer of blowing sponge covers the closed cell sponge, this being a rubber compound which, upon curing produces gases which increase its size like leavened bread and so heightens the consolidating pressure within the hose body.

A breaker ply separates the blowing sponge from the final cover construction which, like the submarine hose, is a composite structure of alternate rubber and breaker layers. The characteristic orange spiral of the floating hose completes the hose which is then cured for the second and final time.

The purpose of the various hose components has now been explained but do they infact perform that role, how do they fail, what back-up is provided? These are some of the questions asked by the F.M.E.A. analysis.

F.M.E.A.

Failure Mode and Effect Analysis provides a means for listing all of the components in a system, innumerating the ways each component can fail and analysing what effect that failure might have on the system. (NB: Such techniques were used to document engineering considerations when space flights were designed). The advantage of such analyses are that design consistency and system integrity are verified. Further engineering considerations may be documented in a manner that satisfies the analysis process without raising questions concerning design philosophy.

In order to accomplish an F.M.E.A Analysis, it is necessary to enumerate the system by listing the components in the system and their intended function. The number of ways in which each component can fail is then enumerated and it is determined if the component has a redundancy (or back-up) for each particular failure mode. Usually, if a redundancy is considered to exist, no effect on the hose is noted; on the other hand, if a redundancy does not exist, the hose may or may not be affected, dependent upon the role of the component.

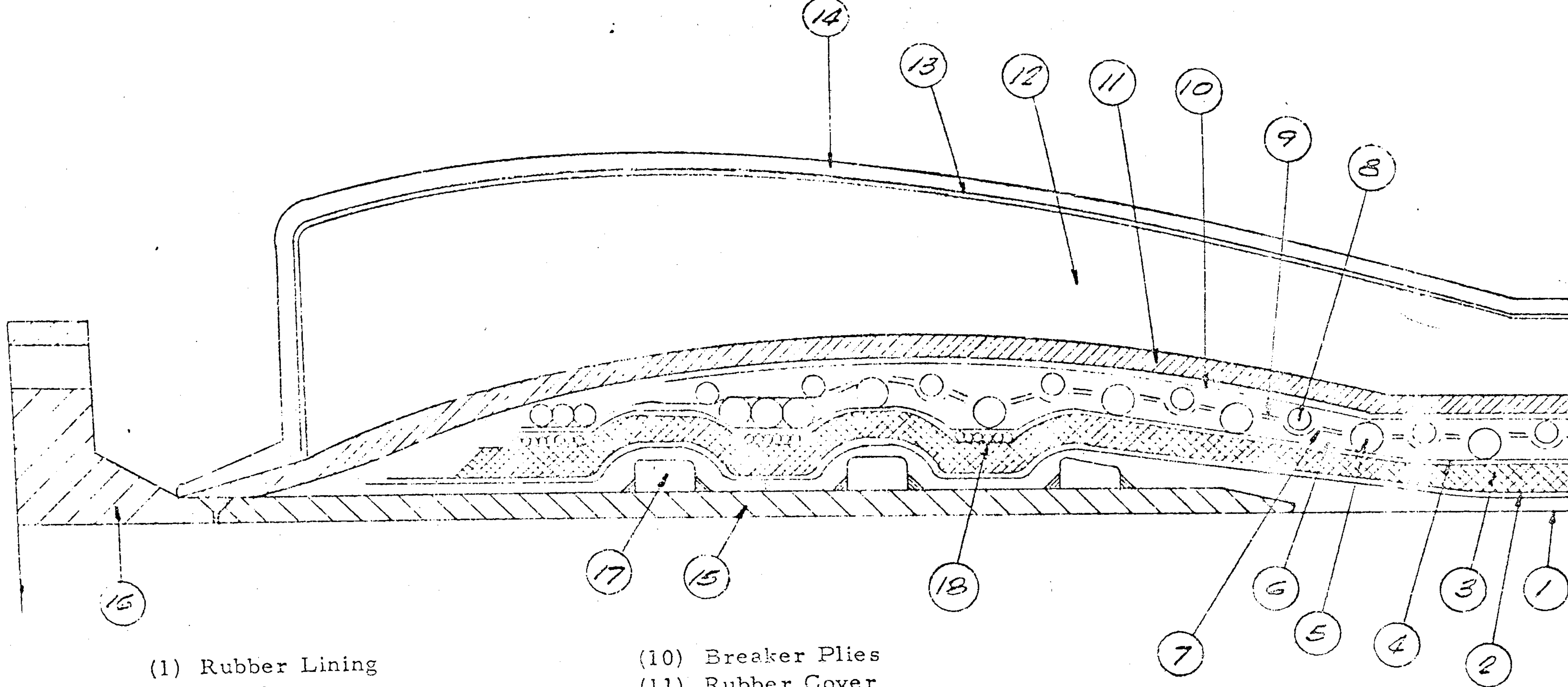
The following paragraphs describe the results of the F.M.E.A. study whilst the F.M.E.A. is presented in Appendix (3).

(a) The rubber lining and first breaker ply are the primary barrier to hydro-carbon leakage and extreme care must be taken to ensure integrity. In process inspection of liner and liner nipple bond is critical; (Current practice of "pricking" blisters in the liner/lining material and/or bore repairs are totally inappropriate).

(b) Reinforcing plies used to resist internal pressure under a safety factor of 5 are suitably redundant, perhaps too much so. Care in application of the reinforcements is however extremely important, (Angle, tension, overlaps etc) if the redundancy factor is not to be reduced and/or wire slippage occurs. Wire "binding-in" at the ends is very important. (Most failures in cases of nipple pull out have been attributed to binding wire failure). The abundance of main plies, all requiring "binding-in" does not help this factor. Thought should be given to alternate, more positive, 'locking' of the reinforcement onto the fitting.

(c) Helical wire movement has been observed in practice and both internal and external protrusion experienced. Considerable thought should be given to alternatives to these 'heavy gauge' helical wires, (Are they necessary? Could they be replaced by discrete hoops? etc). However assuming such items are necessary, great care should be taken in their application, ie: tension, positioning, non formation of voids etc. and some thought given to an adequate bonding system. Rather than assuming at present that the surrounding rubber matrix maintains their orientation.

(d) Observed collapses of nipple ends and the preponderance of failures at or about the ends of shoulder plies etc., points to the need for a much deeper understanding of the required hose stiffness properties. Also the need to gradually change from the stiff hose fitting to the relatively flexible main hose carcase.



(1) Rubber Lining
(2) Breaker
(3) Wire Cords (4-8 Plies)
(4) Breaker
(5) Wire Helic, High Tensile Steel
(6) Filler Rubber
(7) Fabric Layer
(8) Wire Helic, High Tensile Steel
(9) Filler Rubber
(10) Breaker Plies
(11) Rubber Cover
(12) Flotation (Close Cellar Foam)
(13) Breaker
(14) Outer Cover
(15) Nipple
(16) Flange
(17) RIBS (Usually 3)
(18) Binding Wire

FIGURE 18 SCHEMATIC OF TYPICAL CROSS SECTION OF HOSE

XI.1 Introduction

Whilst it was stated in an earlier section of this tesis that one of the aims of the project was to provide the Company with a meaningful test method, proceedure and equipment: it was naturally important to reasses certain assumptions which were made in this regard. In order to assist in this re-evaluation of need, four courses of action were postulated by the writer for discussion:-

a) Do nothing with regard to simulation rig testing but rely entirely on field trials, failure data analysis and other service information, in order to improve product design.

b) Consider the possibility of combining with others, (either customers, competitors or government bodies) in a joint enterprise.

c) To consider the use of a small simple rig(s) that would be able to do specific tests on small hose samples, eg- a simple cantilever type test to evaluate end attachment design.

d) Proceed with the development of a test rig, capable of testing full scale hoses under simulated service conditions.

Discussions followed which involved not only senior management from Oil and Marine Division, also the Directors of Dunlop Engineering Group and Central research division.

It was agreed that the increasing importance of the single point mooring system was leading many oil companies and government agencies to pay more attention to hose performance and reliability. Particularly as without reliable hoses the oil production of many of the developing exporting countries would be seriously affected.

Also noted was, that as the worlds largest hose manufacturer, Dunlop was being increasingly criti cised for not being able to offer a product designed and tested in a scientific and parametric manner .

Based upon these factors it was recognised that the company for both commercial and tethnical reasons, had already relied too greatly on alternative (a).

With regard to option (b), it was already known that an international oil company had decided to persue the problem indeoendently because of the difficulties of getting a concerted action plan between the various users. This background, together with the historically and nationaly based fierce competition between manufactures made such a joint venture not appear attractive. It was also considered that Dunlop had a great deal more to lose in terms of R&D todate, than prospective partners.

Option (c) was discussed at great length, as this appeared to be a viable approach at significantly depreciated capital expenditure, when compared to the original scheme. However during this consideration that at least two or more types of "simple" rig would be required and that, even so, most compound testing (ie- imposing independent force modes, eg; bending + torsion etc) would not be practical unless even these rigs were of a relatively sophisticated design.

It was therefore concluded that the above approach, whilst it might be useful were we already too have identified the predomenent failure modes did not show significant capital savings against the original scheme, and those that did incurred a penalty of much reduced flexibility of any test strategy.

Considering the original scheme, it was determined that this approach was viable and should therefore proceed, although constant attention should be paid to the trimming of any non-essential sophistication that might possibly be added at a later stage once test experience had been gained. In this context it was indicated that Capital expenditure in the region of £200,000 - £250,000 would not be unreconcilable with the overall operation of the division.

The following pages describe the design of the test rig, through the manufacture, installation and commissioning stage, to an indication and analysis of the initial test results.

XI.2 ESTABLISHMENT OF DESIGN PARAMETERS

From the results of both the field tests conducted (Forcados project) and numerous model tests, sufficient data was available upon which to base both the design of a test rig and to initiate a test programme. It was determined that the force modes to be considered were :-

Axial load (ie tension and compression)

Bending loads (in various planes and positions along the hose)

Torsion

Internal Pressurisation *

Further it was also apparent that in order to conduct meaningful tests, the above loads must be capable of application both singularly (nominally in a pure mode), and also in combinations up to a simultaneous application of all loading modes.

In addition to the force modes it was necessary to consider the related displacement functions, also to determine whether any such test rig should be operated functionally in a load or displacement oriented manner. That is should one have the ability to subject the test hose to given deflection situation and to monitor the loads, the converse of the above, or indeed a combination of both methods. (Naturally the course could have a significant bearing on the overall costs of the project). In this respect the evidence available was inconclusive, much dependent on the motion originating mechanism, ie waves, buoy etc and also on the relative stiffness properties of the hose. Because of the probability that both modes actually occur for present hose designs/system service, it was considered prudent to make available both options. (At this stage it seemed likely that smaller diameter hoses might be best suited to a deflection based test whilst larger sizes utilising a load oriented approach, the precise split being a total unknown).

In respect of hose size it was directly established from current sales/ failure figures that the hose sizes of immediate interest were 24in, 20in and 16in respectively in terms of priority. In order to make allowance for future market trends (already apparent), it was considered that 30in and 36in bore sizes should also be inclusive of the test range. In terms of test hose length, naturally the minimum length possible, without introducing inordinate end effects was the optimum consideration. For the range of hoses selected; considering end 'build-ups', special reinforcements etc an optimum minimum figure of 25 ft was established. However, it was similarly recognised that were we to be in a situation to test hoses of alternate manufacturers , we could hardly expect to obtain samples of the appropriate length. Similarly the same argumet applied to the testing of 'service returned' speciments. A maximum length capability of fourty feet was therefore selected, this being marginally greater than the maximum 'standard' hose length currently in general production. Naturally this decision on reprecussion effects on the extent of deflections etc to be accommodated by the design.

In terms of loading frequencies, wave forms etc, much of this information was available ex, the above stated field and model tests. Whilst in service random wear patterns were established, it was considered that, similar to the model testing format, that we should restrict the complicity of testing by employing regular sine waveforms. However the need, eventually, to introduce totally representative conditions, possibly prerecordor from a field test, was noted so as to make provision for future sophistication of the test rig.

Considerable thought was given to the question of test acceleration and scaling at this point in time. Generally these could be achieved in two manners:-

a) Increasing the load/deflection magnitudes employed

b) Increasing the load application frequency(s).

In the first case, analysis of the test results indicted that the majority of service loads were accommodated below 50% of a value that might (with

contd/...

the limited knowledge available) be taken to be the effective design limit. It was, therefore, postulated that it might be possible to accelerate testing by increasing load magnitude with respect to reality. A contingency overload factor of 2 was thus incorporated into the rig design.
Similarly it was postultated in the second case that an increase in application frequency, or a disproportionate application of maximum loads at high frequency could also be used to increase test acceleration.
Further from the field tests and model tests it was recognised that bend flexing tended to occur at higher frequencies than other force modes. It thus appeared appropriate to similarly disproportionately accelerate each of the force modes.
In considering the above two courses of action it was realised that their use during testing would have to be carefully monitored and controlled in order to avoid unrealistic factors to influence test results. In the former case the onset of local plasticity, property variation with load etc must all be considered. Similarly in the latter case hysteresis is an obvious problem; inducing heat generation and possible bending system breakdown which could not be generally representative of reality.
* In addition to the various factors indicated as testing criteria, a study was conducted into the desirability/possibility of incorporating the following, also into the rig design.

Pressurisation (using crude oil)

Oil flow (typical values 5,000 tons/hr)

Oil temperature

Ambient temperature

Ultra violet degradation

Corrosion

Whilst many of the above are valid test criteria, most (certainly oil flow), would be extremely expensive to incorporate such into the design of the test rig as discussed.

XI.3 Fuctional Specification

Because of its direct relationship to the dynamic test rig design, hence Oil & Marine Divisions Research and Development programme; the above specification has been ommitted from this thesis. For review of this specification reference should be made to (I4); this however may be subject to restriction.

XI 4 ALTERNATE DESIGN CONCEPTS

INTRODUCTION

On the basis of the functional design specification put forward, as described in the previous section, a number of alternate design concepts were generated. Radically different designs were purposely sought, engineered and costed in order that a final design decision could be made against the broadest background possible. Although the designs are stated to be "radically different" they all embody certain similarities which were considered to be virtually self selecting because of the type of rig in question. These similarities may be summarised as:-

(a) POWER SOURCE

Because of the high magnitude of loading involved, (both direct and inertia generated), coupled with relatively low velocities of application, hydraulically activated mechanisms offered obvious advantages. This was particularly so for 'linear' motion functions. For rotational functions alternate systems employing D.C. drives were evaluated; these however were found not to be capable of providing the required simultaneous torque/speed relationships deemed necessary.

(b) FATIGUE FLEXING

Production of an alternating flexural mode was sought by a number of methods. However because of the high inertia of the hose under test methods, apart from that of rotating the hose ("Rotary Bending"), resulted in extremely heavy test structures and resultant high power requirement.

Four basic design concepts will be outlined in this section; each may be briefly described as:-

(1) Hose tested in a wholly vertical configuration.

(2) Hose tested in a wholly horizontal configuration, (water or alternate support medium employed).

(3) Hose tested in a wholly horizontal configuration, (mechanically supported).

(4) Hose tested in a semi-vertical mode.

DESIGN 1 - VERTICAL CONFIGURATION

Figure (71), Drawing No: 1-020-0188.

(a) STRUCTURE

It will be seen from the drawing that a vertical configuration has been adopted for the rig and the "rotary bending concept" has been incorporated. The rig consists fundamentally of a tower structure of braced portal form in the height of which five floors or decks are located. The top deck carries machinery and the remaining decks allow access to the hose under test for examination and monitoring purposes. The height from ground level to the topmost floor of the rig is approximately 56 ft.

Access to the decks is by stairways and if desired a small passenger lift can also be incorporated as illustrated. The entire unit is founded on a concrete raft or piled foundation, depending upon site conditions, and the top machinery deck is provided with a weather-proof housing.

It will be seen that at the upper end of the main portal structure a movable crosshead is incorporated. This unit is raised or lowered by a pair of geared hoists and can be secured at any one of five levels to accommodate the required lengths of hose. The crosshead carries a tilting and rotating assembly which applies the bending moment and torque to one end of the hose under test.

A similar tilting and rotating assembly is placed in a separate assembly at ground level. This latter tilt/rotate unit can be raised or lowered through a distance of approximately 6 ft. by an axial loading cylinder which is located inside the bedplate of this assembly.

Both of the tilt/rotate assemblies are equipped with cooling water passages so that the hose can be filled with water for loading purposes and the required quantity of water can be circulated through the hose under test for cooling purposes.

It will be seen from the drawing that the tilting movement at either end which is, of course, the movement required to produce a bending moment on the hose, is achieved by a straight-forward arrangement of hydraulic cylinders. The rotation and torque functions are provided by geared hydraulic motors which rotate a slewing ring inside each tilt/rotate assembly.

All motions are available independently to together in any combination so that the hose under test may be continuously cycled within the repetition frequencies laid down in the specification.

It is assumed that the hose to be tested arrives at the rig on a suitable trailer having been previously fitted at each end with a standard form of adaptor/blanking plate. One pair of adaptor plates is required for each size of hose to be tested. The adaptor plate fitted to the upper end of the hose to be tested is provided with a temporary eyebolt. Having positioned the trailer carrying the hose with the upper end of the hose alongside the rig, the wire rope from a hose hoist located on the top deck of the rig is lowered down through the centreline of the crosshead and through the upper tilt/rotate assembly and attached to the eyebolt in the upper hose adaptor plate. The hose may now be hoisted into position and will be correctly located relative to the studs on the upper tilt/rotate assembly, allowing the adaptor plate to be bolted into position. At this stage the lower tilt/rotate assembly will be in its lowest position and the hose may be allowed to hang vertically within the tower. The lower tilt/rotate assembly may then be raised to meet the adaptor plate on the hose using the axial loading cylinder. If the bolt hole orientation of the lower flange is found to be incorrect, the hydraulic rotate motor of the lower assembly may be used to bring about the required orientation and the bolting of the lower flange may be completed.

Local "inching" controls are provided at the upper and lower levels of the rig to facilitate aligning the tilt/rotate assemblies with the hose adaptor plates during the assembly procedure discussed.

Having secured the hose in the rig, the necessary water connections, which are supplied complete with swivel joints where required, may be connected up and the hose is ready for test.

To facilitate setting up the rig and subsequent monitoring of the test programme, telephonic communication will be provided between the control cabin and each deck level of the rig.

(b) LOAD GENERATION

The test rig comprises the necessary mechanical and hydraulic components to apply axial, bending and torsional loads to a length of hose. These components are held in a support structure, the axis of the hose being vertical when in the unloaded condition. Hydraulic power is supplied by free standing power packs mounted adjacent to the test rig, and control and monitoring facilities are provided at a single console.

OPERATING PRINCIPLES

The loading of the hose is controlled by electro-hydraulic servo systems which may be programmed to perform a pre-determined sequence of tests. The methods of applying the loads are described below.

AXIAL LOAD

The lower end of the hose is attached to a hydraulic actuator which is pivoted to earth. Axial loads are therefore applied by extension or retraction of the actuator.

The hydraulic pressure within the actuator is kept at a substantially constant level by a servo-control system operating on the swash plate control of the variable displacement pumping system. This method of control allows the end of the hose to move as necessitated by its deflections in bending or torsion, the axial load being largely unaffected.

Positional control is also available for use during handling operations, but it is felt that the large movements of the hose end resulting from the application of bending or torsional loads would cause this type of control to be useless for test purposes.

BENDING LOAD

Bending is achieved by the method referred to in the specification as rotary bending. The hose is placed in a desired bend, in one plane, and then rotated about its own axis.

The flanges at the two ends of the hose are bolted to trunnion mounted plates which are pivoted on the hose centre line. These plates pivot on bearings held in the rig support structure. The angular position of each plate is controlled by a hydraulic actuator.

The actuators are servo-position controlled. The angular position of each end of the hose is controlled relative to earth. The two ends of the hose are controlled independently to selected angles.

Pressure control is also available for these jacks, but it is felt that position control will be more suitable for test purposes.

Rotation of the hose as required for rotary bending is achieved by using hydraulic motors driving through a gearbox at each end of the hose.

These motors and gear boxes are also used to apply the torsional loads.

The hydraulic motors are part of a servo-position control loop, and are controlled such that the two ends of the hose remain in synchronism at all times.

TORSION LOAD

The torsion load is applied by hydraulic motors driving through a gearbox at each end of the hose. The hydraulic motors are servo-position controlled, torsion load being applied by rotating the two ends from their unloaded position in opposite senses.

Pressure control is also available for these motors, but it is felt that position control will be the more suitable for test purposes.

SIMULTANEOUS LOADING

All operating conditions are available simultaneously apart from torsion load and rotation of the hose for compound bending. The simultaneous use of these two facilities would lead to the need for extremely large hydraulic components and an excessive power consumption. For this reason it is considered to be impractical.

LOAD INDICATION

Pressure transducer signals from all loading systems are displayed by meters on the console. These meters give an indication of the load level, but due allowance should be given for frictional effects, mechanical inefficiencies etc. which would give rise to some inaccuracies in the conversion of a pressure signal to a load level.

Alternately direct reading load cell may be employed for linear (ie: actuator) operations although pressure indication would be retained for rotational modes.

(c) SYSTEM PERFORMANCE

Three independent hydraulic systems are used to provide motive power for the test rig.

AXIAL LOADING SYSTEM

HYDRAULIC ACTUATOR

The axial load is applied by a hydraulic cylinder. The stroke of this cylinder is estimated at 6 ft to allow for movement of the hose end resulting from bending deflections. The effective area of the cylinder has been chosen to be 85 in^2.

OPERATING LOADS

The system has been designed to give a load at the hose end of 45 ton, that is 30 ton plus an allowance of 50% for acceleration.

OPERATING PRESSURE

The hydraulic pressure corresponding to the maximum load is 1200 lbf/in^2.

MAXIMUM FLOW

The worst operating conditions are estimated to be a sinusoidal operation with a stroke of $\pm$ 18" and a period of 20 sec. These conditions give rise to a peak flow of 90 g.p.m.

Two pumps are provided to supply this flow driven by a single electric motor through a gearbox.

POWER REQUIREMENTS

Under the conditions described, assuming a load amplitude of $\pm$ 30 ton, the peak power requirement is calculated to be less than 100 H.P. Allowance is made in this calculation for system pressure losses, component inefficiencies etc.

An electric motor of 150 H.P. is provided giving a contingency allowance of 50%.

CONTINGENCY ALLOWANCE

Since the calculated sizes of the system components depend on various assumptions as detailed above, it may be necessary to utilise the contingency allowance to increase the actuator load or its velocity.

BENDING SYSTEM

HYDRAULIC ACTUATORS

The bending loads are applied by hydraulic cylinders at each end of the hose. The stroke of these cylinders is estimated to be 21" to allow 50 deg deflection of the hose flange at each end. The effective area of the cylinders has been chosen to be 40 in^2.

OPERATING LOADS

The system has been designed to give a maximum bending couple of 36 ft.ton at each end of the hose, that is 30 ft ton plus an allowance of 20% for acceleration. This reduced allowance is considered to be sufficient since this system is not designed to give cyclic operation.

OPERATING PRESSURE

The hydraulic pressure corresponding to the maximum load is 1100 lbf/in^2.

MAXIMUM FLOW

This system has not been designed for cyclic operation, it has therefore been possible to utilise a relatively small hydraulic pump which charges accumulators situated at each end of the hose. The hydraulic fluid stored in the accumulators is then used to provide movement of the cylinders.

The cylinders are each controlled by an electro hydraulic servo valve. With the size of servo valve chosen it is possible to achieve full stroke of the cylinders (50 deg rotation) in less than 5 sec. This movement may then be repeated after a further period of 5 sec during which the accumulators are recharged.

The pump which has been chosen is a 5 in^3/rev fixed capacity axial piston type, and there is one 8 gallon accumulator and one 10 gallon back-up bottle at each end of the hose.

POWER REQUIREMENTS

A maximum system pressure of 2000 lbf/in^2 is necessary to provide a sufficient operating pressure range for the accumulators. The hydraulic pump has a nominal output of 25 g.p.m. Allowing for the pump efficiency an input power of 40 H.P. is necessary.

An electric motor of 75 H.P. is provided giving a contingency allowance greater than 50%.

CONTINGENCY ALLOWANCE

The contingency allowance would be used to give an increase in the system pressure. With a servo valve control system this could give an increase in load and/or velocity as required.

ROTATING SYSTEM

GENERAL DESCRIPTION

This system is used both to rotate the hose for compound bending and to twist the hose differentially for torsional loading. Initial calculations showed that it was impractical to attempt to fulfil both of these requirements simultaneously, the system has therefore been designed for the separate use of these two facilities.

HYDRAULIC ACTUATORS

The actuators for this system are hydraulic motors of the radial piston, dual displacement type, two at each end of the hose. These motors operate either at a high displacement or at a low displacement depending upon the receipt of a hydraulic pressure signal. The motors drive the hose through a spur gearbox.

The motors are used at low displacement for the rotational mode and at the high displacement for the torsion mode. It is therefore possible with a constant hydraulic pressure and fluid flow to obtain a high speed with a low torque for the rotation and to obtain a low speed with a high torque for the torsion.

The two motors at each end of the hose have a total displacement of 60 in^3/rev for the rotational mode and of 376 in^3/rev for the torsion mode. The motors are geared 15:1 to the hose.

OPERATING LOADS

The system has been designed to give an operating torque in excess of 2 ton ft. at each end of the hose in the rotational mode. This is the torque calculated to be sufficient to rotate the hose under maximum bending deflection assuming 70% energy recovery.

In the torsion mode the system has been designed to give a maximum load of 40 ton ft., the acceleration torque being provided by the natural spring force of the hose.

OPERATING PRESSURE

A system pressure of 2,000 lbf/in^2 is used. This pressure is sufficient to provide the torques quoted for the two operating modes.

MAXIMUM FLOW

In the rotating mode a speed of 12 r.p.m. is required, and in the torsion mode the peak speed is estimated to be less than 1 r.p.m. Reference to the hydraulic motor displacements quoted show that the greatest flow is required in the rotating mode. Including the effects of leakage, the flow required in the rotating mode is 85 g.p.m.

Two pumps are provided to supply this flow driven by a single electric motor through a gearbox.

POWER REQUIREMENTS

The hydraulic system is a constant pressure system running at 2000 lbf/in^2 with a peak flow of 85 g.p.m. Under peak flow conditions the power required is 155 H.P. allowance being made for system efficiencies. An electric motor of 250 H.P. is provided giving a contingency allowance in excess of 50%.

CONTINGENCY ALLOWANCE

The contingency allowance could be used to provide an increase in system pressure or maximum flow, the pumps being capable of supplying up to 130 g.p.m.

ANCILLARY SYSTEMS

All necessary ancillary systems, boost supplies, servo pressure supplies, etc. are provided.

CONTROL

The necessary hand positioning controls will be provided on the rig for initial assembly and dismantling purposes but all other controls will be grouped together in a separate control and recorder cabin, the necessary cable ducts connecting this with the rig being incorporated in the foundations. Alternatively overhead cable ducts may be provided.

Operator controls will include manual inputs of loading, deflection and operating frequencies, together with facilities for selecting force or positional based control. Facilities will be provided for automatic operation of the rig to a pre-determined test programme. Suitable analogue displays will be arranged for all the principal motions and loadings. Analogue voltage outputs of these parameters will also be provided for monitoring and recording purposes. The design of the control system will incorporate as a high priority, safety features to enable both force and deflection limits to be preset by the operator and to protect both rig and operators against mal-operation of the equipment.

The hydraulic power packs will be arranged for remote starting from the control and recorder cabin and suitable safety interlocks will be incorporated to prevent hazards to operators or the rig in the event of hydraulic pressure failure, electric power supply failure and loss of cooling water supply.

NB: Approximate foundation loads have been calculated for budgetary purposes, these are detailed on sketch 1-020-0188 (Figure (72)).

BUDGET ESTIMATE (1975 COSTS)

		£
1.	Main supporting structure complete with five decks, stairways and machinery house.	81.070
2.	Mechanical drives, hose mountings, hoists etc.	100.000
3.	Hydraulic power packs, hydraulic cylinders, pipework, starters and cabling.	53.240
4.	Controls, instrumentation and cabling	35.770
5.	Foundations, including piling, floor slab, ground beams etc.	17.750
		314.880
6.	Erection and commissioning	41.000
	TOTAL COST (DESIGN OPTION 1)	328.830

DESIGN 2 - HORIZONTAL CONFIGURATION

(WATER OR 'FLUIDISED BED' SUPPORT MEDIUM)

(a) STRUCTURE

It will be seen from the sketch that a horizontal configuration has been adopted, also the "rotary bending concept" has been incorporated. The rig consists fundamentally of two drive units, one fixed relative to earth and the other on a track. Between these units the hose is supported in a 'water' tank which may be adjusted to suit the length of hose under test. The total length of the rig is approximately 50 ft.

Examining the concept in greater detail it may be seen that only one drive unit is mounted in such a manner as to apply a bending load to the hose. It is considered that to keep one end of the hose always parallel to the rig results in the smallest possible plan area, whilst still enabling a minimum bend radius to be applied over one end of the test hose.

The drive unit imparting bending loads is mounted on a circular track such that it may be rotated using an actuator arrangement. A similar unit is positioned at the opposite end of the rig. This latter unit is mounted on a longitudinal track and can be moved forwards or backwards through a distance of approximately 6ft by an axial loading cylinder which is located onto a fixed track bedplate. Both of the assemblies are equipped with cooling water passages so that the hose can be filled with water for loading/pressurisation purposes and the required quantity of water can be circulated through the hose under test.

Each unit carries the necessary rotating/torque assemblies to provide the necessary motions and loads for the required tests. In order to rotate the hose, hydraulic motors and gearbox assemblies are utilized. Torque functions are provided by double cylinder and wire drum assemblies as shown on the sketch.

All motions are available independently or together in any combination, apart from simultaneous rotation and torsion.

In the rig envisaged the hose is supported or effectively floated in a suitable medium (either water, heavy liquid or fluidised bed) contained within a tank along the axis of the rig. As may be seen from the sketch, additional tank sections may be added or subtracted as necessary in order to accommodate the appropriate hose length. Seals are provided at both ends of the tank in order to permit the required hose/rig motions to be attained.

It is assumed that the hose to be tested arrives at the rig on a suitable trailer having been previously fitted at each end with a standard form of adapter/blanking plate. One pair of adapters is required for each size of hose to be tested. Having positioned the trailer carrying the hose along side the rig, the hose is lifted on slings by a mobile crane and lowered into the tank. At this stage the hose is allowed to freely float in the support medium. The rear location of the axial loading ram, will have previously been bolted into the correct location bed, to accommodate the hose length in question. By extending the axial ram, previously fully retracted, the driving flange may be brought upto the hose adapter and be bolted into position. Mal-alignment of the bolt holes may be corrected either by rotating the rig drive or similarly the hose which would at this stage be freely floating.

The hose would then be pushed along the tank by extending the axial ram, to its mean position, until the appropriate flanges mate. Misalignment of bolt holes would be corrected by "inching" either rotational drives.

Misalignment vertically within the rig would be accommodated by adjusting the fluid level within the tank.

(b) SUPPORT MEDIUM

In order to prevent stressing of the hose in the vertical plane, support of the hose is considered essential. Whilst "floating hose" may be adequately supported in a water bath, "submarine hoses" would naturally sink under normal conditions. In order to prevent the latter, partial or full "blowing" of the hose bore was considered, ie: maintaining air within the bore. However with regard to safety internal pressurisation of the hose either with air or an air/water mixture is ruled out because of the dangers involved.

Three possible solutions are therefore worthy of consideration as a buoyancy medium:-

(a) WATER - Floating hoses may readily be accommodated, while submarine hoses will require the addition of discrete buoyancy collars.

(b) 'HEAVY LIQUIDS' - A number of liquid solutions are available with a specific gravity greater than that of water. (eg: A typical example of such fluids is in coal particle separation processes).

(c) FLUIDISED BED - The water tank may be replaced by a fluidising chamber, equipped with pneumatic blowers over its length. Utilising readily available particle media the hose could be floated within the fluidised bed.

(NB: Because of the rather unique nature of the latter proposal two prototype test evaluations were conducted):-

(1) Using a small test sample, of equivalent density to the worst case test hose condition, floatation tests were conducted in a small fluidised bed normally used in an experimental rubber curing process. The results of such tests were very encouraging, the principle was established, that is the sample floated and control was found to be relatively easily effected.

(2) Because of the practicalities of extrapolating the results of the above tests to a full scale situation it was considered essential that a full scale section of the fluidising chamber be built and sample tests conducted; (see figure (74) - NB: The hose sections used were ballasted to achieve the equivalent unit density to that of the hose being water filled).

The results of the latter test were partially successful. Whilst it was nominally possible to "float" the samples, problems with regard to bed stability, control and 'dead' areas were found to exist. Because of the latter problems relating to rotating the sections within the medium were apparent, ie: considerable frictional resistance was encountered, this naturally being undesirable from any eventual test stance.

However it is considered that with development, eg: better deployment of the air supply, improved baffles etc. the technique could be viable.

(c) LOAD GENERATION

The rig comprises the necessary mechanical and hydraulic components to apply axial, bending and torsional loads to a length of hose. These components are held in a support structure, the axis of the hose being horizontal. Hydraulic power is utilised being supplied by free standing power packs mounted adjacent to the test rig. Control and monitoring facilities are provided at a single console.

OPERATING PRINCIPLES

The methods of applying the loads are described below.

AXIAL LOAD

One end of the hose is attached to a drive unit which is in turn connected to a hydraulic actuator. This drive unit is free to traverse along a slide under the control of the stated actuator. The latter is pivoted to earth at one of a number of pre-determined points, these being selected as suitable for the various hose lengths to be accommodated. Axial loads are therefore applied by extension or retraction of the actuator.

The hydraulic pressure within the actuator would normally be maintained at a substantially constant level by a servo-control system, thus allowing the end of the hose to move freely, or under a constrained load, as necessitated by its deflections in bending or torsion, the axial load being unaffected unless required.

Positional control would also be available for use during handling or test operations.

BENDING LOAD

Bending is achieved by the method referred to as rotary bending, that is the hose is placed in a desired bend, in one plane, and then rotated about its own axis. The flange at one end of the hose is bolted to a unit which is pivoted about a vertical axis, on bearings held in the rig structure. The angular position of this unit is controlled by a hydraulic actuator.

The actuator is servo controlled and enables position or pressure control options.

(NB: The above facility is only provided at one end of the rig structure in order to reduce the size of the tank or fluidising chamber required).

Rotation of the hose as required for rotary bending is achieved by using hydraulic motors driving through a gearbox at each end of the hose. The hydraulic motors are part of a servo-position control loop, and may be controlled such that the two ends of the hose remain in synchronism at all times.

TORSION LOAD

In order to reduce the size of hydraulic motor and gearboxes required to produce both rotation and torsion, (ie: embracing both extremes of a speed/torque curve), hydraulic actuators are employed to produce torsion.

Load is applied by differentially operating the two actuators at either end of the rig which in turn act upon winding drums via steel cables thereby producing torque. The actuators are servo controlled, torsional load being applied by rotating the two winding drums at either end of the rig in opposite senses.

Load and positional control functions are available.

SIMULTANEOUS LOADING

All operating conditions are available simultaneously apart from torsion load and rotation of the hose for combined bending. Normally during rotation of the hose the winding drums, hence torsion facility, would not be in the drive chain. In order to engage the torsion facility hydraulically activated "dog clutches" would be employed to 'lock' the winding drums into their respective drive shafts. Simultaneously the rotation mode would be electrically interlocked to prevent inadvertent operation and the rotation drive gear boxes similarly de-clutched.

LOAD INDICATION

Either by direct reading load cells, (linear rams) or pressure inferred. All loads are displayed by meters on the console.

(d) SYSTEM PERFORMANCE

AXIAL LOADING SYSTEM

As per Design 1

BENDING SYSTEM

HYDRAULIC ACTUATORS

As per Design 1 excepting that the flow and power requirements are halved by only employing the system at one end of the rig structure.

ROTATING SYSTEM

This system is used only to rotate the hose for compound bending and not for torsional requirements.

HYDRAULIC ACTUATORS

The actuators for this system are hydraulic motors of the axial piston constant displacement type, one at each end of the hose. The motors drive through a gearbox at a ratio of 200:1 to the hose.

Each motor has a displacement of 4.04 in^3/rev.

OPERATING LOADS

The system has been designed to give an operating torque in excess of 2 ton ft at each end of the hose. This is the torque calculated to be sufficient to rotate the hose under maximum bending deflection assuming 70% energy recovery.

OPERATING PRESSURE

A system pressure of 1500 lbf/in^2 is used, this being sufficient to provide the torque required.

MAXIMUM FLOW

Reference to the hydraulic motor displacements quoted show the maximum flow required to be 44 g.p.m. per unit, ie: 88 g.p.m. total, including the effects of leakage etc.

Two pumps are provided to supply this flow.

POWER REQUIREMENTS

The hydraulic system is a constant pressure system running at 1500 lbf/in^2 with a peak flow of 88 g.p.m. Under peak flow conditions the power required is 104 H.P. Allowance being made for system efficiencies.

An electric motor of 150 H.P. is provided giving a contingency allowance of 50%.

TORSION SYSTEM

ACTUATORS

The torque is applied by four hydraulic actuators acting via cables on winding drums of approximately 2 ft diameter. The stroke of these cylinders is estimated at 18 in to allow for deflections of 360 deg. The effective area of each cylinder has been chosen to be 40 in^2.

OPERATING LOAD

The system has been designed to a maximum torque of 40 ton ft, the acceleration torque being provided by the natural spring force of the hose.

OPERATING PRESSURE

The hydraulic pressure corresponding to the maximum load is 1200 lbf/in^2.

MAXIMUM FLOW

The worst operating conditions are estimated to be a sinusoidal operation with a stroke of $\pm$ 9" and a period of 20 sec. These conditions give rise to a peak flow of .85 g.p.m.

Two pumps are provided to supply this flow.

POWER REQUIREMENTS

Under the conditions described assuming a load amplitude of $\pm$ 40 ton ft, the peak power requirement is calculated to be less than 60 H.P.

An electric motor of 75 H.P. is provided giving a contingency allowance of 25%.

(e) CONTROL

As per Design 1.

BUDGET ESTIMATE (1975 COSTS)

		£
1.	Main supporting structure	35.000
2.	Mechanical drives, hose mountings, trunions, gear boxes etc.	55.000
3.	Hydraulic power packs, hydraulic cylinders, motors, starters etc.	51.800
4.	Controls, instrumentation and cabling	33.000.
5.	Foundations, including piling, floor slab etc.	10.000
6.	Options:-	
	(a) Water supported.	
	Tank structure, seals, pumps, pipework etc.	(10.000)
	(b) Heavy Liquid Supported	
	Tank etc. as above plus 7,500 gal of suitable liquid.	(20.000)
	(c) Fluidised Bed Supported	
	Fluidised bed chamber, centrifugal blowers, dispersion grids, etc.	(45.000)
7.	Erection and commissioning	
	(a) + (b)	(20.000)
	(c)	(27.000)

TOTAL COST (DESIGN OPTION 2)	£
(a)	214.800
(b)	224.800
(c)	256.800

DESIGN 3 - HORIZONTAL CONFIGURATION

(HOSE MECHANICALLY SUPPORTED)

Figure (75)

It will be seen from the drawing that the design is very similar to that already described by design option (2). The main differences being:-

(a) The water tank/fluidising chamber is dispensed with and replaced by a series of free standing mobile support trolleys which articulate and move with the hose as it deflects.

(b) Because of the above, a minimal plan area is no longer considered of major significance, thus a bending function is provided at both ends of the rig.

(c) The application of torsional loading via the hydraulic motor/gear box chain described by Option 1 is re-introduced.

Because of the aforegoing the design will not generally be re-described in any detail.

SUPPORT SYSTEM

A chain of mobile trolleys incorporating either multiple rollers or an endless belt system (See Figure (76)) would be used in order to accommodate/support the hose in a horizontal plane. Centre height and diameter discrepancies resulting from the requirement to test various hose sizes would be accommodated by adjustment of the roller or belt system. Similarly length adjustments would be effected by the addition or subtraction of one or a number of unit trolleys.

In order for the 'trolley train' to articulate as the hose deflects, without constituting serious lateral loading on the hose, the trolleys will be fitted with variable direction rollers which will run on a specially prepared concrete or sheet-steel surface. (NB: The tracks on which the "axial loading" drive unit runs will be below floor level and covered with sliding steel shutters to accommodate the trolleys with the hose in a semi-straight configuration.

BUDGET ESTIMATE (1975 COSTS)

		£
1.	Main supporting structure	33.000
2.	Mechanical drives, hose mountings, trunions, gear boxes etc.	70.000
3.	Hydraulic power packs, hydraulic cylinders, motors, starters etc.	50.000
4.	Controls, instrumentation and cabling.	30.000
5.	Foundations including piling, floor slab, trolley surfaces etc.	11.000
6.	Trolleys including support system.	16.000
		209.000
7.	Installation and commissioning	19.000
	TOTAL COST (DESIGN OPTION 3)	209.000

DESIGN 4 - HOSE TESTED IN A SEMI VERTICAL MODE

Figure (77)

(a) STRUCTURE

It will be seen from the drawing that a quasi-vertical configuration has been adopted, and once again the "rotary bending concept" has been incorporated. The rig consists fundamentally of two 'head-stock' units between which the hose is located and allowed to deflect under a combination of self weight together with additional forces applied at the ends.

It will be seen from the drawing that the tilting movement at either end which is, of course, the movement required to produce a bending moment on the hose is produced by a straight forward arrangement of hydraulic cylinders. These act upon a tilting/rotating assembly mounted on trunion bearings within each headstock unit.

It will also be noted that each headstock unit is mounted on a track which enables their relative separation to be adjusted. The free standing unit provides adjustment in order to accommodate the various test hose lengths, whilst the actuator controlled unit provides the axial loading facility.

The rotation and torque functions are provided by hydraulic motors which act through gearboxes. All water circulation/cooling facilities etc are provided.

It may be seen from the drawing that beneath the rig track structure a pit is excavated. This forefills a number of functions:-

(a) Accomodates hose deflections in excess of that allowed by the height of the steel structure.

(b) By virtue of (a) reduces the structural height thus structural, building and associated foundation costs.

(c) Provides a safety feature, whereby the more dangerous elements of testing, ie: flexing at maximum curvature, are conducted with the hose semi-enclosed within the pit.

(d) Enables a modification of the hose weight to be effected, eg: by water flooding of the pit plus addition of floatation elements etc; thus adjusting the loading pattern.

(e) Provides a sump for the re-circulation of hose and hydraulic power pack cooling water.

It is assumed that the hose to be tested arrives at the rig on a suitable trailer having been previously fitted with the necessary adapter plates. The hose would then be lifted on slings by a mobile crane and suspended in position between the two headstocks. Bolting of the hose into position would then proceed much on the same lines as described for designs (2/3). Additionally the head stocks could be tilted in order to accommodate angular deflection or 'droop' of the hose flanges resulting from the lift configuration.

(b) LOAD GENERATION

Generally the load generation characteristics comply with those previously described for design 1. The rotation/torsion modes are however slightly different and will be described.

TORSION

The requirement for torsion generation is limited to one end of the rig only, the opposite end being effectively locked using a hydraulically inserted pin arrangement. Torsion is achieved by employing two axial piston motors driving in parallel through a common 200:1 reduction gearbox. One motor is of fixed capacity whilst the other is a remote pressure controlled variable capacity unit. Pressure energisation of the appropriate pilot valve would cause the variable capacity unit to increase giving a combined capacity, capable of meeting the torsional load required.

ROTATION

At the opposite end of the rig to the above is also mounted a 200:1 reduction gearbox, this time driven by a single fixed capacity motor of the type described above.

In the rotational mode the capacity of the variable capacity motor would be reduced and the inlet/outlet ports connected together permitting oil to circulate freely within the pump casing, ie: the motor would be allowed to "free-wheel".

At the same time flow would continue to both fixed capacity motors, at either end of the rig, permitting rotation of both gearboxes. The system would form part of a servo-position control loop, and may be controlled to maintain the two ends of the hose in synchronism.

The entire system would be interlocked in order to prevent inadvertant selection of both rotational and torsional modes of operation.

(c) SYSTEM PERFORMANCE

AXIAL)
) Similar to previous designs.
BENDING)

ROTATION/TORSION

The system has been designed to give an operating torque in excess of 2 ton ft at either end of the hose at a maximum speed of 12 rev/min in the rotational mode and a torque of 50 ton ft at 3 cycles/min in the torsion mode.

OPERATING PRESSURES

(a) ROTATION ONLY

A system pressure of 1500 lbf/in^2 is used this being sufficient to provide the torque required (for rotation) using a motor capacity of 4.15 in^3/rev.

(b) ROTATION AND TORSION

A system pressure of 4000 lbf/in^2 is employed in order to generate the torque required (for torsion) using a combined capacity of $4.15\ in^3 + 10\ in^3 = 14.15\ in^3$/rev.

MAXIMUM FLOW

(a) ROTATION ONLY

Maximum flow is calculated at 44.0 g.p.m. (A single pump is provided for this flow).

(b) ROTATION AND TORSION

Maximum flow is calculated at 43.0 g.p.m. (A single pump is provided for this flow).

POWER REQUIREMENTS

(a) ROTATION ONLY

Under peak flow conditions of 44.0 g.p.m. the power required is 52 H.P. An electric motor of 60 H.P. is provided giving a contingency of 15%.

(b) ROTATION AND TORSION

Under peak flow/torque conditions the power required is 137 H.P. An electric motor of 150 H.P. is provided giving a contingency of 10%.

BUDGET ESTIMATE (1976 COSTS)

		£
1.	Main supporting structure	40.000
2.	Mechanical drives, hose mountings, trunions, gear boxes etc.	73.000
3.	Hydraulic Power Packs, cylinders, motors etc.	34.000
4.	Cables, controls, instrumentation etc.	26.000
5.	Foundations including, piling, floor slab, pit etc.	12.000
		188.000
6.	Installation and commissioning	21.000
	TOTAL COST (DESIGN OPTION 4)	206.000

SUMMARY OF DESIGNS

(1) FINANCIAL

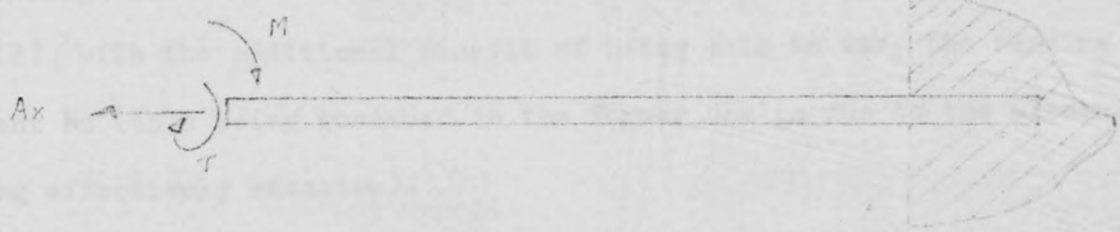
DESIGN	COST	% DIFFERENTIAL (BASED ON LOWEST COST)
1	£328.830	+ 60%
2 (a)	£214.800	+ 4%
(b)	£224.800	+ 9%
(c)	£256.800	+ 25%
3	£209.000	+ 2%
4	£206.000	0%

Based upon the above, brief financial statement, it was concluded that options 1 and 2(c) carried such a cost penalty that they were discounted without recourse to additional technical considerations.

(2) CONCEPTIAL

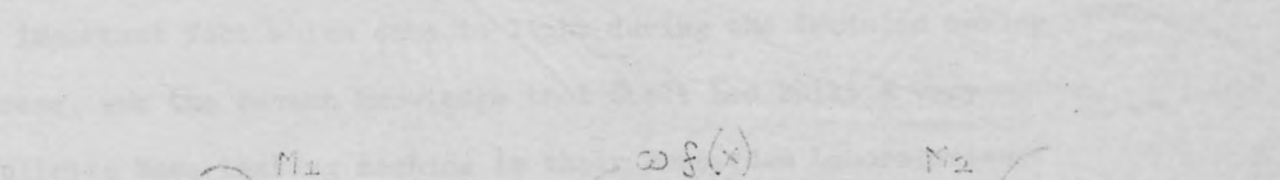
The basic type of loading available from each test method may be approximated as:-

DESIGN 2 (a) & (b)



DESIGN 3

DESIGN 4



(NB: It should be appreciated that the above are only crude ideal approximations, which neglect the effects; in cases 2 and 3, of the support media).

It may be seen that the loading pattern capability becomes increasingly complex as one progresses through the designs in chronological order. Design (3) obviously has all of the capabilities of (2), with the additional benefit of being able to vary the bending moment Hz (this being governed in the former design due to the element being effectively encastre).

Design (4), it may be seen offers far greater potential in combined loading than the previous designs, however its ability to conduct 'pure' tests is severely limited by an inability to totally remove the 'self-weight' element of loading, thus the latter in combination with additionally desired loading always results in a complex loading situation. (NB: This would be un-important were the test hose homogeneous and isotropic, however because of the, as yet, unknown inter-relationship between intrinsic properties such complexities might lead to difficulties in analysis).

One important fact which came to light during the decision making process, was the recent knowledge that Shell had built a very simplistic hose testing machine in their Amsterdam Laboratories; (commissioned late 1975). This rig (see Figures (78)-(80)) could be crudely approximated to design option (3).

"The test hose, whilst supported in a horizontal plane, on mechanical supports, could be subjected to a symmetrical bend configuration. In this condition; rotation of the hose, driven from one end only, could be produced in order to simulate alternating flexing.

No axial loading, Torsional, differential bending or pressurisation facilities were available."

Whilst very simplistic, the Shell rig was the first of its type and thus should be commended. However the one feature which the Shell rig did exhibit, from the limited information available to Dunlop, which was of considerable benefit to our own design study was the effect that the supports appeared to have on the test hose. It may be seen from Figure (81) that about each mechanical support bulges developed, these occurred very rapidly after only a short period of testing. Consequently the test hose rapidly took on an appearance likened to a 'string of sausages' with each support at a "necked section".

Whilst it was not readily apparent why the supports should have such a significant effect it was felt unwise to continue with our own "mechanically supported design" whilst such uncertainty existed.

(NB: since the time denoted Shell are known to have tried various modifications to their support system with little if any improvement).

The design options were thus reduced to 2 (a)/(b) and (4).

During the final design selection process option 2 (b) was deleted because of:-

(a) The cost of the necessary fluid.

(b) Storage facilities required for the fluid whilst tank sections were changed - Not included in original budget estimate.

(c) Most of the fluids available were found to be damaging to conventional hose cover materials. (ie: test hoses would require a specially made cover compound).

Finally option 2 (a) was similarly deleted:-

(1) Restriction of method to the testing of floating hoses or buoyancy supported submarine hoses.

(2) The 'wet' environment making direct hose observations and measurements etc. difficult.

(3) Restricted test modes available without increasing the rig plan area, ie: tank size.

(4) Testing capabilities not as comprehensive as option (4).

(5) Complex/time consuming hose change-out procedure - particularly for varying hose lengths (ie: adding or subtracting tank sections).

Design Option (4) was thus finally selected.

XI.5 DESIGN DETAILS

The design option selected, namely option (4), was refined into a detailed design specification/drawings and the elements put out to tender.

STRUCTURAL DESIGN

Based upon the initial conceptial design study a detailed design project was undertaken, in conjunction with the Division's Engineering/ Drawing Office, resulting in the production of a set of manufacturing drawings. These eventually formed the basis of a detailed specification document against which tenders were invited and the construction work eventually sub-contracted.

The results of the design study, calculations, drawings etc are included as (18). By virtue of the rig structure involved it was found necessary to design on a basis of structural stiffness rather than simple stressing methods, the former being the critical criteria.

Although the design study is enclosed as an appendix it is worthy to note at this stage a number of design modifications with respect to the initial design concept.

Referring to Figure (82) and comparing it against Figure (77) (previous section) the following may be noted as marginal conceptial modifications:-

(a) One head stock has been rigidly fixed, the other accommodates both movement resulting from axial loading and the accommodation of varying hose length. This is performed quite simply by using the rear (axial loading) hydraulic actuator mounting carriage to accommodate varying hose lengths whilst the actuator/headstock is used to accommodate change due to axial load.

Two cost benefits resulted from the above change:-

(1) Simplification of the rail/slide arrangement(s).

(2) No ancillary winch equipment is required to reposition either headstock. (ie: It may be seen that the actuator carriage/headstock arrangement can be "walked" along the bed in a crab-type mode).

(b) The application angle of the axial ram has been reduced to a horizontal elevation in order to produce a more efficient loading system.

(c) By installing the torsion locking mechanism (refer to drawing No:GTS 1486 Referenc(18)) on the, (hose side) output side of the respective gearbox it has been possible to install a lower specification gearbox (ie: Lower torque capacity) than that at the torque drive headstock.

HYDRAULIC SYSTEM DESIGN

Again based upon the initial design study a draft specification was issued to a number of companies for budget quotations. Based upon the results of these quotations a detailed study was undertaken in conjunction with Keelavite Hydraulics Limited, and a final design specification compiled.

The final design specification is included as Referenc(19).

Basic elements of the design are as follows:-

HYDRAULIC EQUIPMENT

For the purpose of clarity, it is better to consider the equipment in three groups:-

(a) Hydraulic Power Supplies

(b) Rotary Actuators

(c) Linear Actuators

(a) HYDRAULIC POWER SUPPLIES

(1) END BENDING AND AXIAL LOAD - SEE FIGURE (83)

The components would be mounted within a steel structure and all pipe work up to the outlet hand valves completed. Operation of the hand valves would enable the operator to run the two test function separately or together at a reduced overall specification. Control of the main pressure unload valve would be provided from the start and safety panel mounted in the electronic equipment. For safety purposes the following fault conditions would remove the electrical power, low oil level, high oil temperature, loss of main pressure. Each pump would be individually protected by its own relief and unload valve.

The unit would be capable of delivering mineral oil at a flow rate of 44 gpm and a pressure of 3000 psi. The flow could be selected in four equal multiples of 11 gpm.

(2) HOSE ROTATION AND TORSIONAL LOAD - SEE FIGURES (84) & (85)

In both cases the same basic components are used. The main difference being the installed horse power. The prime movement is supplied by electric motors. Variable capacity, electro hydraulically controlled pumps would be used. The pumps are boost fed by fixed capacity gear pumps working at low pressure. The system would be electrically interlocked, via pressure switches such that the main pump could not, under normal conditions, be started unless boost supply pressure was present. Back to back high pressure relief valves would be fitted to prevent excessive pressures being developed in any one of the two lines. Failure of boost supply, main supply, temperature control, oil level would cause electrical power to be removed.

The circuit components would all be mounted on a welded steel structure and piped as per circuit. All outlet and return lines be terminated in connection consistent with pipe sizes required to give the correct oil velocities. The oil level and temperature switches would be mounted on the oil reservoir. All necessary adjustable controls and visual indicators would be placed to give ease of access.

(b) ROTARY ACTUATORS

(1) COMBINED HOSE ROTATION & HOSE TORSION DRIVE

The requirement for this dual function is required at one end of the test hose only. The combined action would be achieved by employing two axial piston motors in parallel, one fixed capacity unit and one remote pressure controlled variable capacity unit (refer to Figure (84)). Energisation of the pilot valve would cause the variable capacity unit to increase giving a combined capacity, capable of meeting the torsional load torque required. The same pilot valve would simultaneously operate a five port directional valve, the energised pilot valve placing the direction valve spool in a position permitting oil flow to the variable capacity unit. The back to back non return valves enable oil to be fed to the motor when it is not in use but being driven by the fixed unit.

(2) HOSE ROTATION

For this requirement the pilot valve seolenoid is de-energised. The variable capacity reduces, the directional valve connects inlet and outlet ports together permitting the oil to circulate freely since the fixed unit will be driving it as a pump via the gearbox common shaft. To cater for oil losses during this condition of operation oil will be fed to the variable unit from the boost supply. As an added precaution a back to back relief valve would be connected across the ports. At the same time the feed pipes to the unit would be blocked. At the other end of the hose, rotation would be produced using one fixed capacity unit fed by a variable capacity pump.
(Refer to Figure (85)).

(c) LINEAR ACTUATORS

All three actuators would be double rod, double acting rods centre trunnion mounting facilities. Each end of the stroke would be protected by hydraulic cushioning to reduce possibility of damage occurring. The load rod of each would be fitted with a load transducer providing electrical signals proportional to applied load. To provide linear displacement control rotary inductive transducers driven by a linear/rotary instrument gearboxes would be employed. Drive to the gearboxes would be provided by attaching drive wires to each rod end and wrapping it around the input pulleys. The gearboxes would be mounted on the cylinder bodies. Associated with each cylinder assembly would be a servo valve and micro filter assembly. The assemblies would be mounted adjacent to their respective cylinders and coupled via flexible hoses.

CONTROL FUNCTION DESCRIPTION

(a) LINEAR

(i) DISPLACEMENT

(ii) LOAD

(b) ROTARY

(i) TORQUE

(ii) DISPLACEMENT

(a)(i) DISPLACEMENT

The control and command modes would be selected prior to applying hydraulic power. A rotary inductive displacement transducer energised by a 400 Hz carrier supply, would via a 400 Hz demodulator produce a D.C. feedback voltage proportional to cylinder position. This signal would be compared with the command and the difference fed to the servo valve drive amplifier (refer to Figure (86)) as an instruction. The cylinder would move in response until the difference between the two signals disappeared, at which stage the cylinder would stop.
The output signals from the demodulators would also be fed to level detectors which could be set to trip and stop system if desired stroke/angle were to be exceeded.
Indication of stroke/angle would be provided on an analogue meter.

(a)(ii) LOAD

Proportional control of load would be achieved in virtually the same manner. The difference being that a transducer mounted on the cylinder rod, between the cylinder and load, together with a high stability, strain gauge amplifier would produce an electrical feedback signal proportional to the load being applied.

The feedback signal apart from performing a control function would be used to drive an analogue meter to provide indication and a level detector to give system protection.

Three term controllers (ie: Integral, Differential and Proportional gains) would be used to optimise the control loop transfer function.

(b)(i) TORQUE

Across both sets of hydraulic motors would be connected an inductive pressure transducer, energised by a 4KHz carrier generator. The D.C. output signal from the associated 4KHz demodulator would be proportional to the pressure developed across the motors. It would therefore be proportional to torque developed and provide the torque feedback signal. Torque control is required on torsion only hence a servo control loop around this end only, has been provided.

It would be possible to set the mean level required and add a cyclic signal.

Indication of the torque value would be provided and level detectors used to protect the system.

The error signal (difference between command and feedback) signal would instruct the servo valve to drive the hydraulic motors until the torque reaction developed by the hose, reduce the electrical error to a finite value.

The pressure transducer connected across the other hydraulic motors would be used in conjunction with the control transducer to measure the torque differential developed across the hose and use the signal for indication and protection purposes (refer to Figure (87)).

(b)(ii) DISPLACEMENT

To provide rotation of the test hose about its axis at selected speeds we propose that relative displacement of hose ends be used as a control parameter. Towards achieving this, control synchros would be used.
A velocity loop would be closed around one hydraulic transmission drive enabling the required speed to be set. Indication of hose speed (R.P.M.) and displacement (degrees) would be provided at each hose end.
As the rotation started the electrical signal balance in the synchro-chain would be disturbed. The receiving synchro would develop via a 400 c/s demodulator a D.C. signal proportional to differential displacement. This signal would act as a command to the other hydraulic transmission drive which would rotate to restore the synchronisation between ends. Basically the end under velocity control would act as the command and the other end as a slave drive. The differential synchro would enable the hose ends to be set between 0-180 degrees out of alignment and retain relationships while rotating.
Indication of end phase relationship would be indicated in degrees, two separate analogue meters. To minimise risk of damage to hose, while rotating two level detectors would be employed. One would accept the differential torque signal and stop system if maximum acceptable value were exceeded. The other would compare the differential end displacement in terms of voltage and stop the system if the preset acceptable valve was exceeded.
The three term controllers would permit servo performance optimisation.

XI.6 INSTALLATION/COMMISSIONING

Naturally if one were to describe the total installation and commissioning process, the bulk reportage would greatly exceed the total size of this thesis. In order to overcome this problem it is intended to only briefly summarise these aspects in the text, whilst for greater detail, reference should be made to the appropriate installation and commissioning report; (reference 15).

Installation

Figures (88) through (96) show various stages throughout the test rig installation progress. Whilst many minor problems were encountered during this period, none of these were totally unexpected considering the type of 'heavy' engineering work involved.

One of the more difficult problems to solve on a practical basis was a method of getting the necessary supplies, (electrical and hydraulic), to the various rams and motors situated on the moving structure of the rig. Because of the hydraulic flow rates and pressures involved the prospect of accommodation long lengths of rather bulky flexible hose was daunting. In order to overcome this problem it was initially proposed that short flexibles could be employed in conjunction with a number of self-sealing 'quick connectors', 'tapped' into the supply lines etc at strategic points. (This applied particularly in relation to infrequent movements of the rig structure in order to accommodate various test hose lengths). However, a brief feasibility study into this concept proved that:-

a) Connectors were extremely difficult to make/break, frequently involving fluid loss and/or air inclusion.

b) Costs of such connectors were also found to be extremely expensive. Various alternate options, including the use of reeling or winding drums, were considered. The final solution adopted, however, (see Fig. 92), used a double catenary layout to provide the necessary mobility.

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The above example is used merely as an indication of the type of problem encountered. Many such problems were related to changes in rig siting and the building characteristics which were not finalised to a late stage in the project.

Probably the greatest problem during both manufacturer and installation processes was the co-ordination of the various firms and personnel involved. Ensuring that the civil, mechanical, hydraulic and electrical component aspects of the project remained on schedule, so as to correctly'slot together' during the installation phase proved to be the most exhausting and exasperating feature of the entire project. Much use was made of critical path analysis, (Ref 15), during this time.

Commissioning

Generally commissioning proceeded satisfactorily although some problems were experienced.

Static system response in terms of load/displacement indication was initially calibrated using equipment test certificates in conjunction with an appropriate input device:

eg Load cells were replaced by a standard reference network of appropriate gain, and the control system(s) forward path gain appropriately adjusted in conjunction with the load cell calibration curves. Similarly the pressure transducers were set up using a 'dead weight tester', the appropriate torque indication being calculated from a knowledge of motor and gearbox ratios and efficiences.

Displacement systems were calibrated by using a dial test gauge attached to the cylinder rods. Angular displacement was similarly calibrated in conjunction with a knowledge of the appropriate moment arms.

The appropriate gain values for the feed back loops were initially calculated on the basis of theoretically predicted inertia values for both rig and test hose. (NB the worst case condition involving the largest test hose was chosen for these calculations, reference (16)). Final adjustment was then performed imperically in order to finally trim the systems. (Although the specification

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nominally required sinusoidal operation at a set frequency range, ramp and square wave options were incorporated into the frequency generator design. In order to prevent equipment damage, under such conditions, the systems were optimised under square wave maximum frequency conditions. This was performed by increasing the negative path gain until instability was produced, then by reducing the gain by 5%). Within the frequency range specified, system sensitivity was found to be more than adequate before any indication of instability occurred.

Whilst no inherent problems were found in the design of the axial loading and bending moment application equipment, initial problems were experienced in relation to the rotation and torsional systems:-

a) It was found that as rotational speed was increased, log increased until 'pull-out' occurred at approximately 6 rev/min. This could be improved by increasing the overall system gain, however, it was found impossible to obtain synchronism at 12 rev/min whilst maintaining stability below 4 rev/min. (It may be remembered, see figure (87), that the operational mode of the above system was to inject a velocity signal into one head-stock unit rotation chain (master) and then to slave the opposing unit from the output of the tacho generator of the former system).

From the following calculation it appeared inevitable that such a phenomena would occur:-

Natural frequency of the system = 0.5 Hz = 3.14 rad/sec.

Permissible loop gain for gain margin of 12 dbs = 1 sec^{-1}

$$12 \text{ rev/min} = \frac{12 \times 2}{60} = 1.2 \text{ rad/sec}$$

Velosity lag = 1.2 = 68° @ 12 rev/min

It thus proved impossible to obtain synchronism at 12 rev/min with stability below 4 rev/min.

In order to overcome this problem a tacho feedforward path was introduced into the system. That is a proportionate value of the master control

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signal is simultaneously injected into the slave system. This modific-ation was found to be successful, and following system optimisation the system was left with the following performance.

2 rev/min	zero lag)	NB At constant speed, the lag can be adjusted
4 rev/min	1° lag)	to zero or any value by use of the differen-
12 rev/min	5° lag)	cial syncro.

b) Calibration of the torsion measuring system proved to be a problem. Initially the pressure transducers employed had been calibrated with respect to gearbox torque output, via a knowledge of the characteristics of both hydraulic motors and gearbox losses. (NB It was realised during the early design stages that inferring torque values via monitoring pressure would not prove to be a very accurate system. However, because of the extremely high cost penalty to be incurred by directly measuring such high torque under rotating conditions, pressure measurement was considered the only available option).

The problem initially determined was that meter indicated torque, increased with speed, whilst no external load was placed on the gearboxes; see figure (95). In other words the nett inefficiency of the hydraulic - mechanical system was found to increase with speed. Further tests were conducted by loading the gearboxes employing a simple dynamometer break. Indicated torque/speed characteristics were thus obtained for a range of gearbox output torques, see figure (96). These tests proved that the problem arose directly from the no-load characteristics and were not further compounded. It was seem that by subtracting the no-load curves from each of the load curves, unity linearity was obtained between indicated and measured torque, figure (97).

On the basis of the above it was proposed that the meters/control system be re-arranged to indicate not torque inferred from differencial pressure, but torque available externally, by modifying such as a function of speed. Also shown on figure (95) are indicated torque values averaged either side of zero and straight lines joining zero to the curves at 9 rev/min.

contd/

These lines may be regarded as signals derived from the respective tachos, which scaled as shown and subtracted from the pressure signals, would enable the actual torque available to be indicated with reasonable accuracy.

The modified signals, if subtracted from one another, and fed to the differential torque meter, would then allow the torque available to produce torsion to be more accurately displayed.

The system was thus modified giving the characteristics as shown below:-

Speed	Master Error tons ft	Slave Error tons ft	Nett Error tons ft
0	0	0	0
2	+ 0.47	+ 1.01	+ 0.53
4	+ 0.48	+ 0.87	+ 0.39
6	+ 0.30	+ 0.54	+ 0.24
8	+ 0.12	+ 0.19	+ 0.07
10	- 0.11	- 0.20	+ 0.09
12	- 0.24	- 0.63	+ 0.39

The previous two examples give an indication of the type of problem encountered during commissioning, how they were tackled and eventually solved.

XI.7 TEST PROGRAMME/EVALUATION

The general objectives of the test rig 'total' programme may be broadly categorised as:-

(a) The evaluation of characteristic hose properties and their variation with mechanical fatigue.

(b) Comparative evaluation of alternate designs and construction techniques.

(c) Evaluation of competitor's hoses and design concepts.

Resulting in:-

(d) The improvement of current hose technology (both design and manufacture).

(e) The ability to approximately predict ultimate hose failure.

In order to achieve the above objectives there appeared a number of different lines of approach which might be adopted:-

(1) Test a representative range of existing standard product designs in order to ascertain a knowledge of their present basic characteristics and limitations.

(2) Concentrate on the testing and improvement of one particular type of hose; (eg: buoy connection hose).

(3) Conduct a comparative design evaluation in relation to a series of specially prepared specimens (eg: Textile reinforced v steel reinforced etc).

(4) Test hoses in response to customer/marketing pressure, (ie: new developments etc).

(5) Evaluate hoses returned (retired) from service.

All of the above possibilities were put forward by various elements of the Division, none could be regarded as completely singular in application or effect. Each represented the various stand points of departmental factions within the Company.

Much discussion/debate ensued, most not upon technical grounds, as to which was the correct approach. Eventually logic prevailed resulting in Option (1) being selected as the correct approach, (certainly with regard to the first 12-18 months of testing). After such an initial test period it was considered that the future test programme would, in the main, be dictated by the results obtained. Almost certainly such future tests would embrace some, if not all, of the points made above.

During the initial period it is intended that the hoses indicated are tested:-

TEST PROGRAMME SUMMARY

		HOSE BORE			NOTES
TYPE		16"	20"	24"	COMMON FEATURES
5152/1	Std. Sub. (Textile)	+ ✓ [2]	✓ (9)		) -VE Pitch Ø Ø of helices
*5155/1	Selflote (Textile)	✓ (1)	✓ [10]		)
5156/1	Std. Sub. (wire)	✓ (3)	✓ [8]	✓ (5)	) -VE Pitch Ø Ø of helices
*5157/1	Selflote (wire)	✓ [4]	✓ (7)	+ ✓ [6]	)
5156/2	Samp.Sub. (wire)	✓ [14]	✓ [16]	✓ (11)	) Difference in pitch of helices only
*5157/2	Samp.Flt. (wire)	✓ (13)	✓ (15)	+ ✓ [12]	)

NB:

1. *Selflote hoses to be tested in carcase form only.

2. All hoses of 30ft length.

3. Number in parenthesis indicates test order.

4. ( ) indicates static tests only (non destructive)

5. [ ] indicates static and dynamic tests (ie: destructive tests).

6. + indicates hoses that are available (inspection rejects booked to test rig.

TOTALS:

DESTRUCTIVE - 8

NON DESTRUCTIVE - 8

It may be seen from this table that a total of sixteen tests are envisaged during the initial testing period. The balance struck between destructive tests (ie: static + dynamic) and non destructive (ie: static only) at eight to each option, and the distribution noted is deemed satisfactory to maximise the information/(Cost + time) ratio. It is envisaged that hoses required for non destructive tests could be taken from, and then normally returned to customers orders.

Reviewing the experience gained from previous test methods, the following time scale predictions would appear to be of an appropriate order:-

(a)	Rig/hose preparation	- 4 days
(b)	Non destructive tests (static)	- 5 days
(c)	Destructive tests (Dynamic) (1 day = 10 hours)	- 24 days

Table (16) indicates the proposed programme against a time scale. This suggests a total requirement of 336 days in order to complete the programme outlined.

On a basis of a 46 week year, (initially at least each week consisting of 6 x 10 hour shifts), a total period of 414 days would be available. Assuring 5% machine down time for unscheduled maintenance etc. (ie: 20 days) a float time of 58 days is available. This period would nominally be used for the testing of service return hoses (ie: static tests for property correlations, approx. 10 days/test), and or repetition of tests previously conducted for statistical significance purposes.

Whilst work is undertaken by the test rig it is proposed to continue the simple bend tests etc., which are already conducted by Technical Department. It is considered that these tests should provide a pointer to the repeatability of test results; (ie: if initial static properties exhibit a high degree of repeatability it would not be unreasonable to see this reflected in the general test results). To this end initial bend test results appear promising in the above respect.

It is considered that towards the conclusion of this initial test period that:-

(a) Basic hose properties for a range of hoses will be known.

(b) Some information will be available as to the above variation with respect to fatigue.

(c) Some degree of correlation between property degradation and time will be established.

(d) A predominant form or type of failure may be identified.

Upon completion of the above work the results achieved should serve as the initiator of further research. The detection of a predominant type of failure for example, (eg: helical wire displacement) if substantiated by service failure analysis, would automatically indicate the need for redesign and test of that feature.

In many ways the hoses selected for test form an extremely broad basis for such work. There is, by nature of the constructions, a considerable number of correlations which could be made with respect to:-

(a) Fabric reinforcement

(b) Wire cord reinforcement

(c) Helical wire involvement

Determination of basic hose properties should also enable a systematic approach to fundamental design changes which might be necessary to meet specified requirements. Such hoses would form an integral part of the second stage of the test rig programme. At this stage it would be in order to consider, in outline, the test specification for the first stage tests:-

TEST PROGRAMME:

Nominally any test form would consist of an initial series of static tests, (non destructive), following by a comprehensive programme of dynamic tests, (destructive), ultimately to failure. Dynamic tests would so be arranged that a comprehensive range of service related load cycles (in terms of torsion, axial load, bending etc), of predetermined duration would be repeated 'en bloc' until failure occurred. Interspersed between each block of cyclic tests would be a repeat of the initial static tests. These would be used to assess the degradation of the initial hose properties in relation to repeated fatigue cycling.

STATIC TESTS

Three major parameters are considered adequate in order to quantify basic hose properties:-

(a) Bending stiffness

(b) Torsional stiffness

(c) Axial stiffness

The above however cannot be regarded as constants, they are only static for any specified loading condition. It is therefore considered essential that the variation of the above parameters are determined with respect to:-

(d) Load magnitude

(e) Combined loading

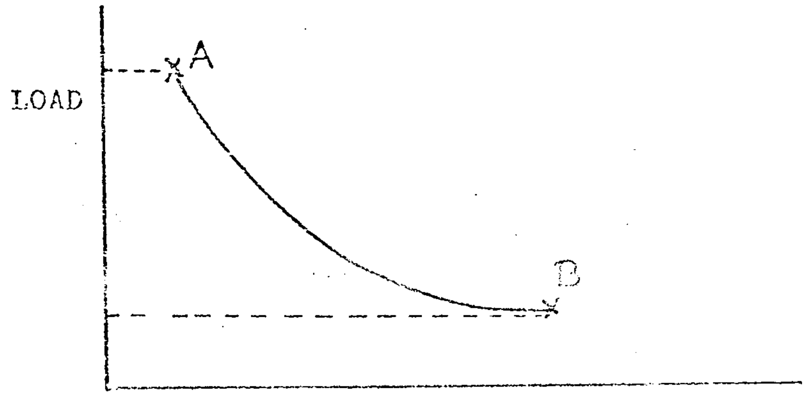
(f) Internal pressure

(g) Radius of curvature, axial extension, twist etc.

It may already be seen that combinations of the above parameters result in an enormous number of interactions. Such that these combinations may be studied, it becomes necessary to select certain relationships, on a statistical basis, such that the entire field may be projected upon a relatively small measurement sample. A static test programme designed to serve just this purpose is indicated by Table (17). It may be noted that various combinations of bending moment, torsional load, axial load and internal pressure are used in order to define the necessary relationships.

DYNAMIC TESTS:

The prime directive of dynamic tests is the evaluation of mechanical fatigue, with respect to hose properties and ultimate failure:-



Cycles to failure (log scale).

Obviously if the load applied is sufficiently great a single application (ie static load) would be sufficient to destroy the hose, (ie Point A). Equally at some relatively minor loading the hose may effectively never fail, (ie Point B). Although of interest, particularly the higher limit, these are not generally applicable to practical situations where the loads imposed fall between these extremes.
Design of the dynamic test programme must obviously be designed to establish such general concepts as the above fatigue curve, (probably different with respect to loading mode, hose type etc). Also to establish a similar measurable quantity (ie hose property) which might establish the life status of a hose (ie life period consumed), thereby enabling some form of linear damage hypothesis to be established.
At this stage of the development of the test programme, it is extremely difficult to quantify in exact terms the necessary tests required. However, it may be stated that the initial tests will be very dependent upon the data collected as a result of the Forcados study and more recently model testing. Such data provides the necessary load values

and application frequency, although not the probability of such a load occurring. The latter requires probability data relating to environmental conditions prevailing at typical SBM installations. A limited degree of this information is already available via wave histogram data.

With such data, each mode of loading may be described by a three dimensional histogram, (ie: load magnitude, application frequency, probability), as typically shown by Figure (98). In general principle the test rig input would be derived from such a source, computation of the nodal points would provide the necessary classification indexes, and the Z component the proportional test duration.

Initially each mode of loading (ie: load type) would be run consecutively rather than concurrently. This is in order that basic initial relationships regarding destinct effects of each type of load may be isolated. Eventually simultaneous application or combined dynamic loading would be utilised.

TEST EVALUATION

Throughout the test period loads and deflections will constantly be monitored and used to detect changes in the hose structure properties as each test proceeds. . In addition such techniques as X Ray, thermograph and acoustic emission will be applied in an attempt to examine the changes that are occurring internal to the hose structure:-

X RAY:

Referencing against initial plates. This technique may prove of value in determining displacement of such items as helical wires, fittings, 'binding in' wires etc.

THERMOGRAPHY:

Examination of thermal gradients, as a result of dynamic tests, may serve to indicate areas of significant cyclic stress loading (ie: hysteresis effects), hence bond breakdown.

ACOUSTIC EMISSION:

Again reference against an initial 'pattern' could serve to indicate the presence and/or growth of voids or delaminations.

An attempt has been made to develop a broad outline as to a proposed test strategy in relation to an initial period of twelve to eighteen months.

It is accepted that this programme is by no means complete or comprehensive and will certainly require modification in the light of test results.

XI.8 INITIAL TEST RESULTS

Hose type and number: 24 inch Dunlop double-helix submarine hose no. 42272, 30 feet long.

Static tests

Vacuum test	: in this test, which was carried out before the dynamic tests, no irregularities were observed.
Stiffnesses	: see Table 14

Dynamic tests (Fig.)

a = 10°; cycles 1 - 5 000	: change in the external diameter (Fig. 99 and 100); no change in power consumption.
a = 20°; cycles 5 000 - 10 000	: change in the external diameter, see (Figs 101 and 102); change in power consumption (Fig. 104). A sudden change occurred between total numbers of bending cycles of 8298 and 9052 (i.e. between 3298 and 4052 bending cycles at 20°).
a = 20°; cycles 10 000 - 15 000	: change in external diameter (Fig 103); the power consumption decreased continuously (Fig. 104).
a = 30°; cycles 15 000 - 18 404	: the external diameters were not measured. The power consumption decreased considerably during this part of the test (Fig 105). After a total number of 16 057 bending cycle an external hose temperature of 35°C was measured at a distance of 3.6 metres from the driven end of the hose. The increase in temperature is an indication of friction between loose materials and reinforcements in the hose. At 17 789 bending

contd/

cycles the external hose temperature at a distance of 3.6 metres from the driven end had risen to 52°C. Delamination was observed there and final failure occurred at a total number of 18 404 bending cycles.

The final failure is shown in Figs. 106 - 108.

Pressure testing

Pressurization of the hose in a straight position was carried out after each series of 5000 bending cycles. The results are given in Fig. 114.

Destructive tests

Peel test : (adhesion of rubber to the reinforcement; see Fig. 109 and ASTM D-413).

Rubber to steel cord : 10 N/mm

Rubber to nylon (between steel cord and inner helix): 3 N/mm

Tensile test : (carried out on a cylindrical core, 0.04 m in diameter, taken in the radial direction).

Pulling speed: 1 inch/minute.

The results are listed in Table 15.

DISCUSSION OF RESULTS

From the dynamic tests it may be concluded that deformation of the helices (observed as bulging of the hose) introduces stresses into the carcass. At locations where turns of the helices increase in diameter, radial tensile stresses occur in the reinforcement layers. These stresses are highest between the steel cords and the inner helix (Fig. 110).

When the hose is rotated on the rig in a bent condition then, in addition to the aforementioned stresses, alternating radial tensile stresses occur. This is because during bending the carcass contracts at the convex side and extends at the concave side of the hose while the helix maintains its position.

The difference in behaviour between helix and steel cord under bending is due to the difference in winding angle.

As for the hose tested the adhesion was lower for rubber to nylon than for rubber to steel cord, delamination started, at 3.60 m from the driven end, at the nylon/rubber interface. The gradual increase in delamination caused a loss of bond between carcass and helices. The carcass thus could contract there without being restricted by the steel helix, flexibility increased and the hose finally kinked.

Between the individual wires of the steel cord capillaries were observed, which in case of damage to the hose might enable seawater and/or oil to penetrate into the carcass (see Figs. 111 and 112). It can also be seen from Figs. 111 and 112 that the thickness of the rubber between the steel cord layers was rather small, namely in the 0.1 mm range.

From the hose model (Figs 115-120) it is clear that subjecting the hose to bending, torsion and tension will result in relative movements of the steel cord reinforcements. Observation of a crossing of two steel cords reveals that under conditions of bending and tension the angle between the crossing steel cords changes, while under bending the location of the crossing also shifts along the wires, which causes a tangential displacement of the crossing. The thinner the layer of rubber that holds the steel cords together, the smaller the beneficial effect of the elasticity of the rubber; the connection will become stiff and the rubber-to-steel bond in the carcass will be unnecessarily highly loaded. With a larger thickness of the rubber, stresses in the rubber and, hence, the loading of the rubber-to-reinforcement bonds will be lower, and hose life will therefore be longer.

Criteria for deterioration of the hose

Pressure testing with measurement of temporary elongation, as required by the SPM Forum Hose Standards, yielded no information on deterioration of the hose during dynamic testing (see Fig 114). Permanent elongation, however, did indicate deterioration.

contd/..

XI.9 FINANCIAL EFFECT

Marketing Statements

"Our planned sales of offshore oil hose are:-

1976	£6,659.000	market share 49%
1977	£7,371.000	market share 50%
1978	£8,159.000	market share 51%

These figures assume the availability of a dynamic testing facility and without it we must face the prospect of at least 5% of our market share, being lost in 1976, 7½% in 1977 and 10% in 1978 being lost to competition. This is because by that time our credibility in the market place will have fallen to a very low ebb.

Sizes of offshore hose are increasing and many terminals are being designed to incorporate a 30" diameter hose in the future. It is of vital importance from an economic and ecological point of view that this hose will be reliable in service as the hose industry cannot afford similar problems with 30" hose as those encountered with 24".

Additionally, sales over the next few years will be more and more to the developed nations who will be having to import oil for economic survival. These nations are, of course, far more pollution conscious than the developing nations and consequently our ability to prove and demonstrate reliability will be essential".

1. Market Results

In the Marketing statements the results of not having the Test Rig are stated as shown below:-

		1976 £		1977 £		1978 £
Total Product Turnover		6,659,000		7,371,000		8,159,000
Proportion Lost	5%	332,950	7½%	552,825	10%	815,900
Gross contribution lost on lost sales	34%	113,203	35%	193,489	35%	285,565

2. Additional Expenses

Based on Technical statements the additional running expenses involved in this project will be as below:-

	1976 £
Salaries including Fringe Costs	3500
Wages including Fringe Costs	5500
Rates	850
Heat, Light, Power & Water	5000
Maintenance	3000
Total	17850

3. Capital Expenditure

	1975 Jul/Dec £	1976 Jan/Jun £	1976 Jul/Dec £	1977 Jan/Jun £	1977 Jul/Dec £	1978 Jan/Jun £	1978 Jul/Dec £
Buildings	10000	-	-	-	-	-	-
Machinery	38000	64000	4000	-	-	-	-
Total (as per section 4)	48000	64000	4000	-	-	-	-

Depreciation

	1976		1977		1978	
	Jan/Jun	Jul/Dec	Jan/Jun	Jul/Dec	Jan/Jun	Jul/Dec
	£	£	£	£	£	£
Building 40 years	125	125	125	125	125	125
Machinery 5 years	-	10000	10560	10560	10560	10560
Total	125	10125	10685	10685	10685	10685

4. Working Capital

In the 1975/77 Management Plan the relationship of working capital to turnover, on average, is £10,246,000 turnover to £1,740,000 in working capital.

Therefore, the loss of turnover as stated in item 1 above will result in the following reduction in working capital or conversely that amount of working capital must be associated with this fixed capital expenditure for the pruposes of evaluation:

	1976	1977	1978
	£	£	£
Turnover Lost	332950	552825	815900
Associated Working Capital	56542	93882	138558

5. Summary

	1976	1977	1978
	£	£	£
Turnover Lost	332950	552825	815900
Contribution Lost	113203	193489	285565
Additional Expenses	(17850)	(17850)	(17850)
Depreciation	(10250)	(21370)	(21370)
Profit at Risk	85103	154269	246345
Expenditure			
Fixed	115600	-	-
Working	56542	37340	44676
Average Funds			
Fixed	110475	94665	73295
Working	56542	75212	116220
Total	167017	169877.	189515
Rate of Return	50%	90%	129%

6. <u>Pay Back</u>

Pay back would be:-

Fixed Expenditure	in 25 months
Total Expenditure including associated working capital	in 28 months

<u>DCF</u>

The DCF rate is in excess of 50%.

7. <u>Other Effects</u>

Existing Management Plan forecast results

	1976 £000's	1977 £000's	Projected 1978 £000's
ANFE	3067	3644	4329
Profit	1544	1702	1867
Return	50.3	46.7	43.1

Results if no Test Rig

	1976 £000's	1977 £000's	Projected 1978 £000's
ANFE	2900	3474	4139
Profit	1459	1548	1621
Return	50.3	44.5	39.1

Year		Initial Expenditure & Realisation	Tax Allowance	Profit/Loss before tax & Depreciation	Tax 52%	Net Cash Flow	DCF 50%	
1.	P B WC	38000 10000 -	(19760) (2288) -	- - -	- - -	25952 -	1.000	25952
2.	P WC	67600 56542	(208) -	(95553) -	- -	28581 -	.667 -	19064 -
3.	WC	37340	(208)	(175639)	49584	(88923)	.444	(39482)
4.	WC	44676	(208)	(267715)	91332	(131915)	.296	(39047)
5.			(208)	(267715)	139212	(128711)	.198	(25485)
6.			(208)	(267715)	139212	(128711)		(58998)
7.			(208)	(267715)	139212	(128711)		
8.			(208)	(267715)	139212	(128711)		
9.	P B WC	(1000) (10000) (124702)	(208)	(267715)	139212	(264413)		
10.		-	-	-	209777	209777		

TABLE II

STIFFNESSES AT VARIOUS INTERNAL OVERPRESSURES

Internal overpressure, bar	0	5	10	16
Bending stiffness, 10^6 N m^2				
R = 3.65 m	0.12 ± 19 %			
5.40 "	0.15 ± 33 %			
7.70 "		0.30 ± 31 %		
17.75 "	0.23 ± 46 %			
18.15 "		0.47 ± 27 %		
20.00 "			0.66 ± 25 %	
22.90 "				0.89 ± 22 %
Torsion stiffness, 10^6 N m^2	12.25 ± 3.7 %	12.4 ± 2.4 %	12.4 ± 2.4 %	n.d.
Tensile stiffness, 10^6 N	25.4 ± 25 %	31.8 ± 20%	34.2 ± 15%	n.d.

n.d. = not determined

TABLE 12

TENSILE STRENGTHS AT DIFFERENT SECTIONS
THROUGH THE THICKNESS OF THE HOSE
IN ORDER OF FAILURE IN TENSILE TEST

Nylon reinforcement layer no.	Ultimate tensile strength, N/mm^2	Remarks
3, outer surface 3, inner surface	1.3 1.6	
5, inner surface 5, outer surface	1.7 0.6	
1	2.25	Torn off right through this layer up to 1st steel cord layer
2, outer surface	2.3	Layer failed during dynamic test.

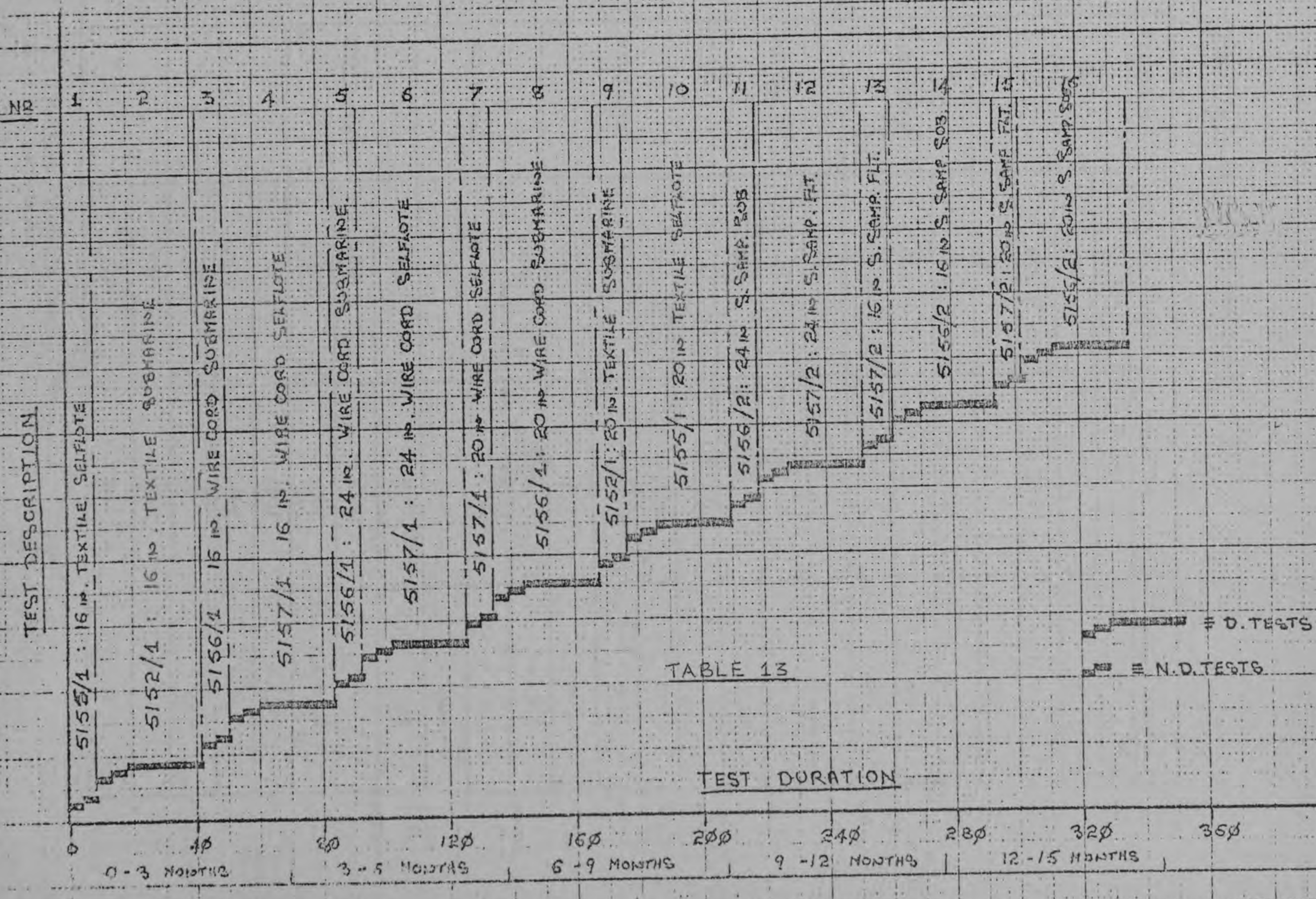
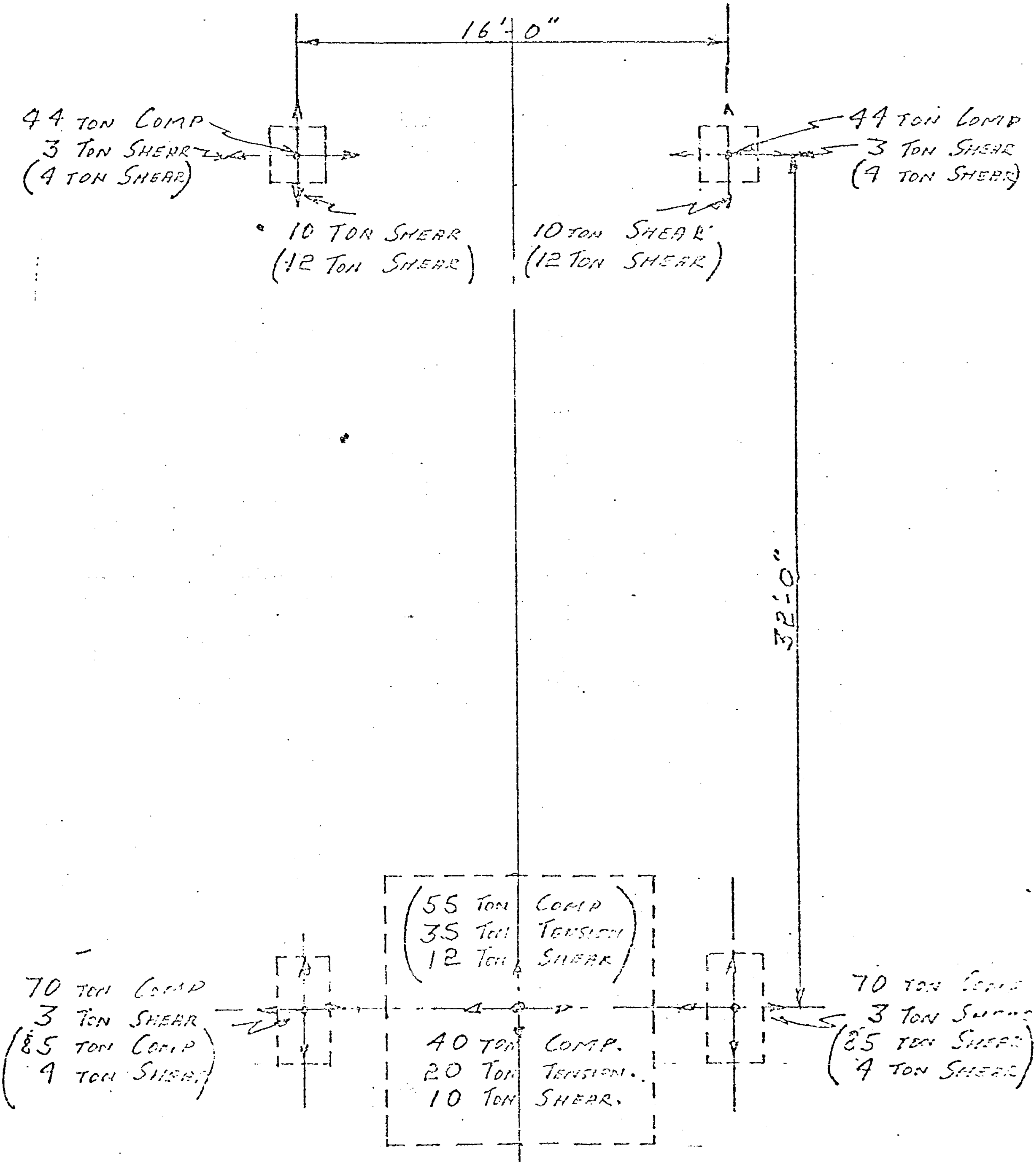

TEST Nº
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
TEST DESCRIPTION
5155/1 : 16 in. TEXTILE SELFLOTE
5152/1 : 16 in. TEXTILE SUBMARINE
5156/1 15 in. WIRE CORD SUBMARINE
5157/1 16 in. WIRE CORD SELFLOTE
5156/1 : 24 in. WIRE CORD SUBMARINE.
5157/1 : 24 in. WIRE CORD SELFLOTE
5157/1 : 20 in WIRE CORD SELFLOTE
5156/1 : 20 in WIRE CORD SUBMARINE
5152/1 : 20 in. TEXTILE SUBMARINE
5155/1 : 20 in TEXTILE SELFLOTE
5156/2 : 24 in S. SAMP. SUB
5157/2 : 24 in S. SAMP. FLT.
5157/2 : 16 in S. SAMP. FLT.
5155/2 : 16 in S. SAMP. SUB
5157/2 : 20 in S. SAMP. FLT.
5155/2 : 20 in S. SAMP. SUB
≡ D. TESTS
≡ N.D. TESTS
TABLE 13
TEST DURATION
0
40
80
120
160
200
240
280
320
360
0 - 3 MONTHS
3 - 5 MONTHS
6 - 9 MONTHS
9 - 12 MONTHS
12 - 15 MONTHS

TABLE I4 - STATIC TEST PROGRAMME

PRESSURE	BENDING MOMENT	TORSION	AXIAL LOAD
0 lb/in^2	0	$\pm$ T	0
	.33M	0	0
	.66M	$\pm$ .50T	0
	M	0	0
100 lb/in^2	0	$\pm$ .50T	0
	.66M	0	0
	M	$\pm$ T	0
200 lb/in^2	0	0	+ A
	0	$\pm$ T	0
	.66M	0	0
300 lb/in^2	0	+ .50T	0
	.33M	+ .50T	0
	M	0	0
	0	- .50T	0
300 lb/in^2	.66M	$\pm$ T	+ .50A
	M	0	$\pm$ A
	0	- .50T	+ .50A
200 lb/in^2	.33M	$\pm$ T	- .50A
	.66M	+ .50T	+ A
	M	- .50T	+ .50A
100 lb/in^2	0	+ .50T	+ .50A
	.33M	- .50T	+ A
	.66M	0	+ .50A
0	0	$\pm$ T	+ A
	.66M	$\pm$ T	0

T) Maximum allowable loads
M) for each test
A) hose.



LOADS IN BRACKETS INCLUDE 50% INCREASE IN POWER.

FIGURE 72

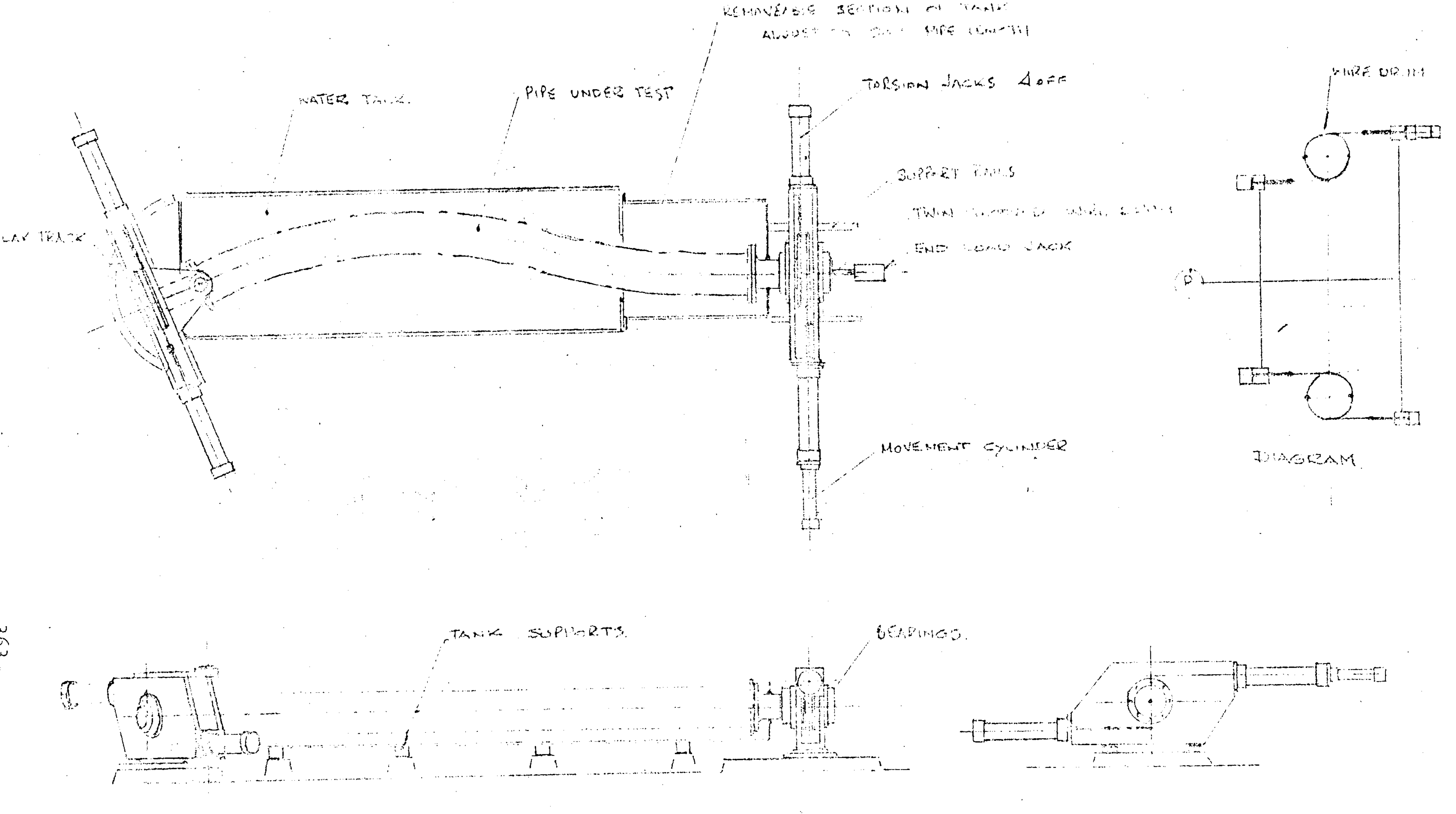

WATER TANK.
PIPE UNDER TEST
TORSION JACKS 4 OFF
SUPPORT RAILS
END LOAD JACK
MOVEMENT CYLINDER
WIRE DRUM
DIAGRAM
TANK SUPPORTS
BEARINGS.

FIGURE 73

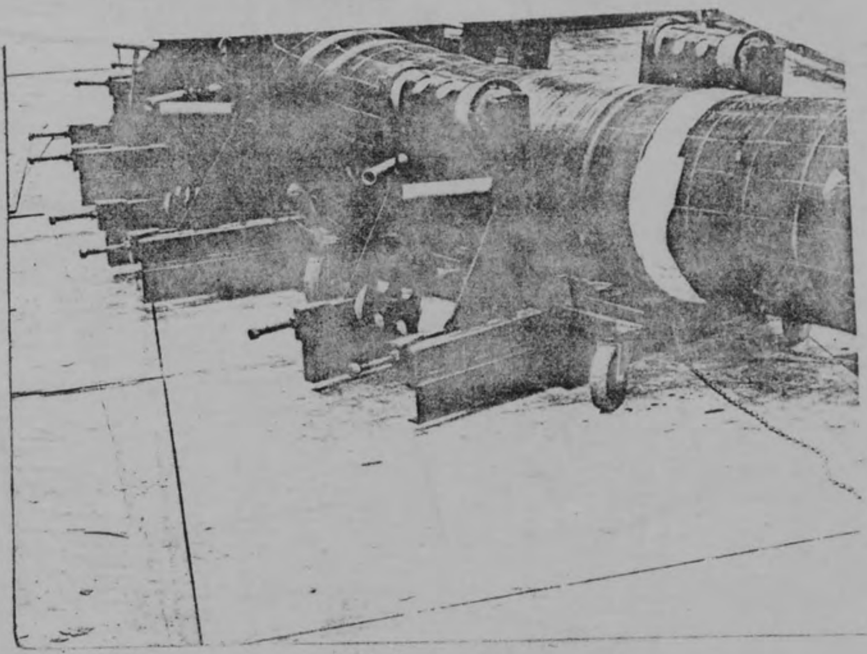
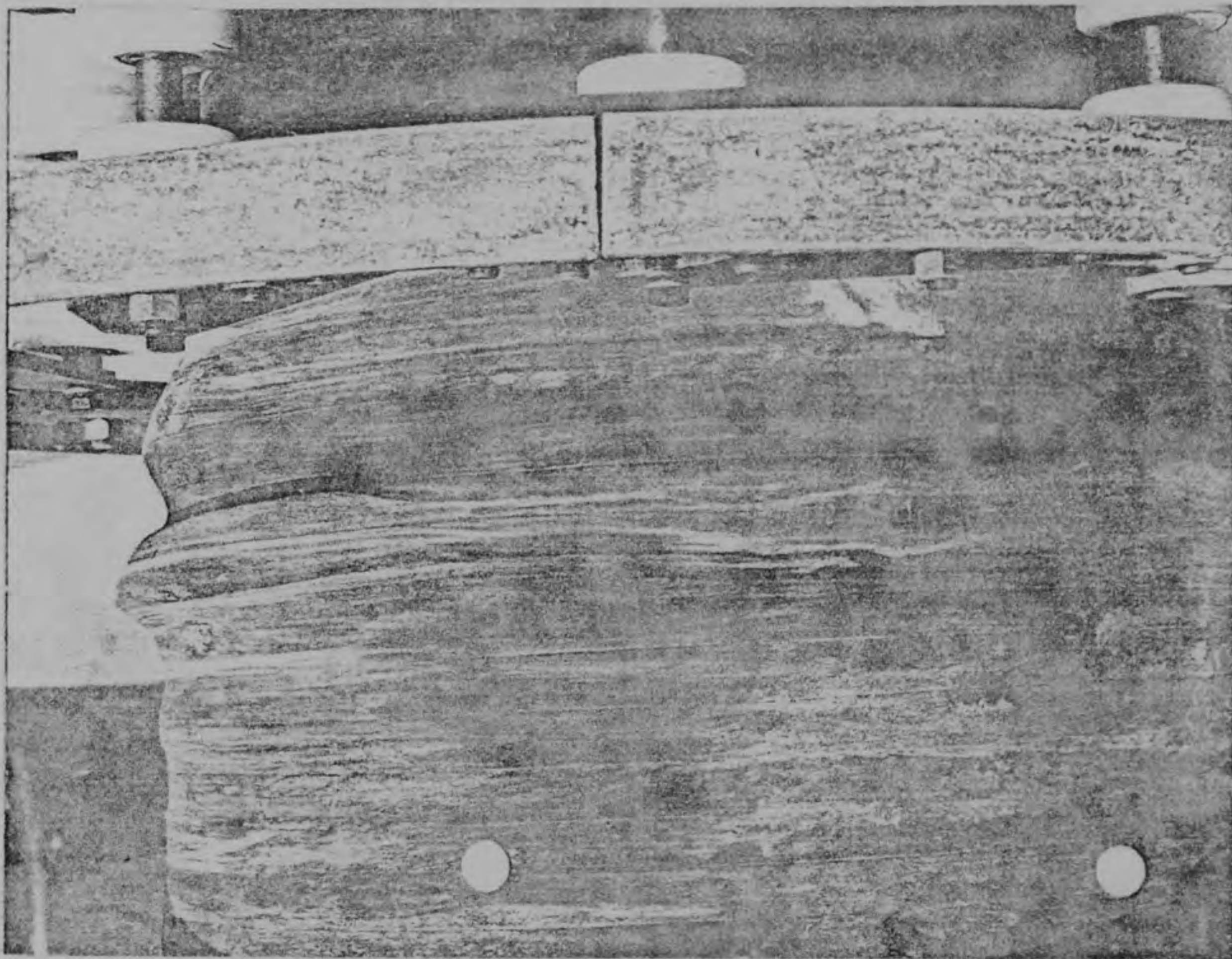
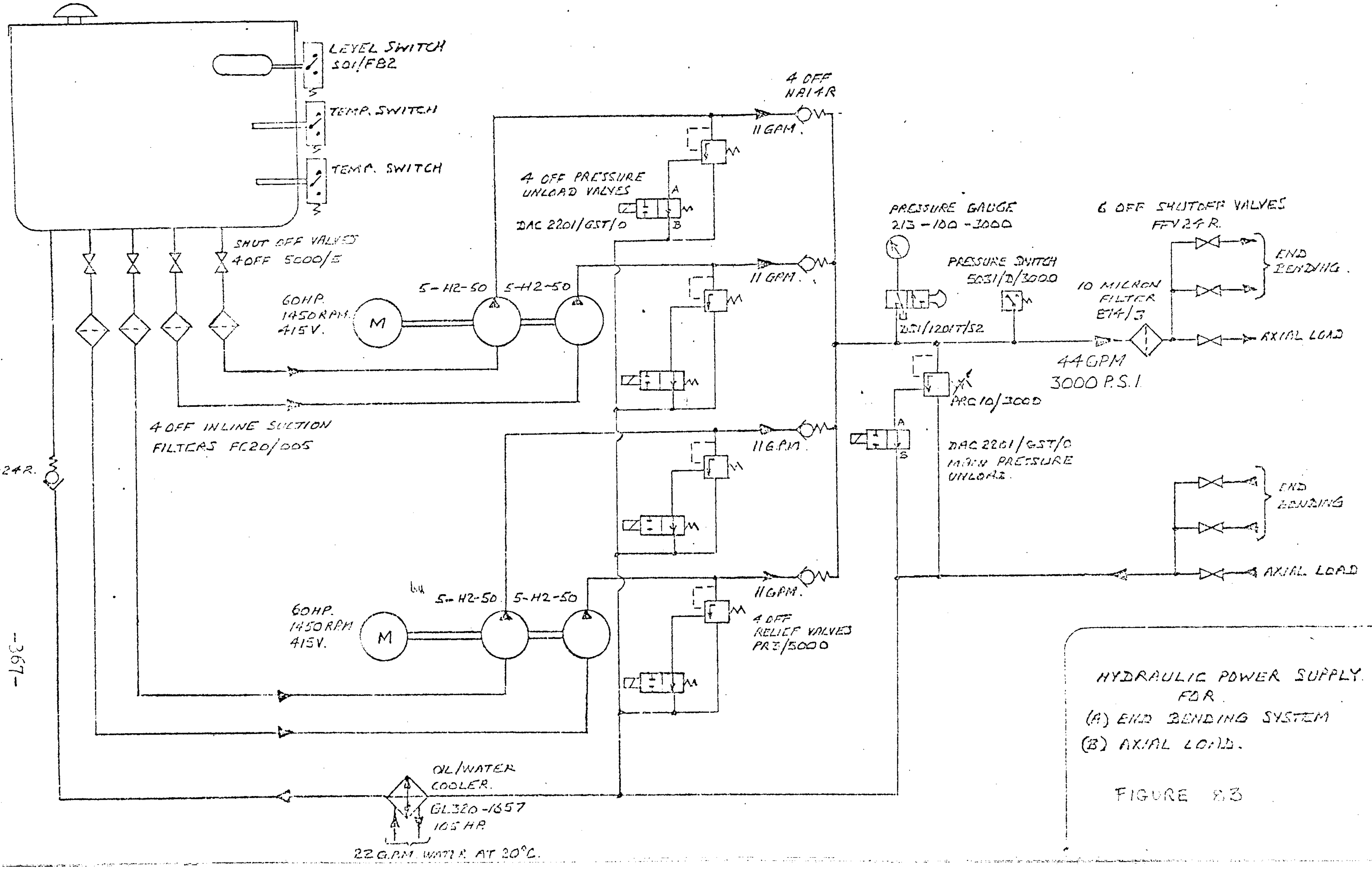


FIGURE (76) - HOSE SUPPORT TROLLEY

FIGURES (78) & (79) K.S.L.A. TEST RIG

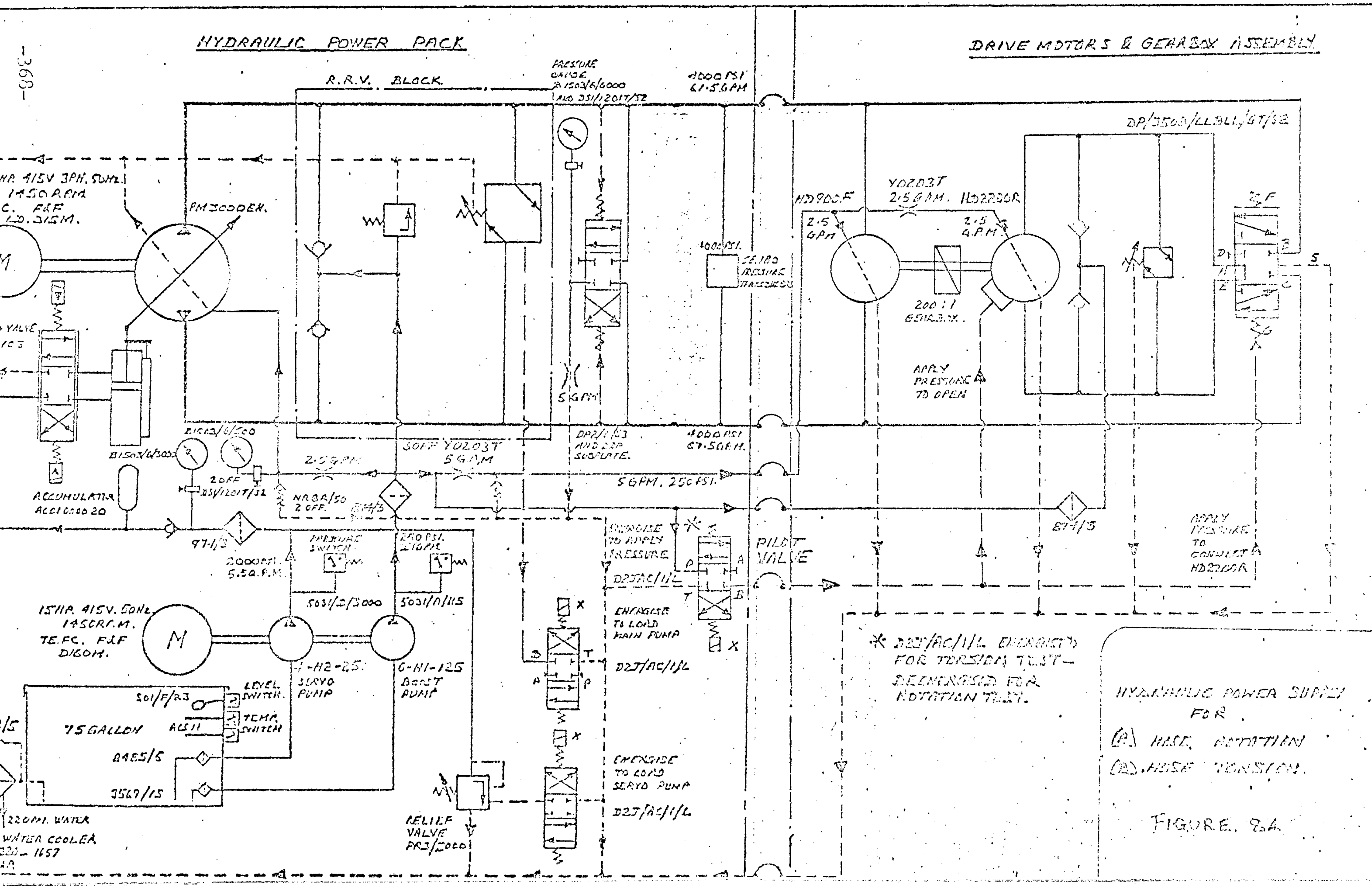


FIGURES (80) & (81) K.L.S.A. TEST RIG



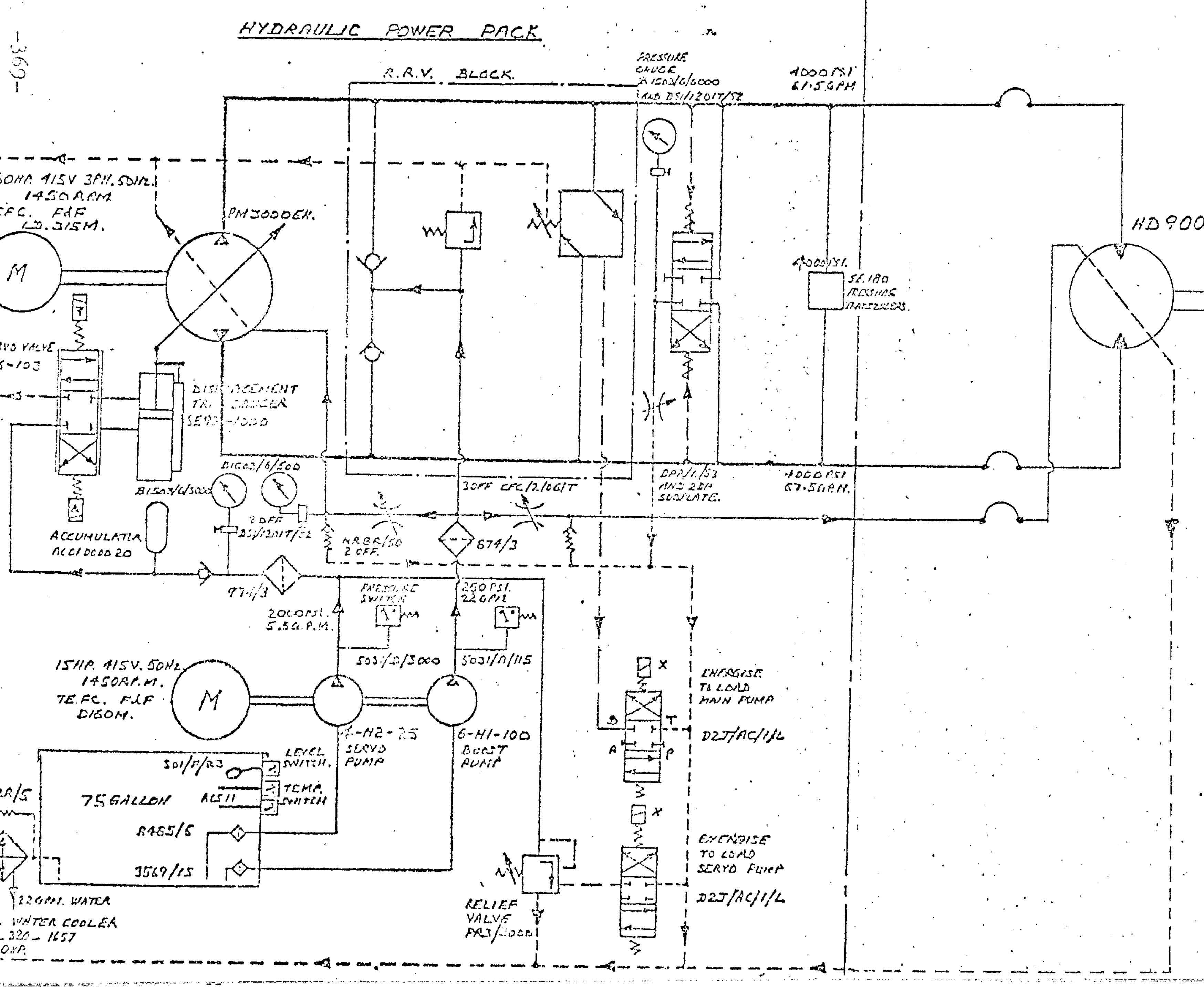
HYDRAULIC POWER SUPPLY FOR
(A) END BENDING SYSTEM
(B) AXIAL LOAD.

FIGURE 8.3



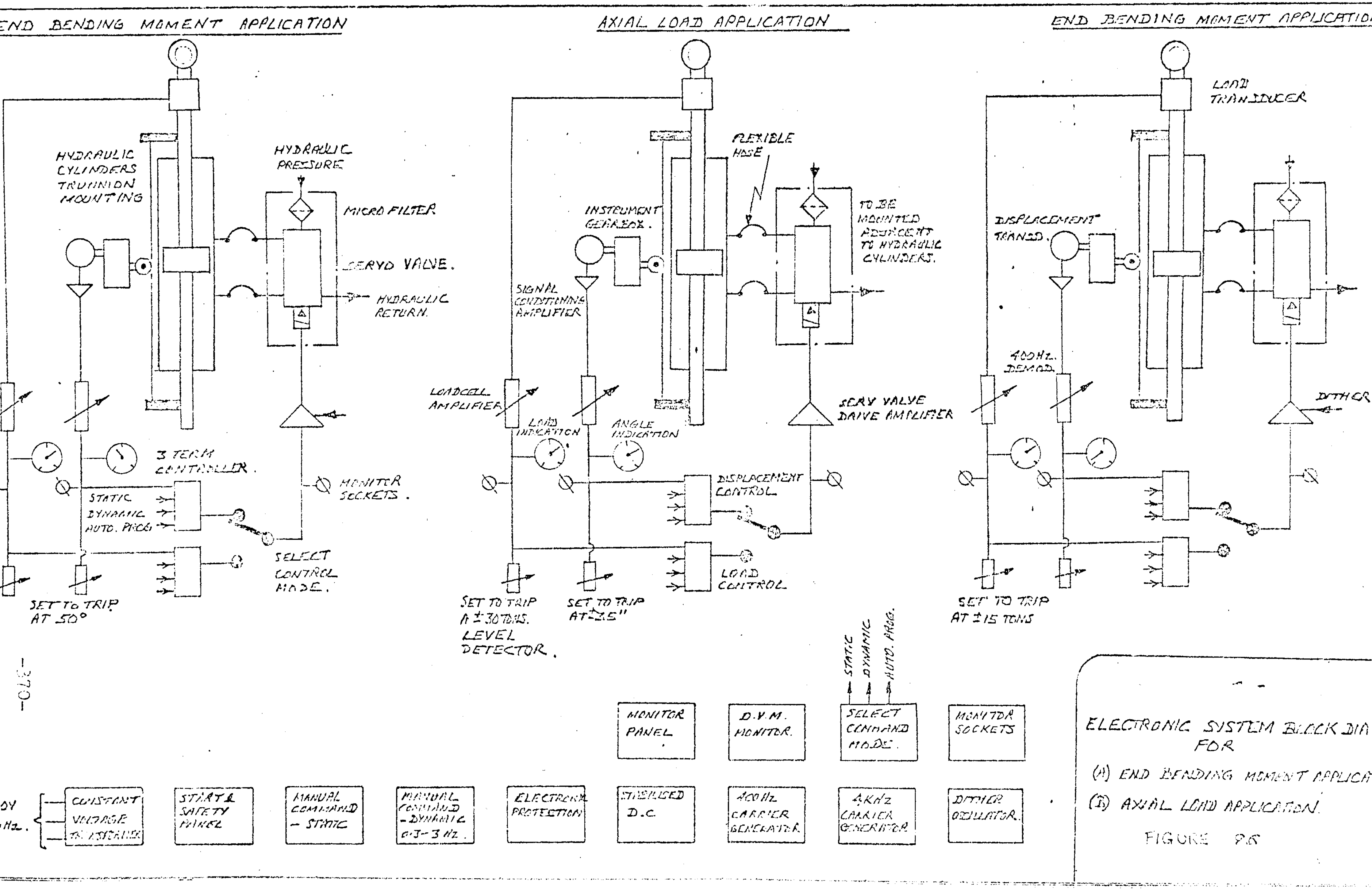
HYDRAULIC POWER SUPPLY FOR
(A) HOSE ROTATION
(B) HOSE TORSION.

FIGURE. 8A

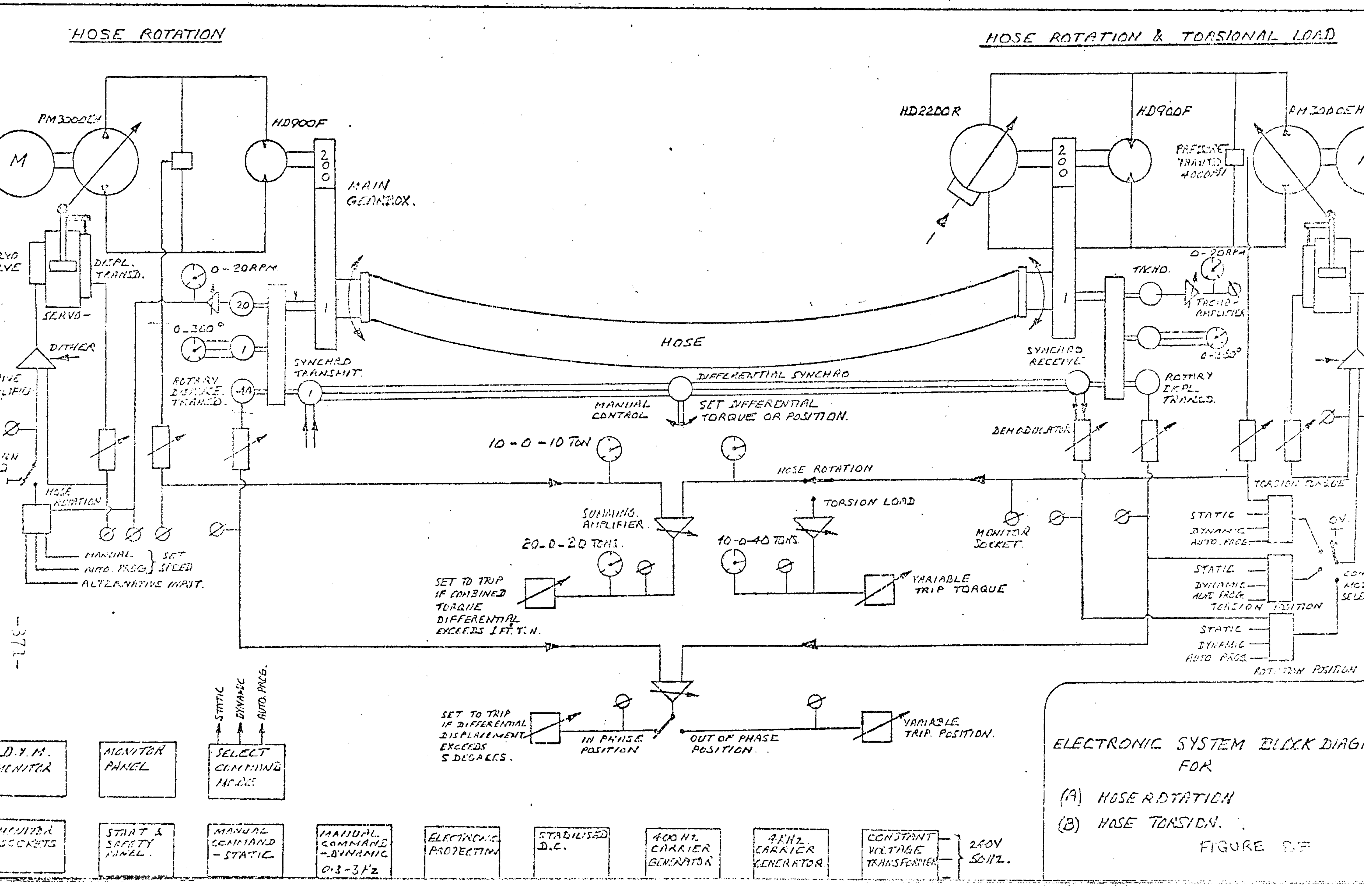


HYDRAULIC POWER SUPPLY FOR HOSE ROTATION:

FIGURE 86



ELECTRONIC SYSTEM BLOCK DIA
FOR
(A) END BENDING MOMENT APPLICA
(B) AXIAL LOAD APPLICATION.
FIGURE 9.6



ELECTRONIC SYSTEM BLOCK DIAGRAM FOR
(A) HOSE ROTATION
(B) HOSE TORSION.
FIGURE

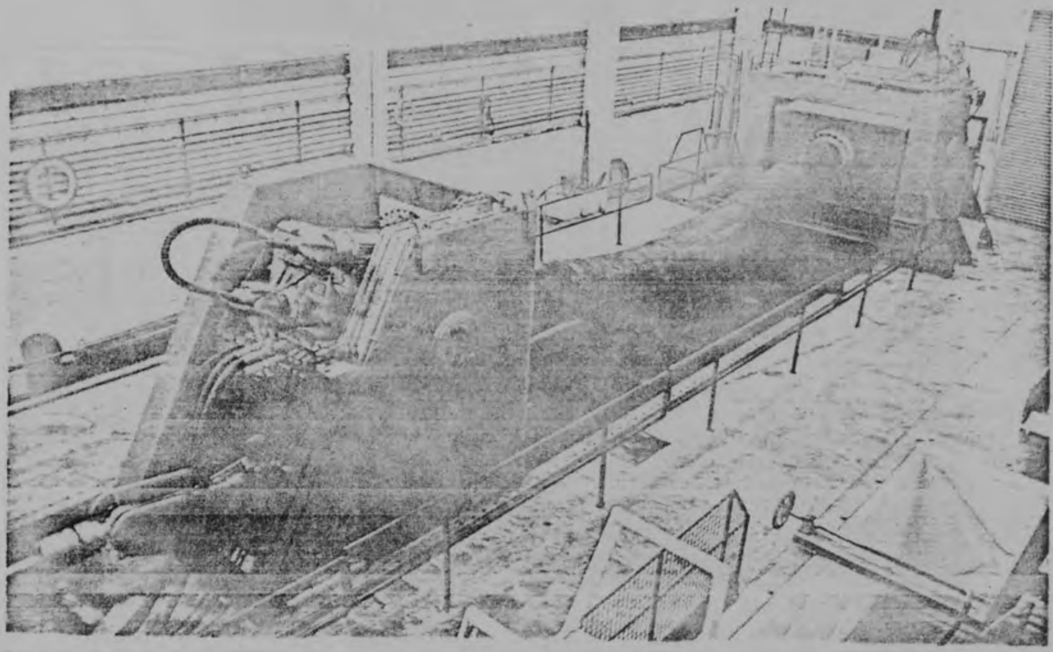


FIGURE (88) TEST RIG GENERAL VIEW

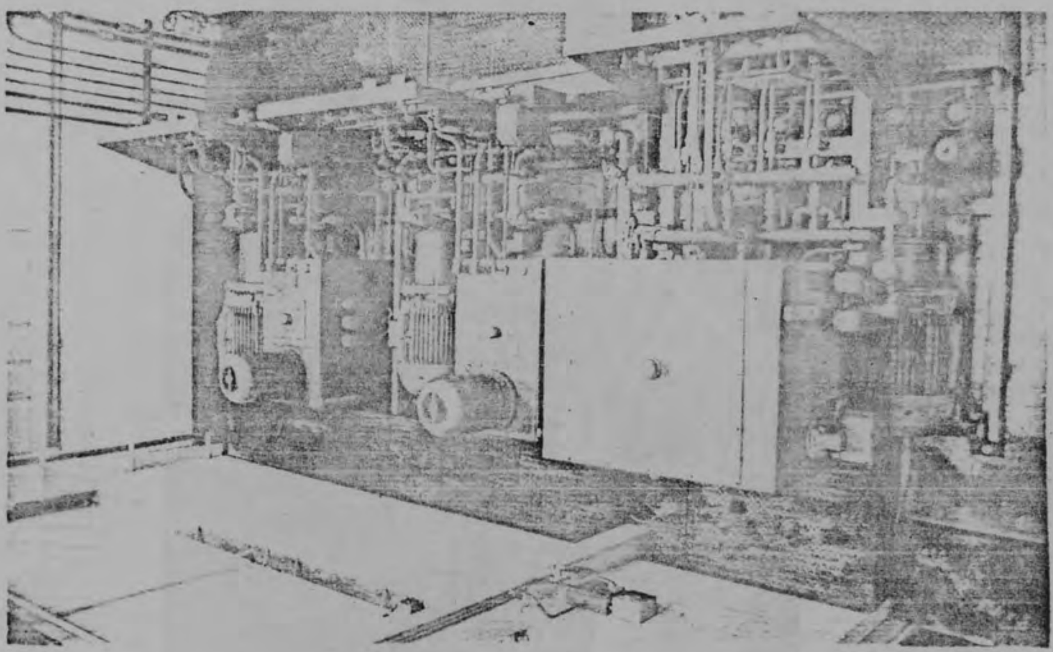


FIGURE (89) TEST RIG POWER PACKS

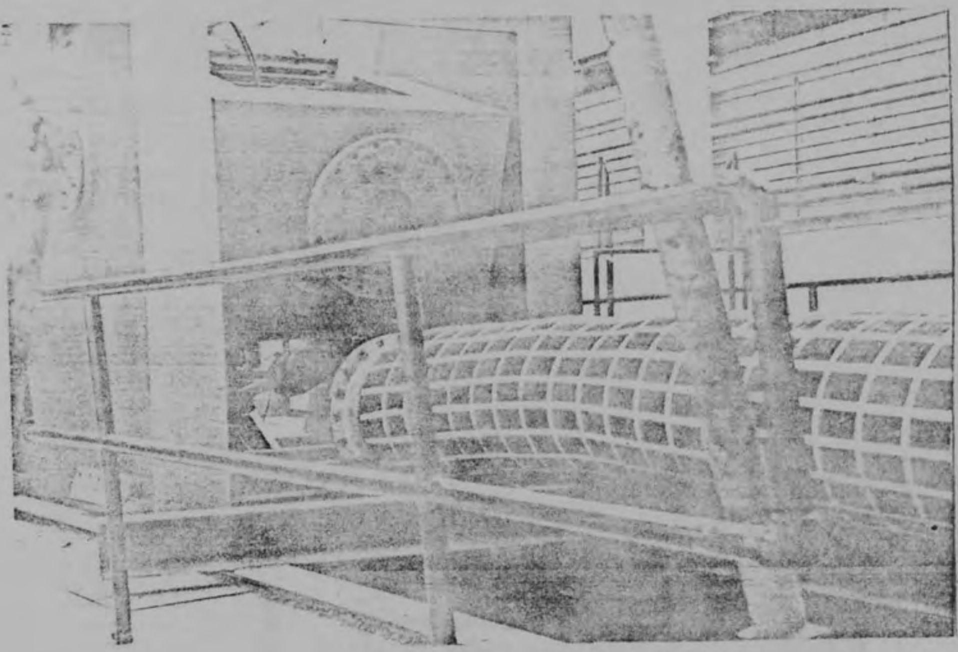


FIGURE (90) TEST RIG DRIVE UNIT

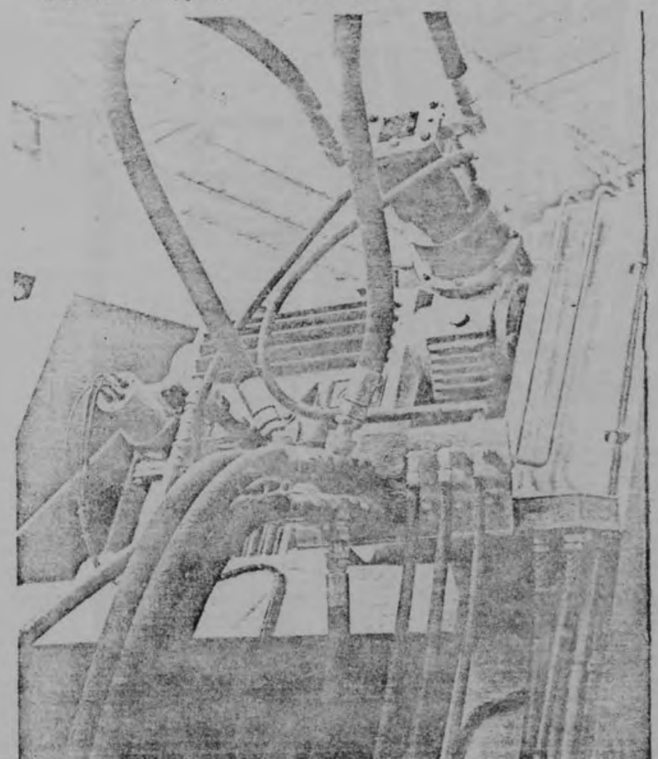


FIGURE (91) TEST RIG GEARBOX DRIVE

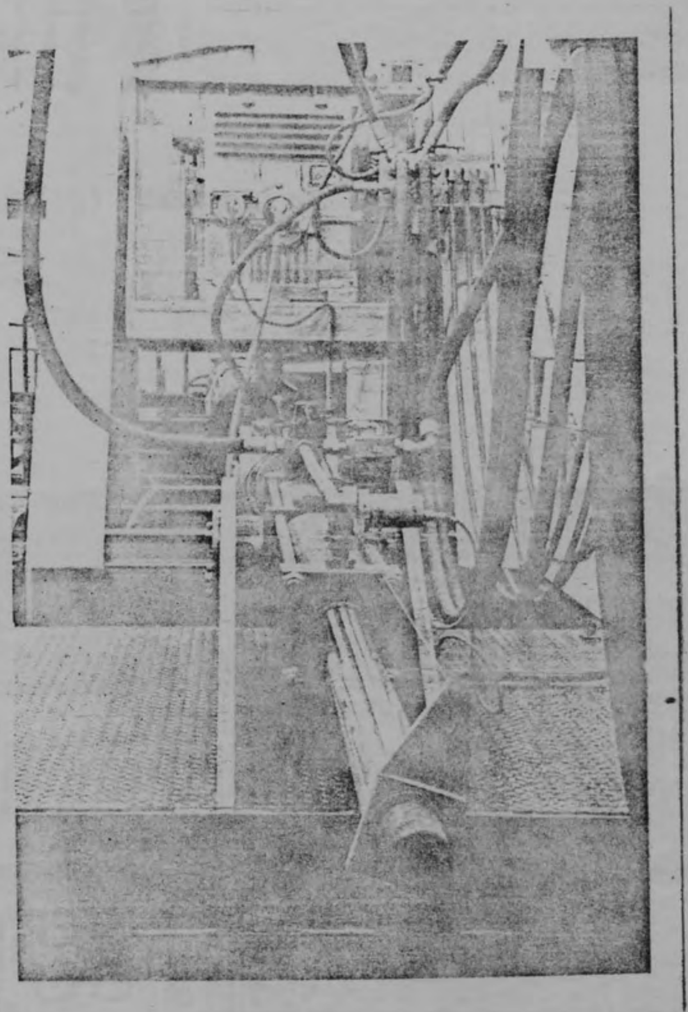


FIGURE (92) TEST RIG
AXIAL LOADING SYSTEM / HYDRAULIC CATENARYSUPPLY HOSE

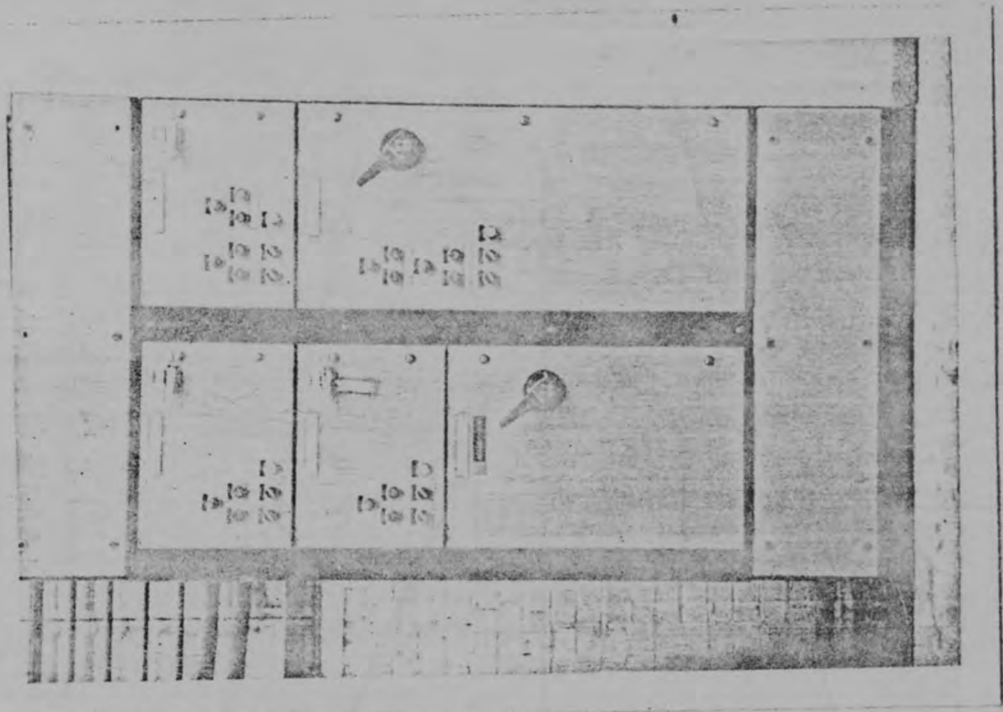


FIGURE (93) TEST RIG STARTER CABINET

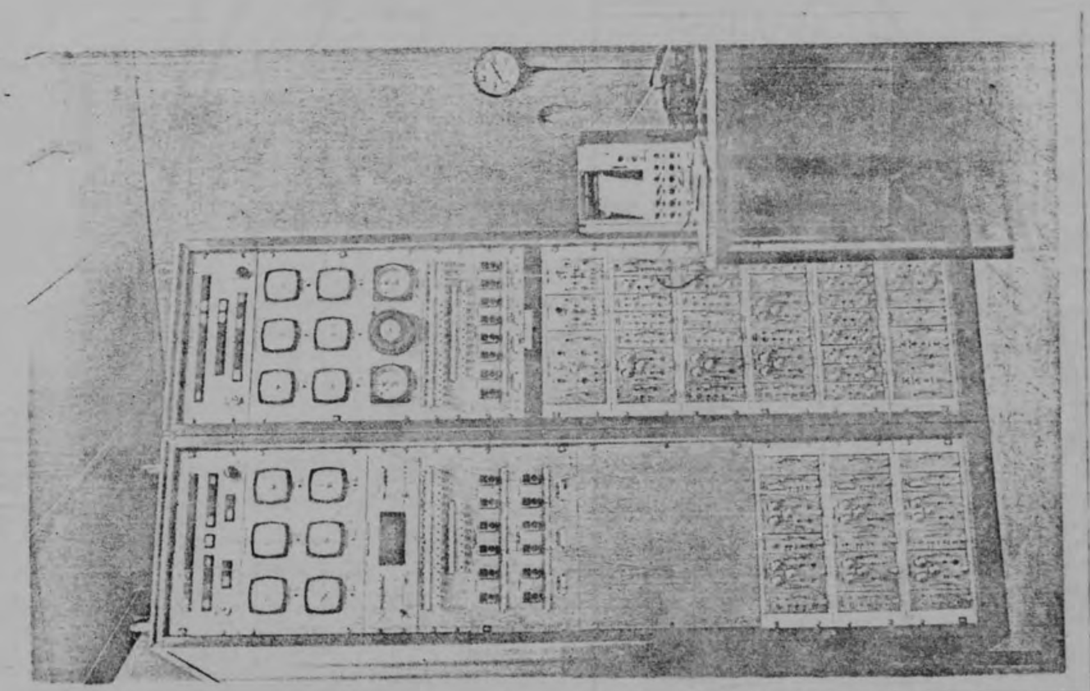


FIGURE (94) TEST RIG CONTROL CONSULE

GRAPH 3

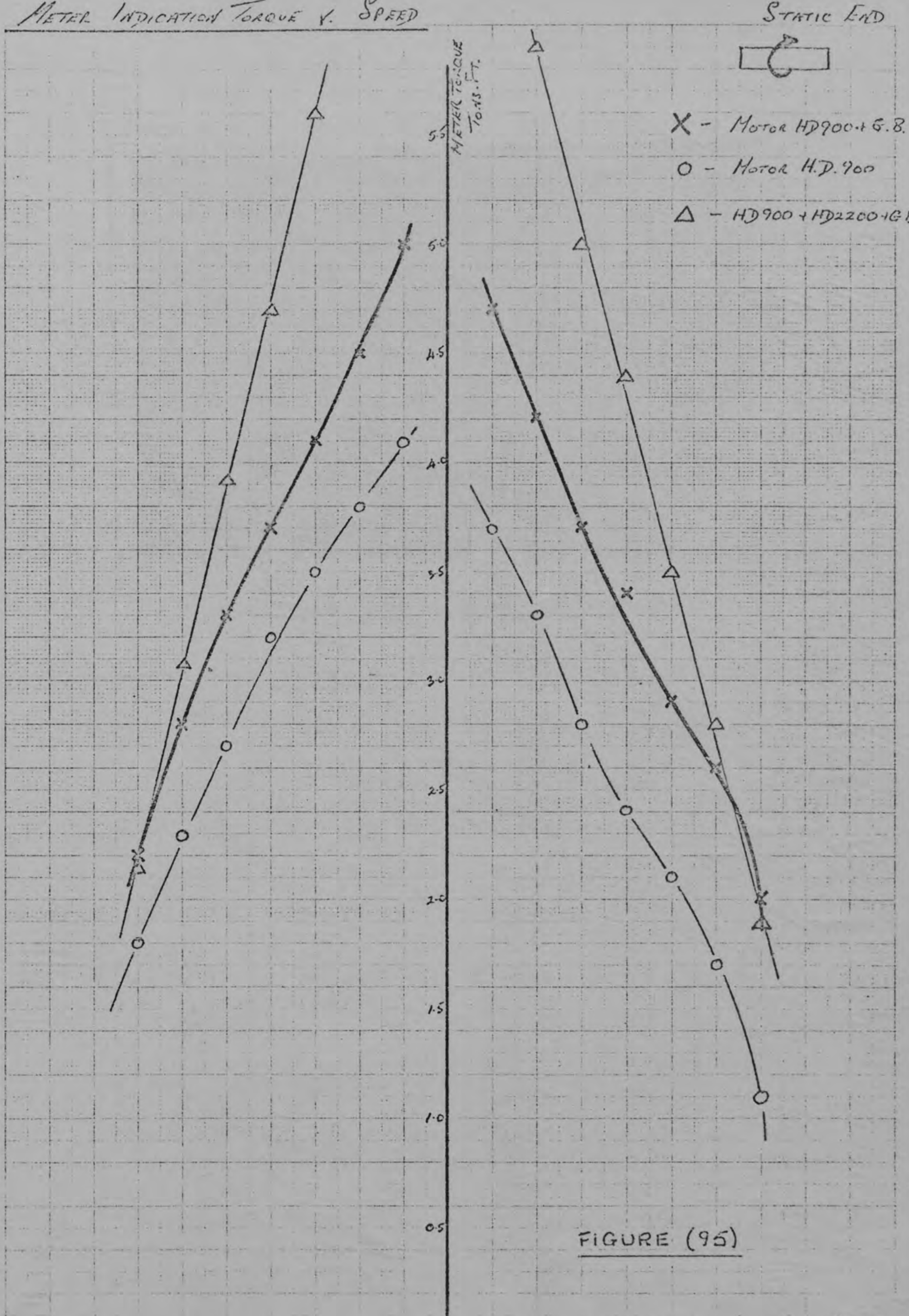


FIGURE (95)

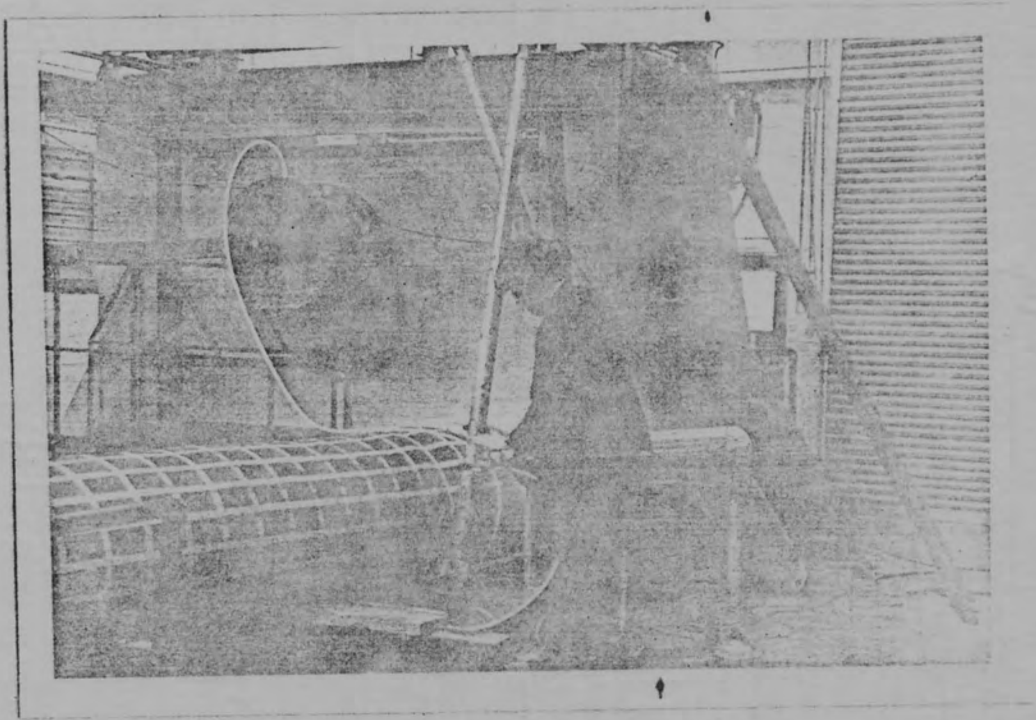
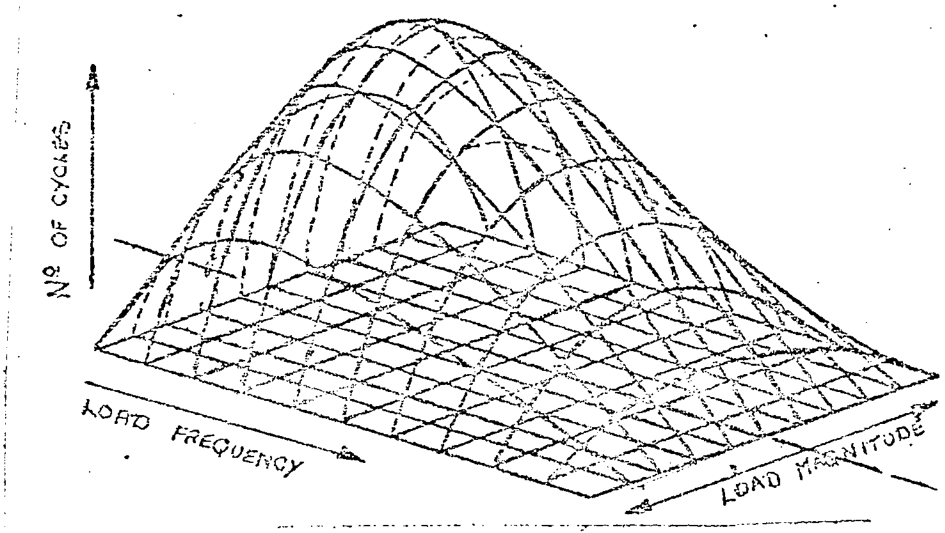


FIGURE (96) - TEST RIG TORQUE CALIBRATION

GEARBOX(S) INDICATED TORQUE
COMPENSATION
INDICATED TORQUE (TON FT)
TORSION / ROTATION DRIVE
ROTATION DRIVE
REV / MIN
× MEASURED
O AVERAGED
COMPENSATION CURVES

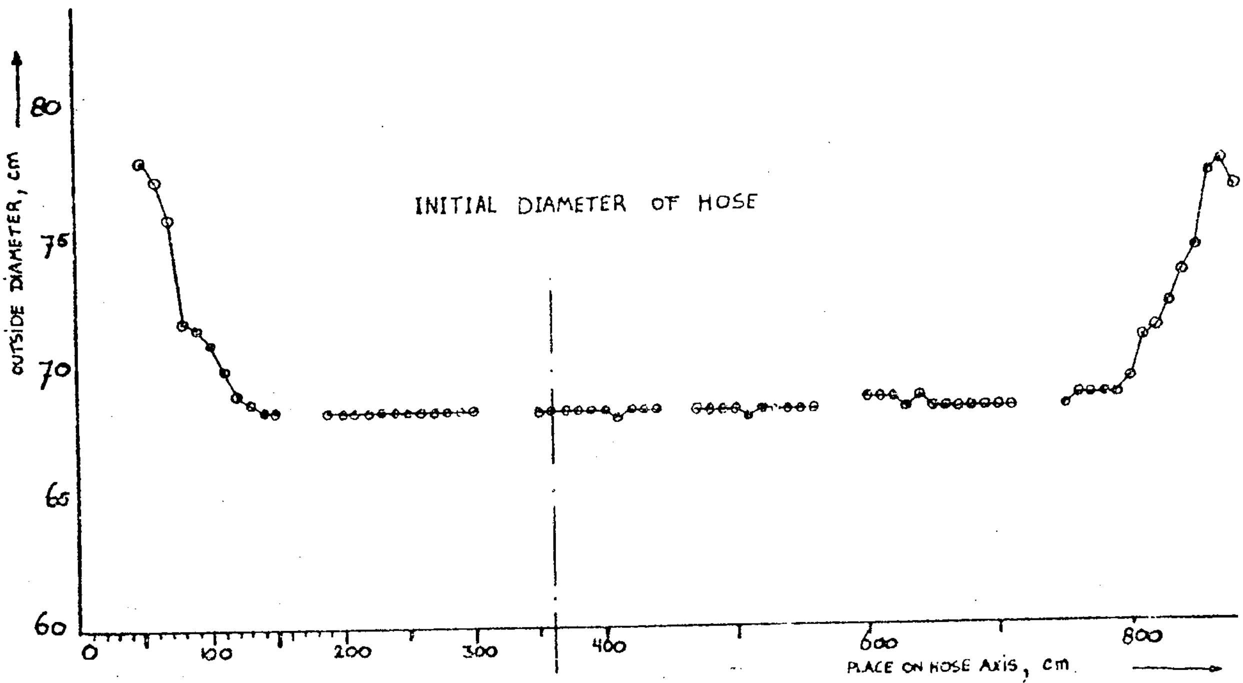
FIGURE (97)

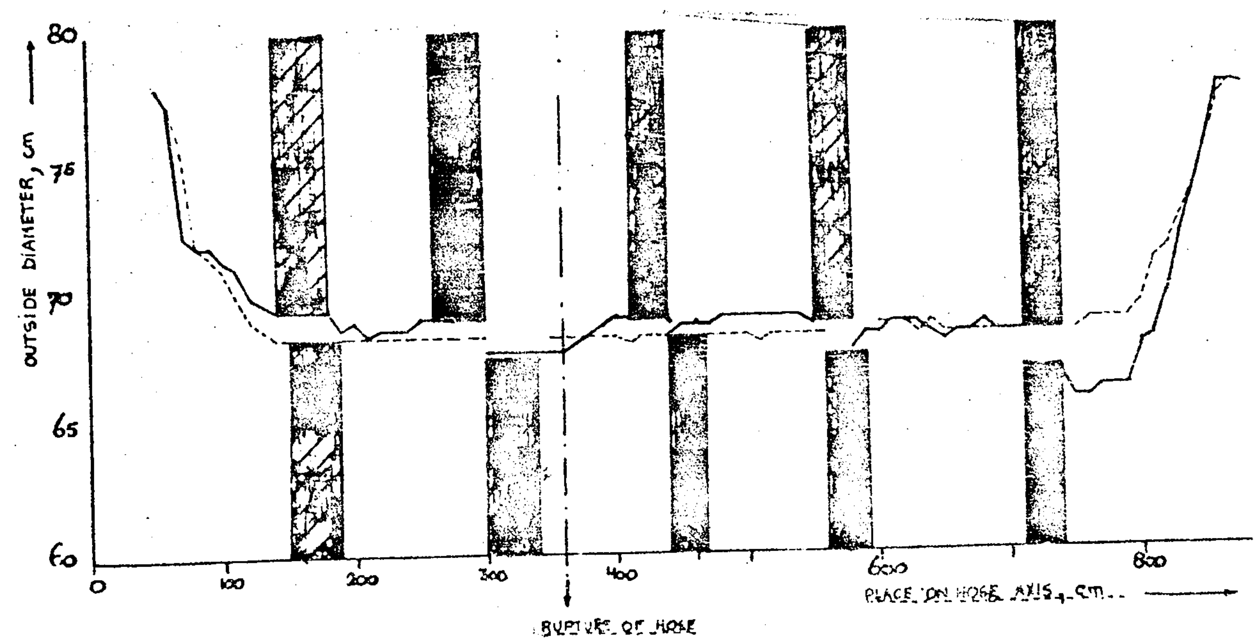


TEST RIG LOAD
INPUT FUNCTION

FIGURE (98)

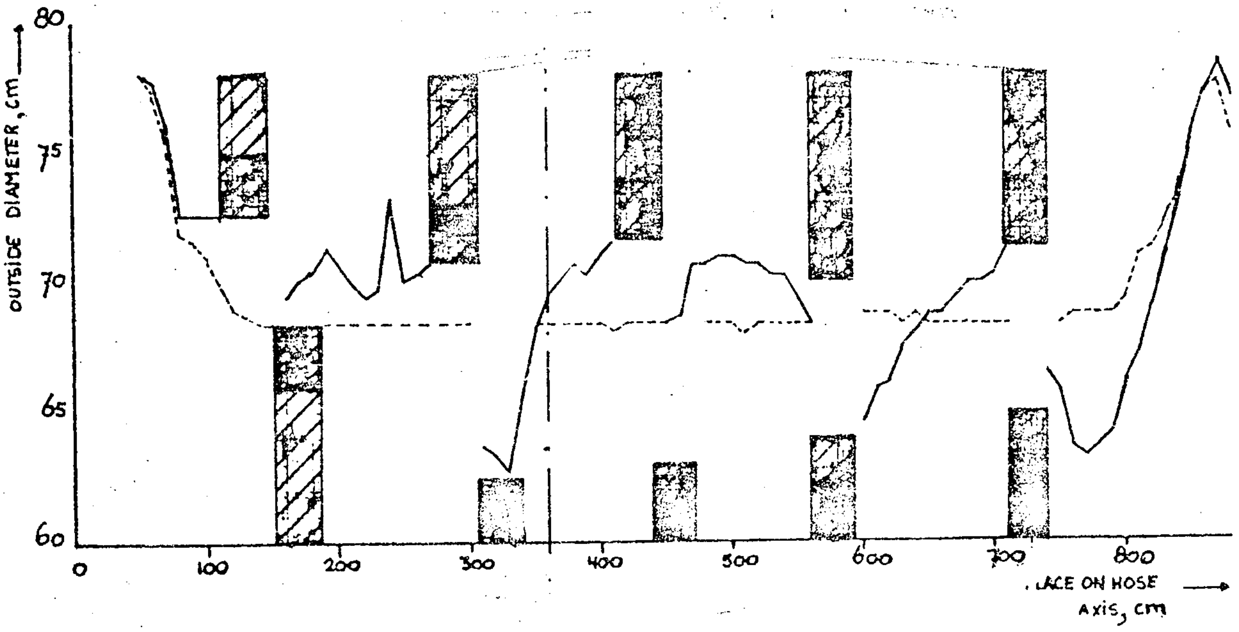
FIGURES 99 AND 100





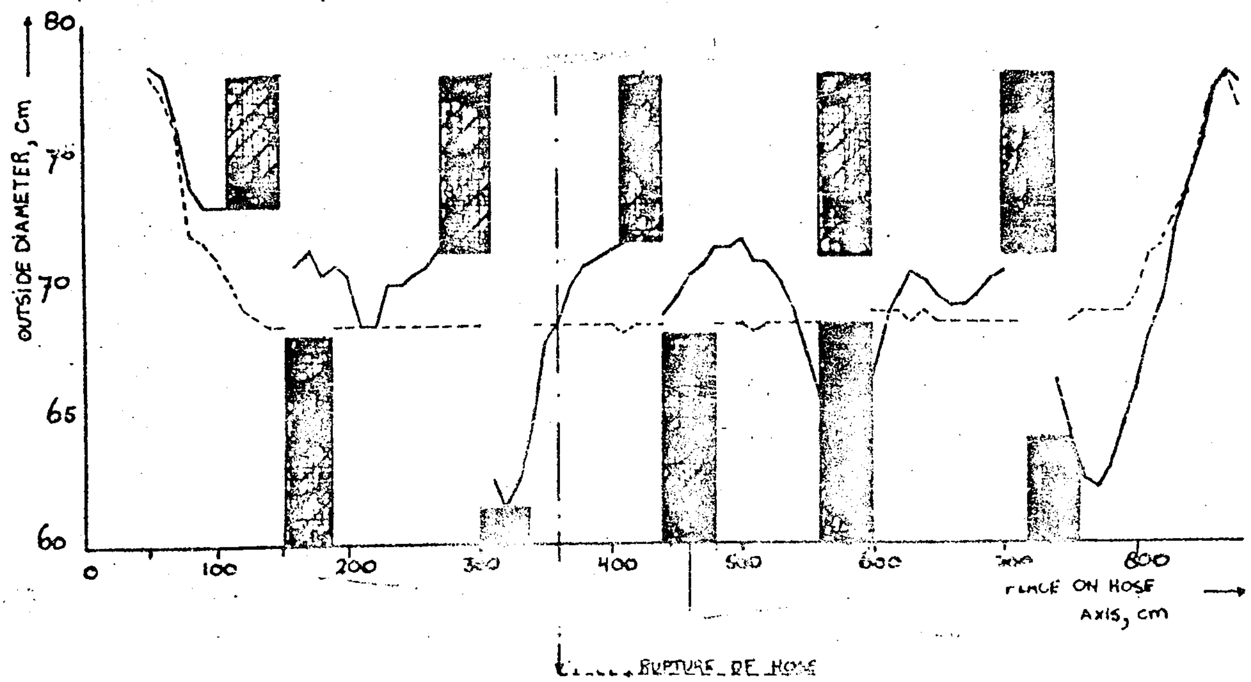
——— DIAMETER AFTER 5000 cycles AT α=10°

------- INITIAL DIAMETER



DIAMETER AFTER 3209 CYCLES AT $\alpha=20°$

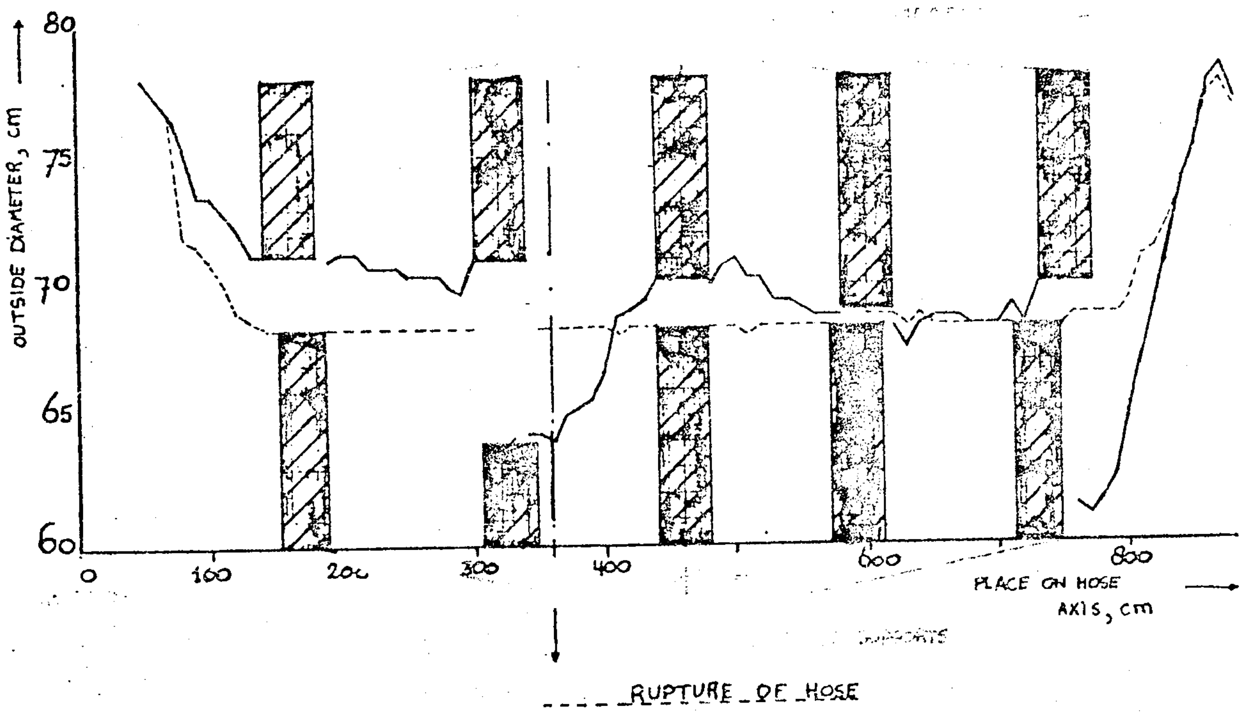
INITIAL DIAMETER



DIAMETER AFTER 5000 CYCLES AT $\alpha=20°$

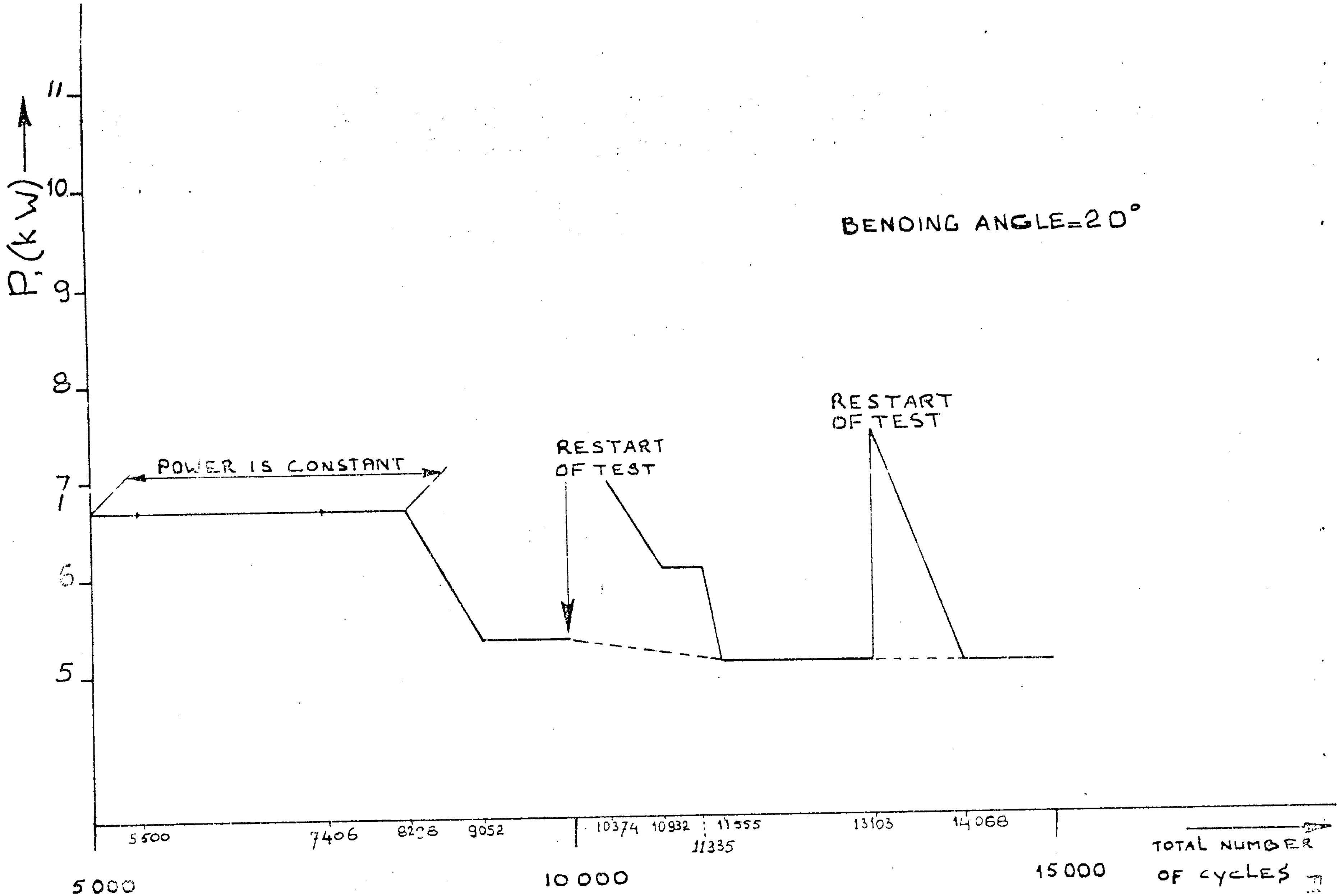
INITIAL DIAMETER

FIGURE 103



DIAMETER AFTER 10000 CYCLES AT $\alpha = 20°$

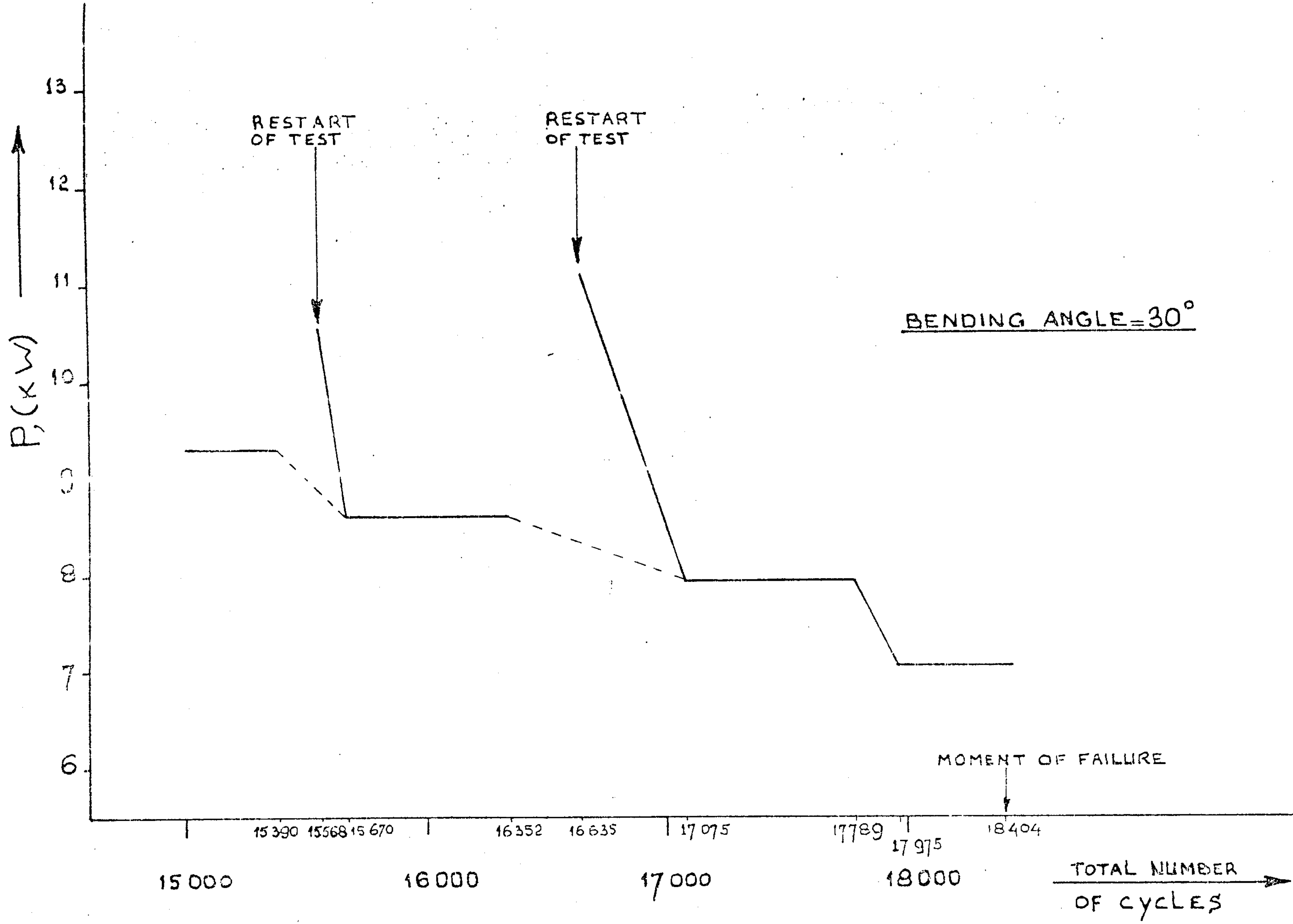
INITIAL DIAMETER



POWER CONSUMPTION OF HOSE DRIVE

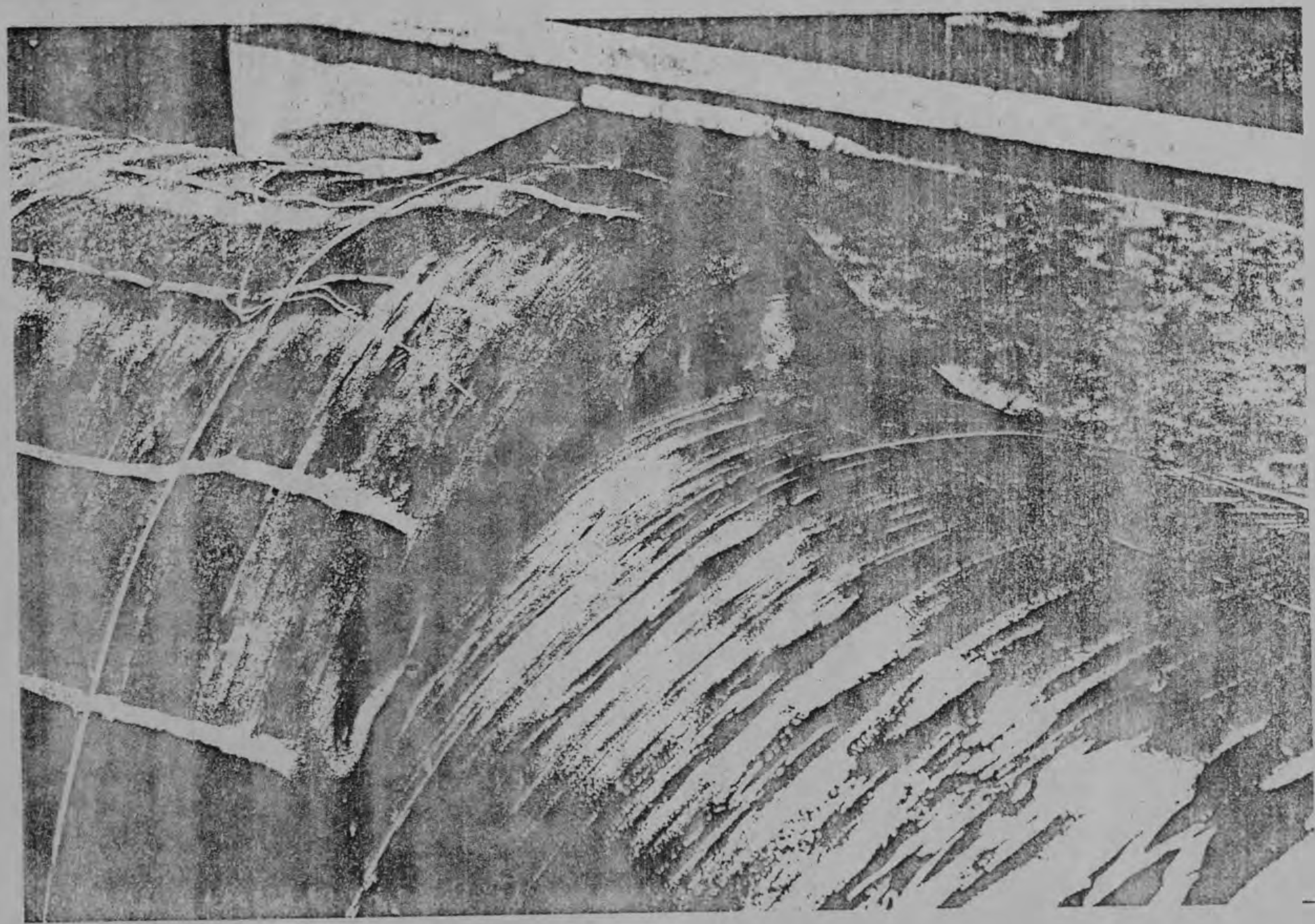
FIGURE 104

FIGURE 105

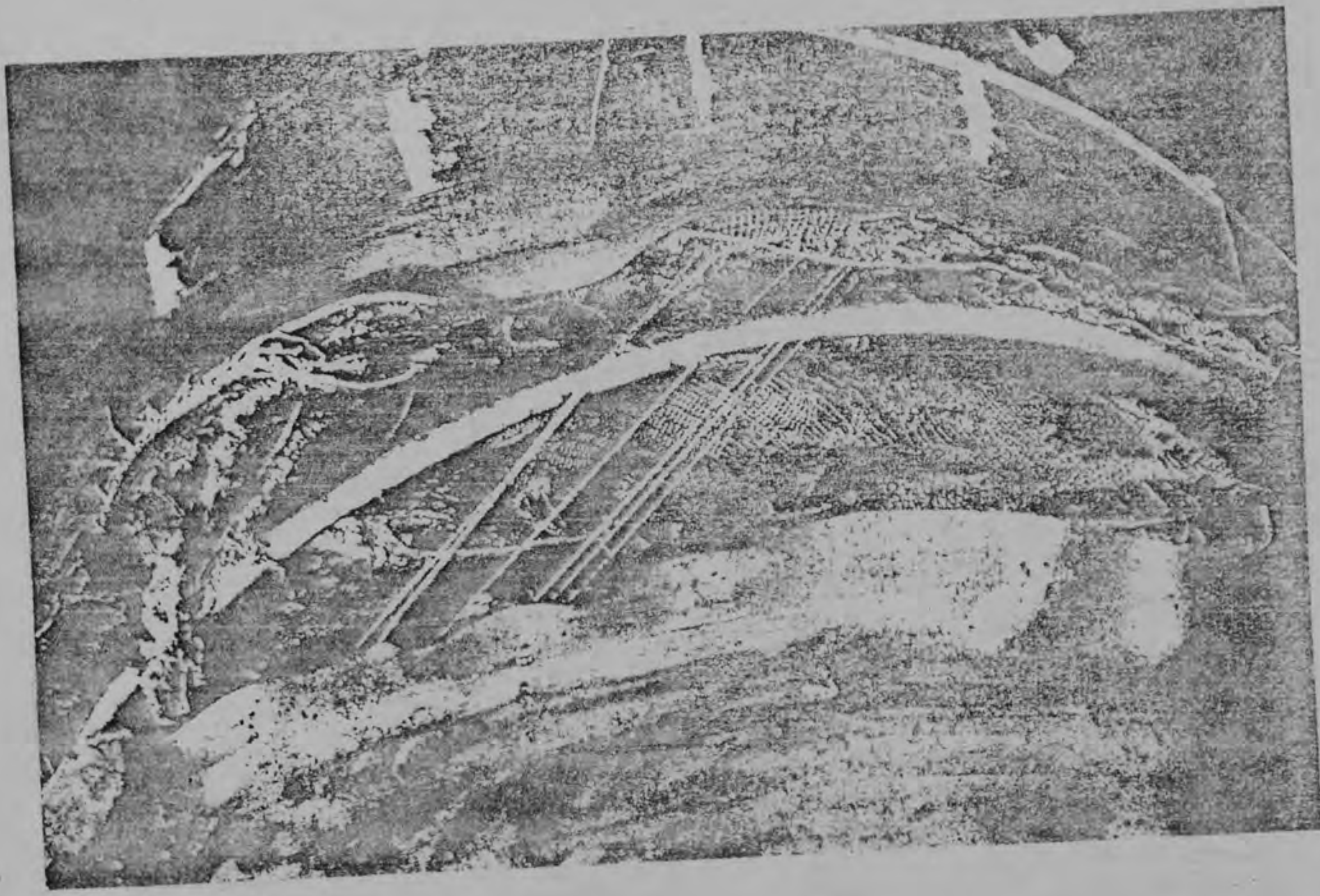


POWER CONSUMPTION OF HOSE DRIVE

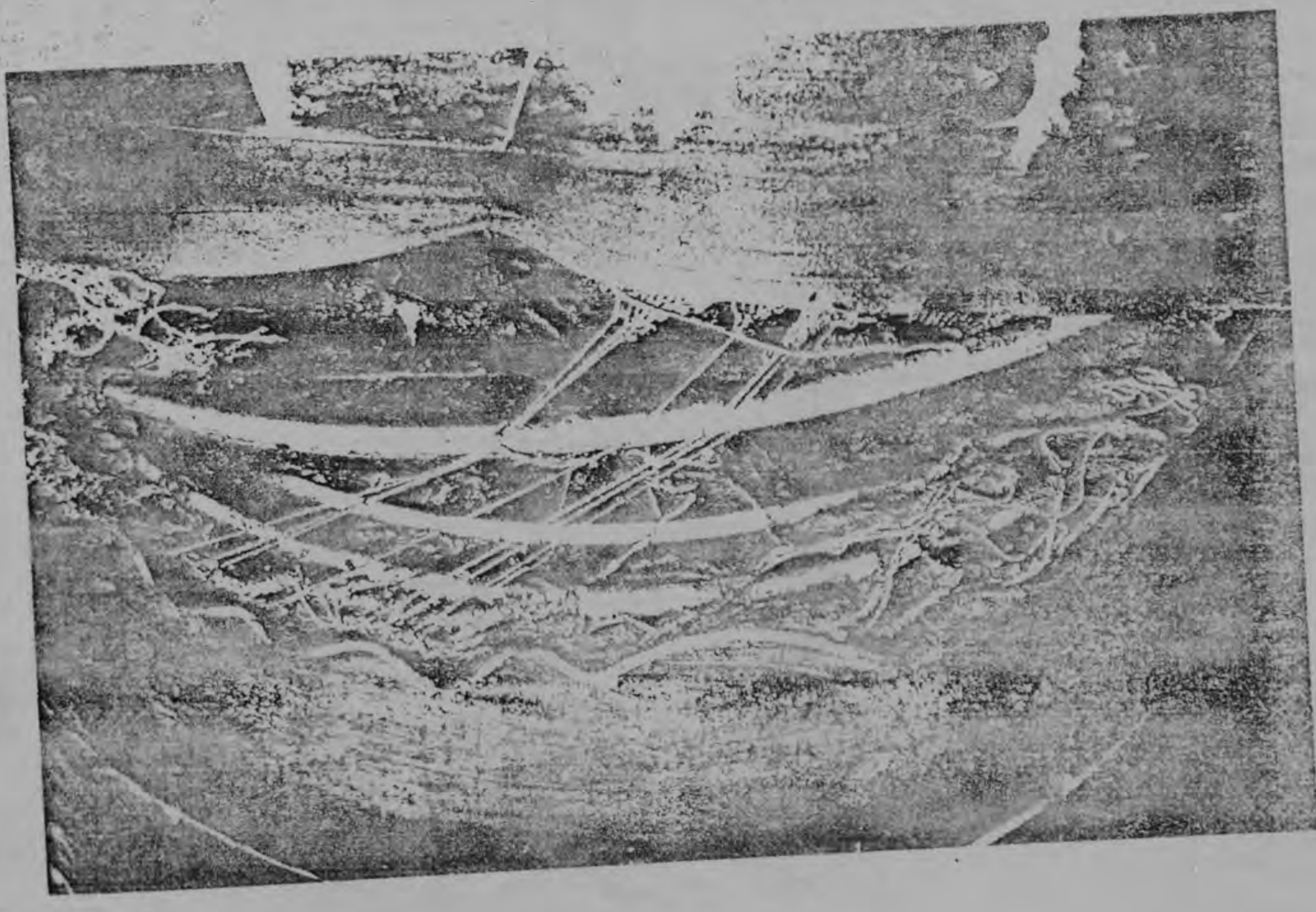
FIGURE 96



FAILED 1973–TYPE 24–INCH DOUBLE–HELIX UNDERWATER DUNLOP HOSE
(CONCAVE SIDE)

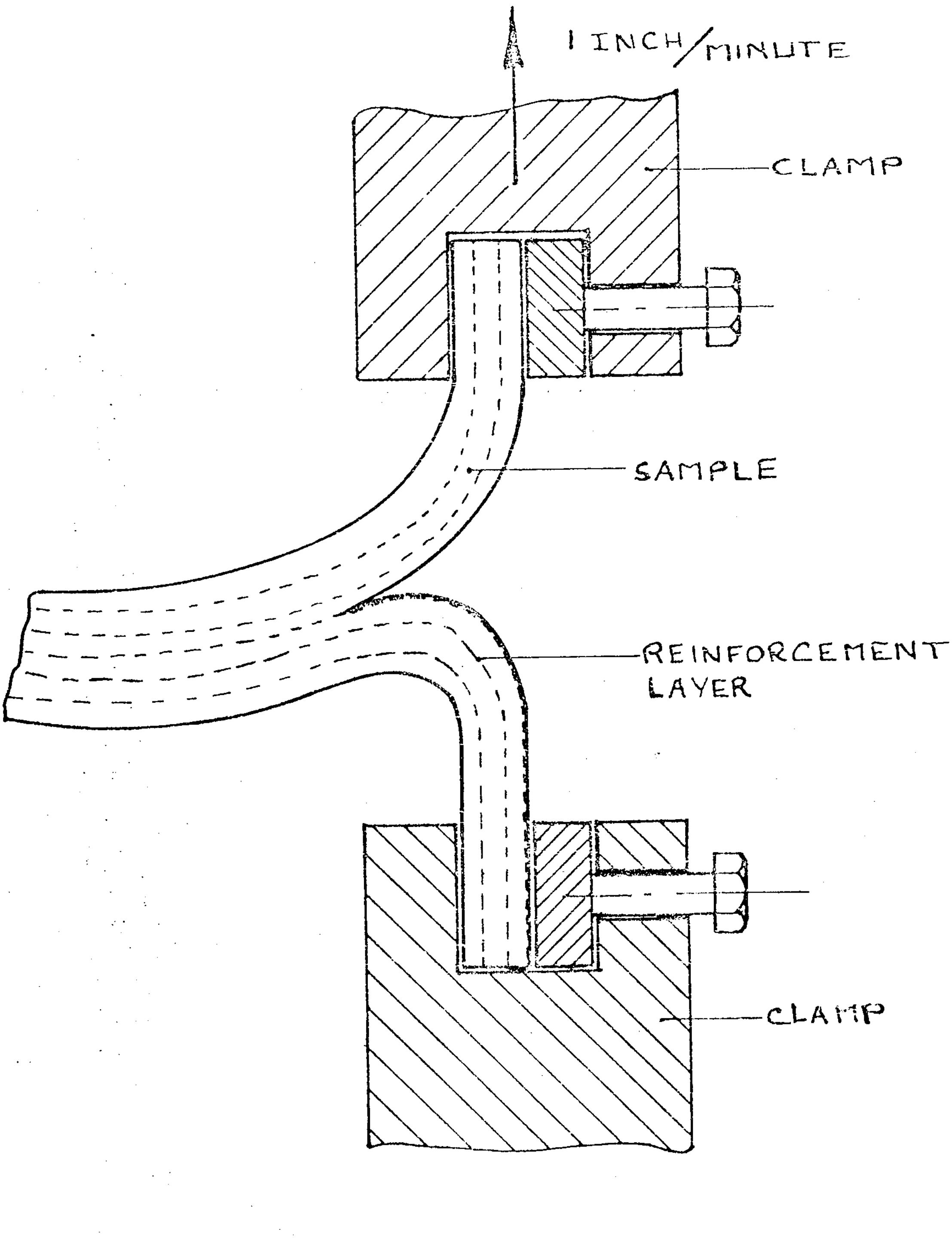


DETAIL OF FAILURE AT THE CONVEX SIDE



DETAIL OF FAILURE AT THE CONVEX SIDE

FIGURE 109



PEEL TEST

FIGURE 110

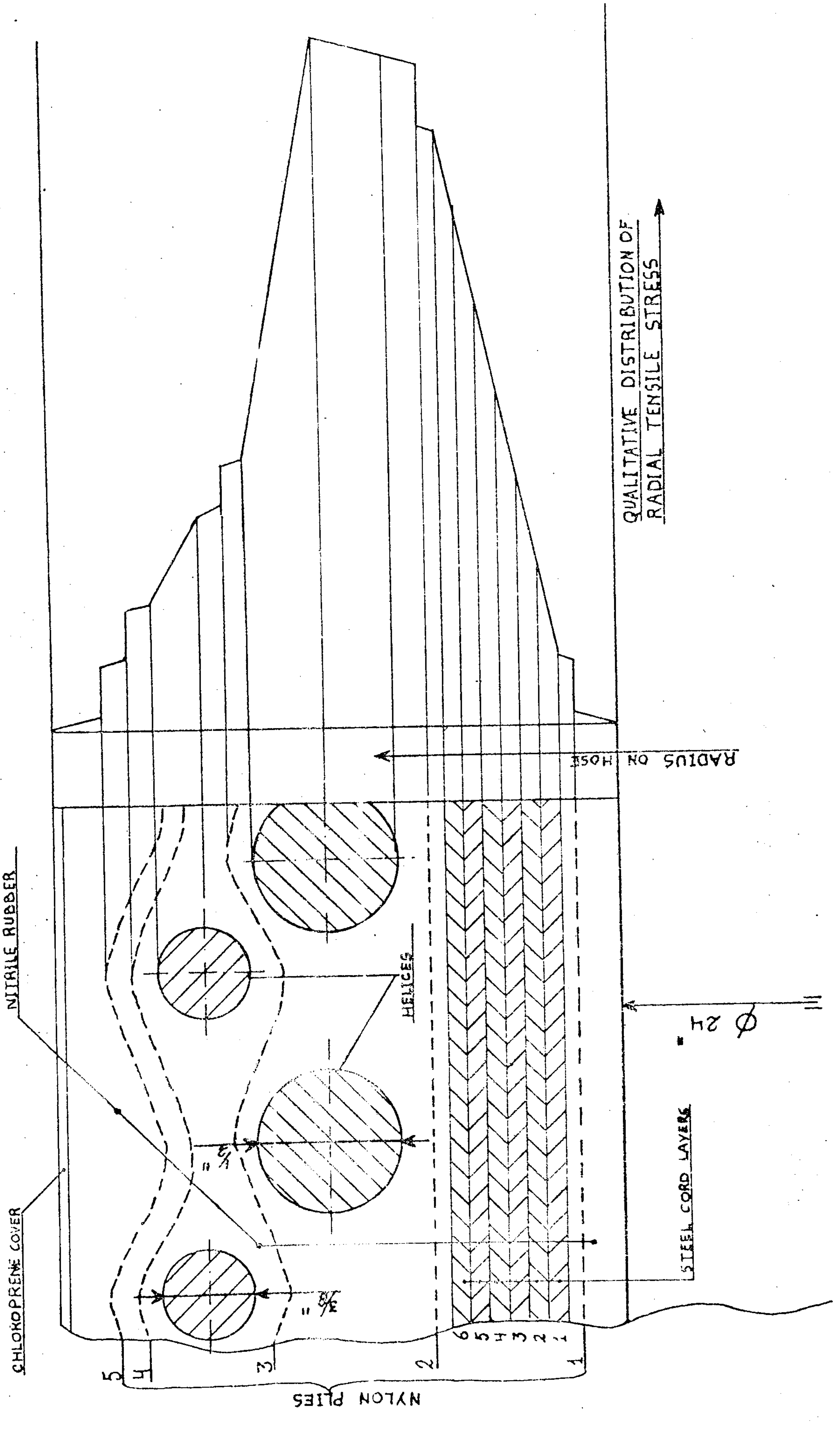
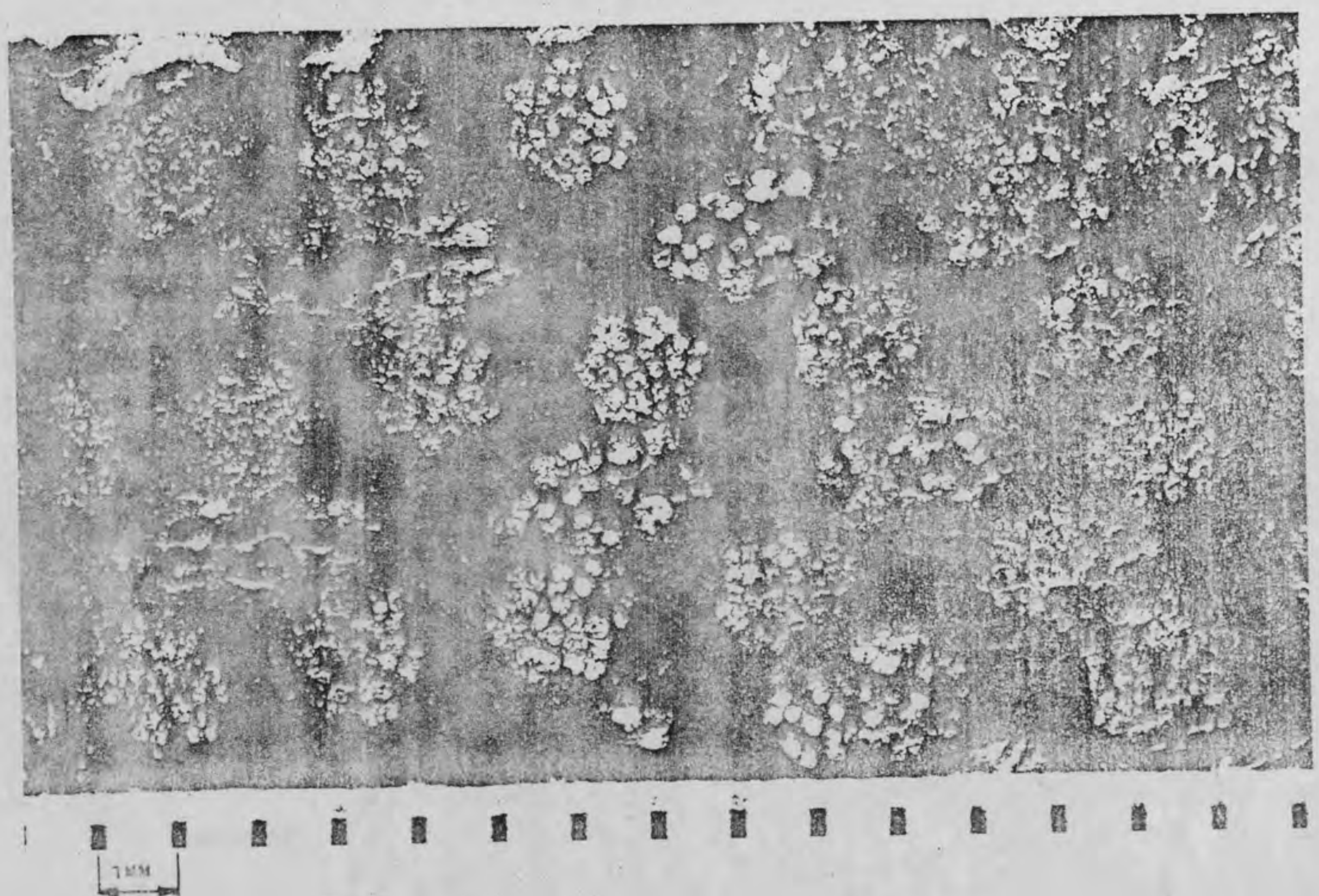
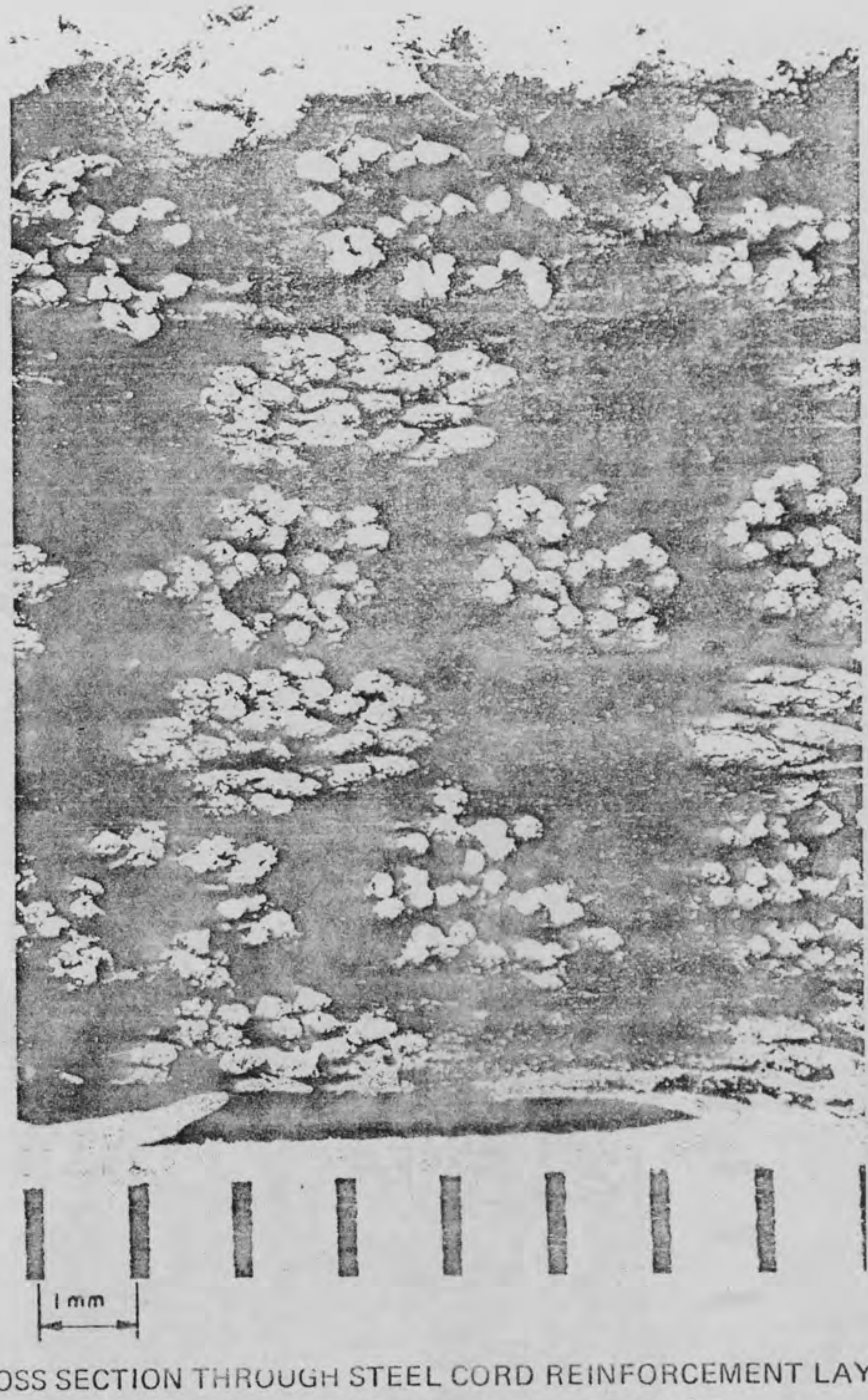

QUALITATIVE DISTRIBUTION OF RADIAL TENSILE STRESS
RADIUS ON HOSE
NITRILE RUBBER
HELICES
CHLOROPRENE COVER
STEEL CORD LAYERS
NYLON PLIES
Ø 24"

FIGURE 411

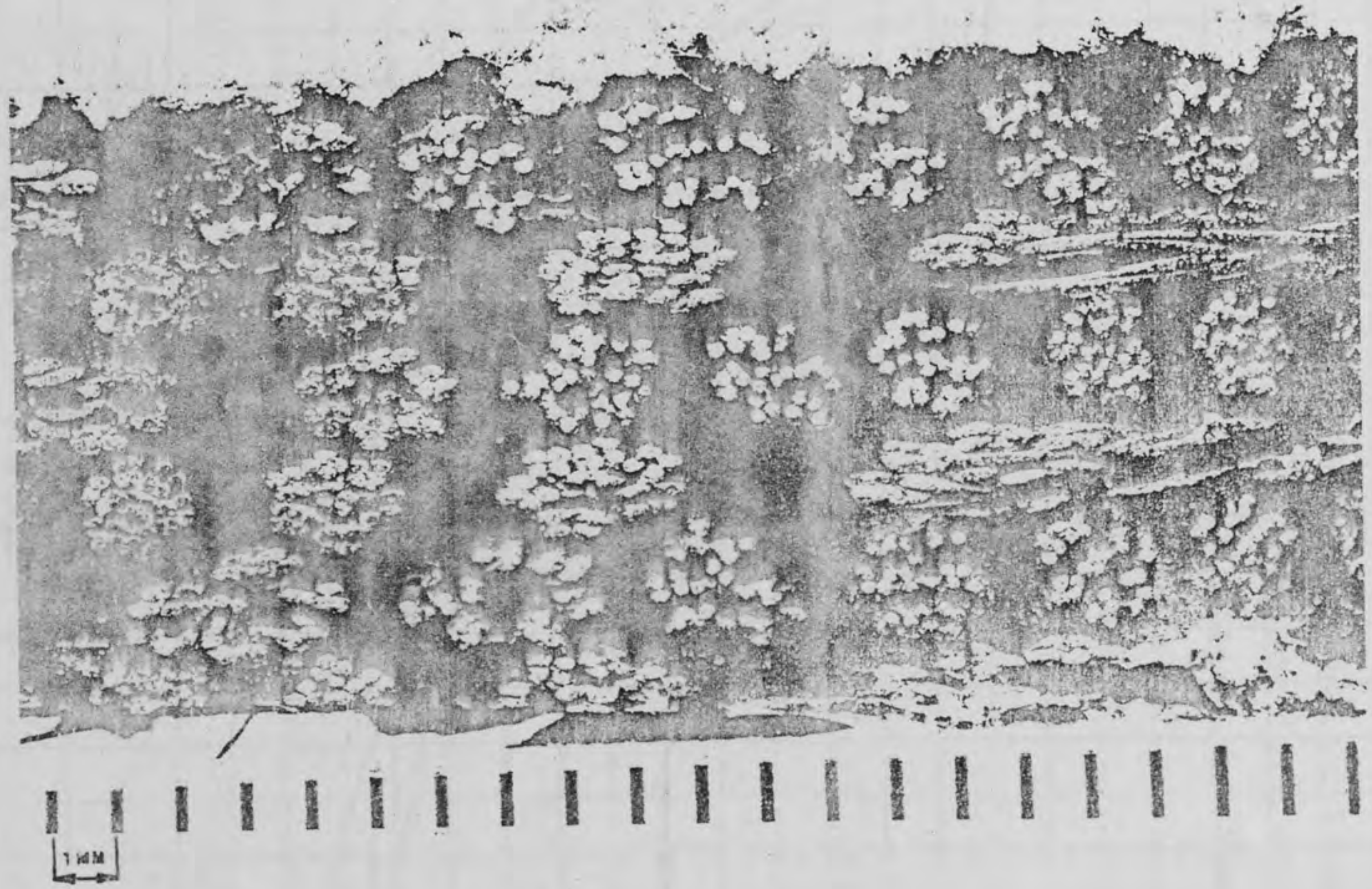


CROSS SECTION THROUGH STEEL CORD REINFORCEMENT LAYERS
AFTER ALTERNATING BENDING TEST SHOWING CAPILLARIES
IN STEEL CORDS AND THE SMALL DISTANCES BETWEEN CROSSING STEEL CORDS.
THE ABOVE SAMPLE WAS TAKEN FROM A SECTION BETWEEN 4.30 m AND 4.80 m
FROM THE DRIVEN END OF THE HOSE



CROSS SECTION THROUGH STEEL CORD REINFORCEMENT LAYERS
AFTER ALTERNATING BENDING TEST
SHOWING CAPILLARIES IN STEEL CORDS
AND THE SMALL DISTANCES BETWEEN CROSSING STEEL CORDS.
THE ABOVE SAMPLE WAS TAKEN FROM A HOSE SECTION
BETWEEN 4.30 m AND 4.80 m FROM THE DRIVEN END OF THE HOSE

FIGURE 113



NEARLY LONGITUDINAL SECTION THROUGH STEEL CORD REINFORCEMENT LAYERS
AFTER ALTERNATING BENDING TEST SHOWING CAPILLARIES IN STEEL CORDS
AND THE SMALL DISTANCES BETWEEN CROSSING STEEL CORDS.
THE ABOVE SAMPLE WAS TAKEN FROM A HOSE SECTION
BETWEEN 4.30 m AND 4.80 m FROM THE DRIVEN END OF THE HOSE

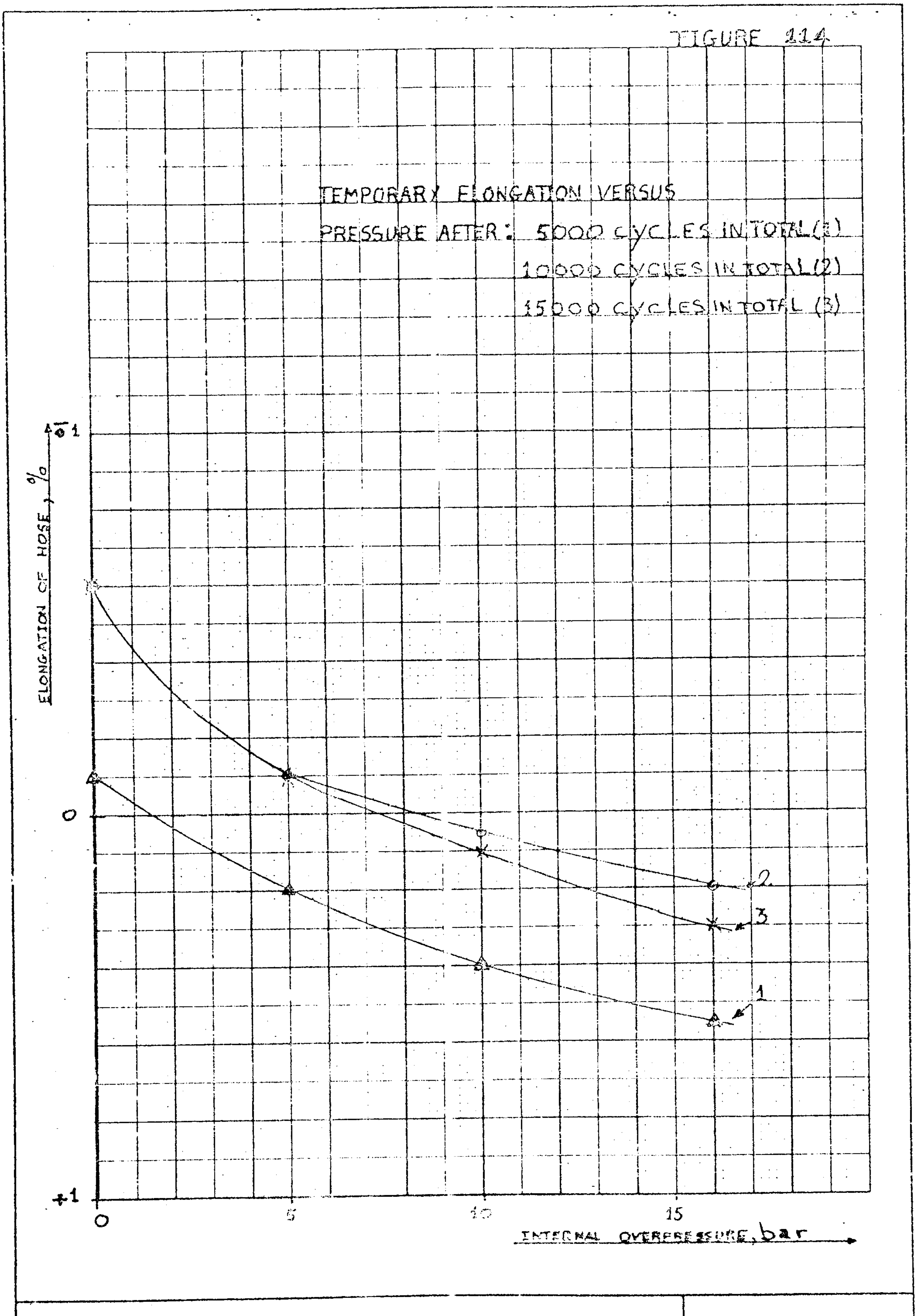
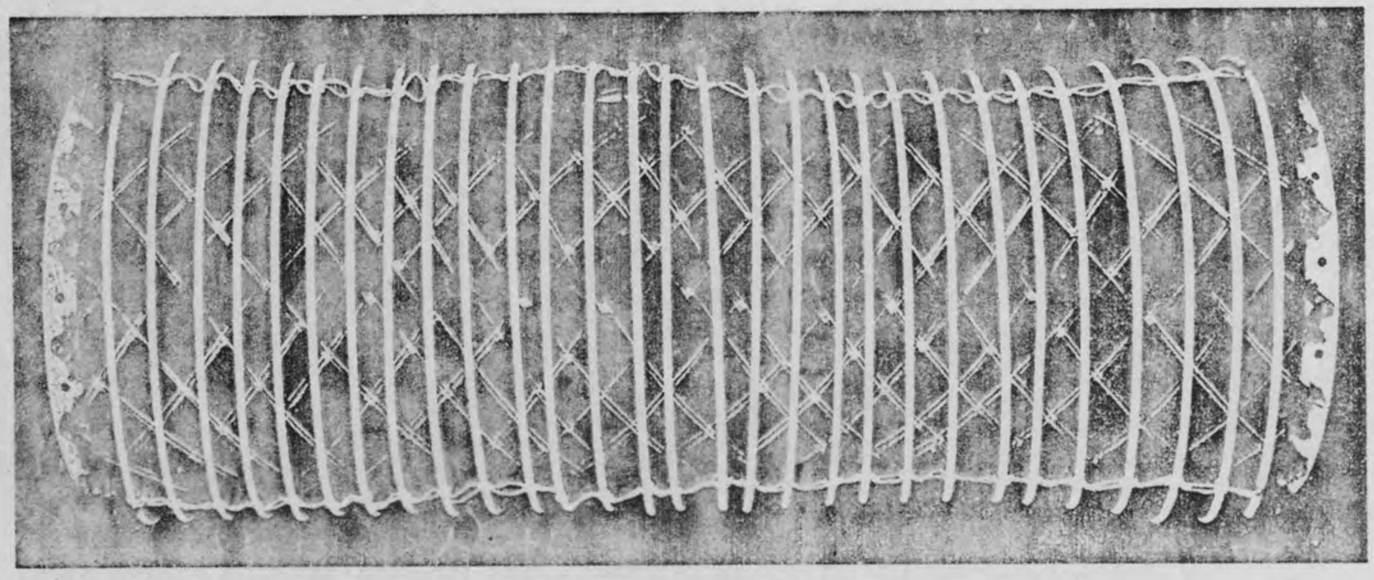


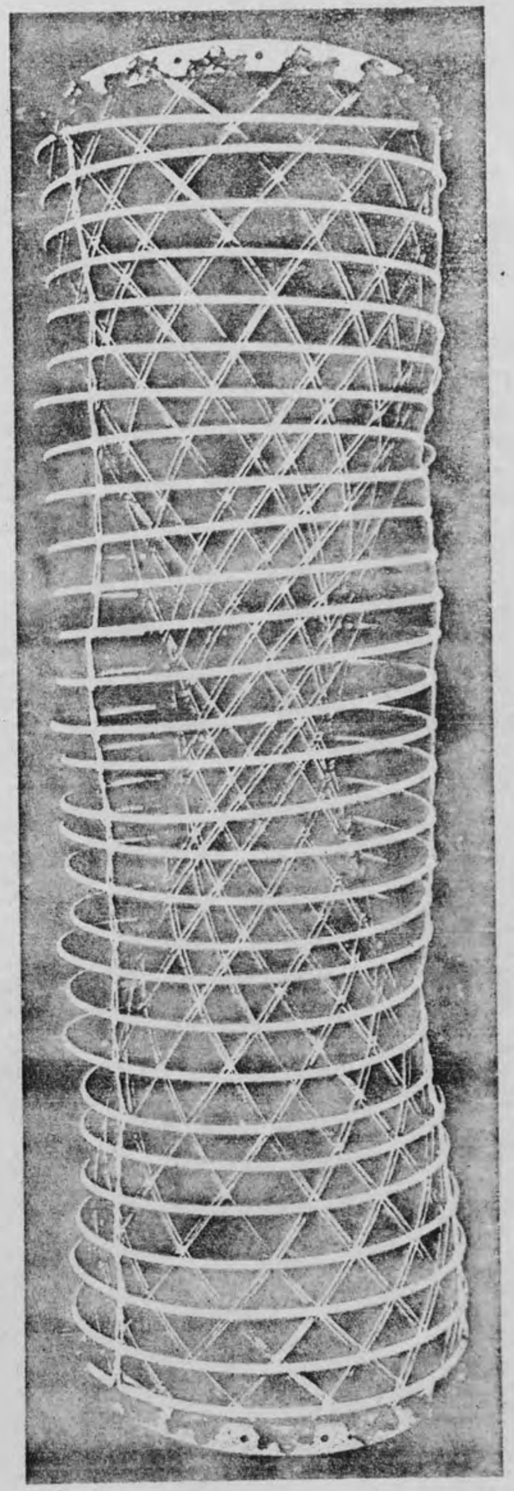
FIGURE 114
TEMPORARY ELONGATION VERSUS
PRESSURE AFTER: 5000 CYCLES IN TOTAL (1)
10000 CYCLES IN TOTAL (2)
15000 CYCLES IN TOTAL (3)
ELONGATION OF HOSE, %
1
0
1
0
5
10
15
INTERNAL OVERPRESSURE, bar
1
2
3

FIGURE 115



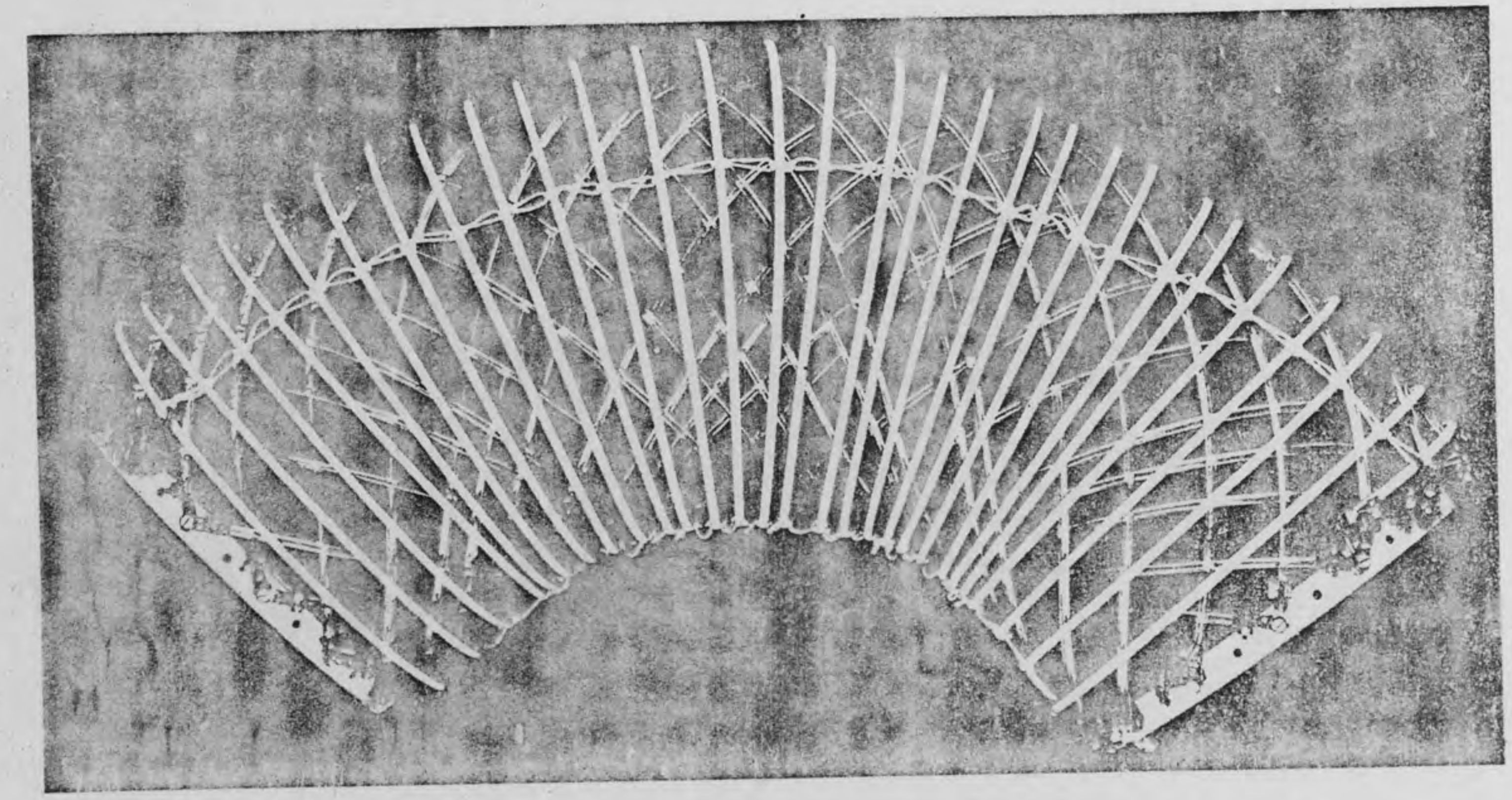
HOSE MODEL – IN NEUTRAL POSITION – CONTAINING A HELIX
AND TWO OPPOSITELY WOUND REINFORCEMENT WIRES
WHICH FORM THE HOSE CARCASS

FIGURE 116



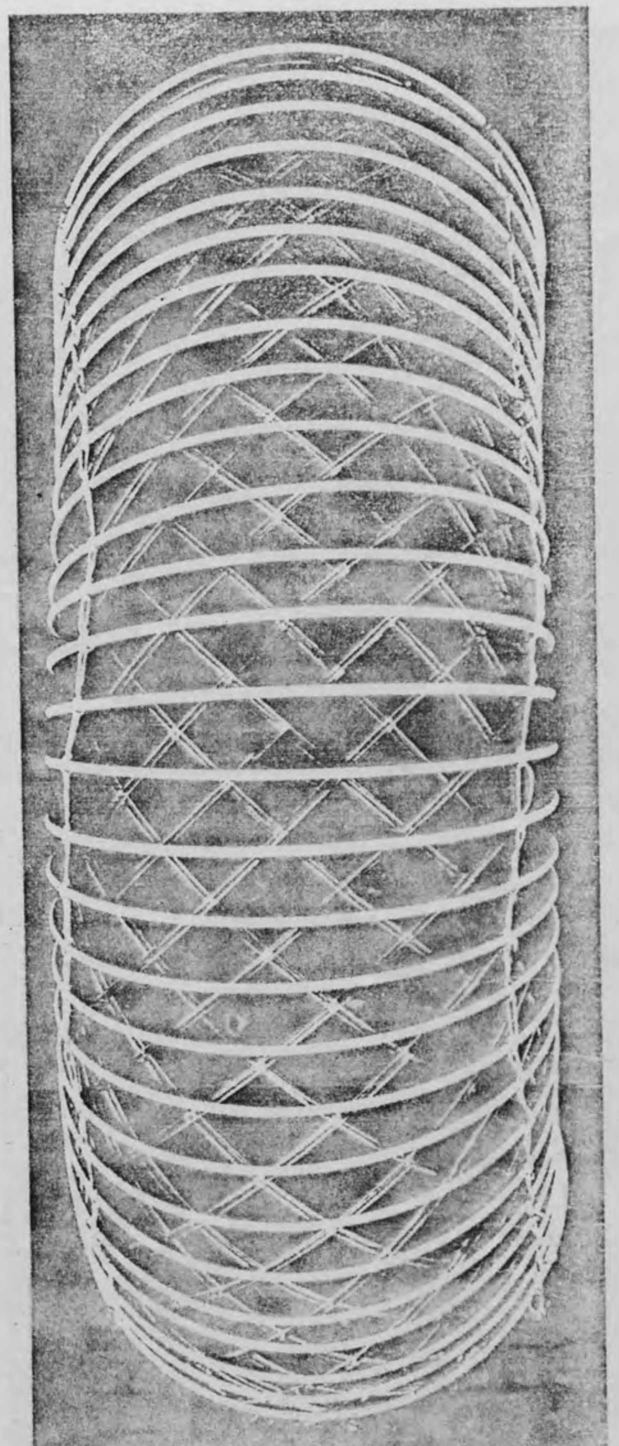
HOSE MODEL UNDER TENSION; NOTE THE DECREASE IN THE CARCASS DIAMETER

FIGURE 117



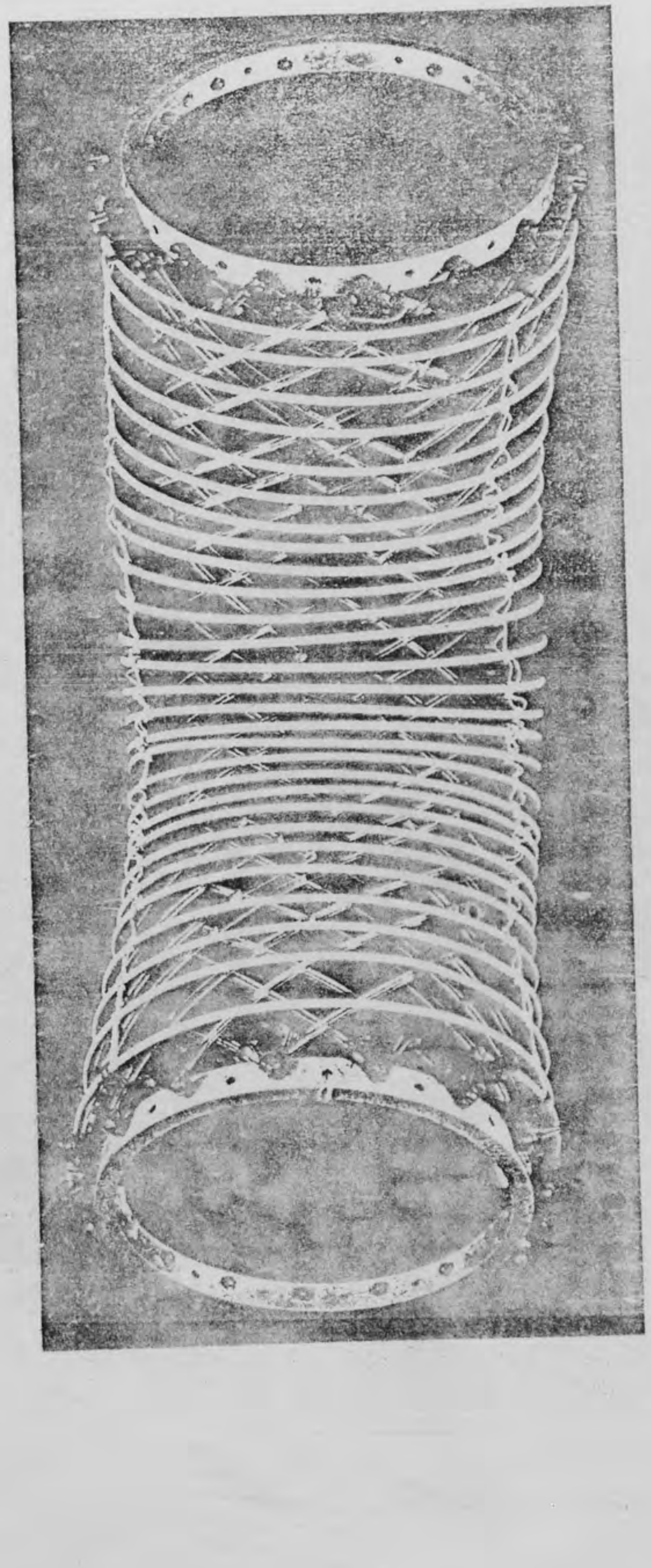
SIDE VIEW OF BENT HOSE MODEL; NOTE THE DIMENSIONAL DIFFERENCES
IN THE OBLIQUE REINFORCEMENT WIRE PATTERN
DUE TO MUTUAL SHEAR

FIGURE 118



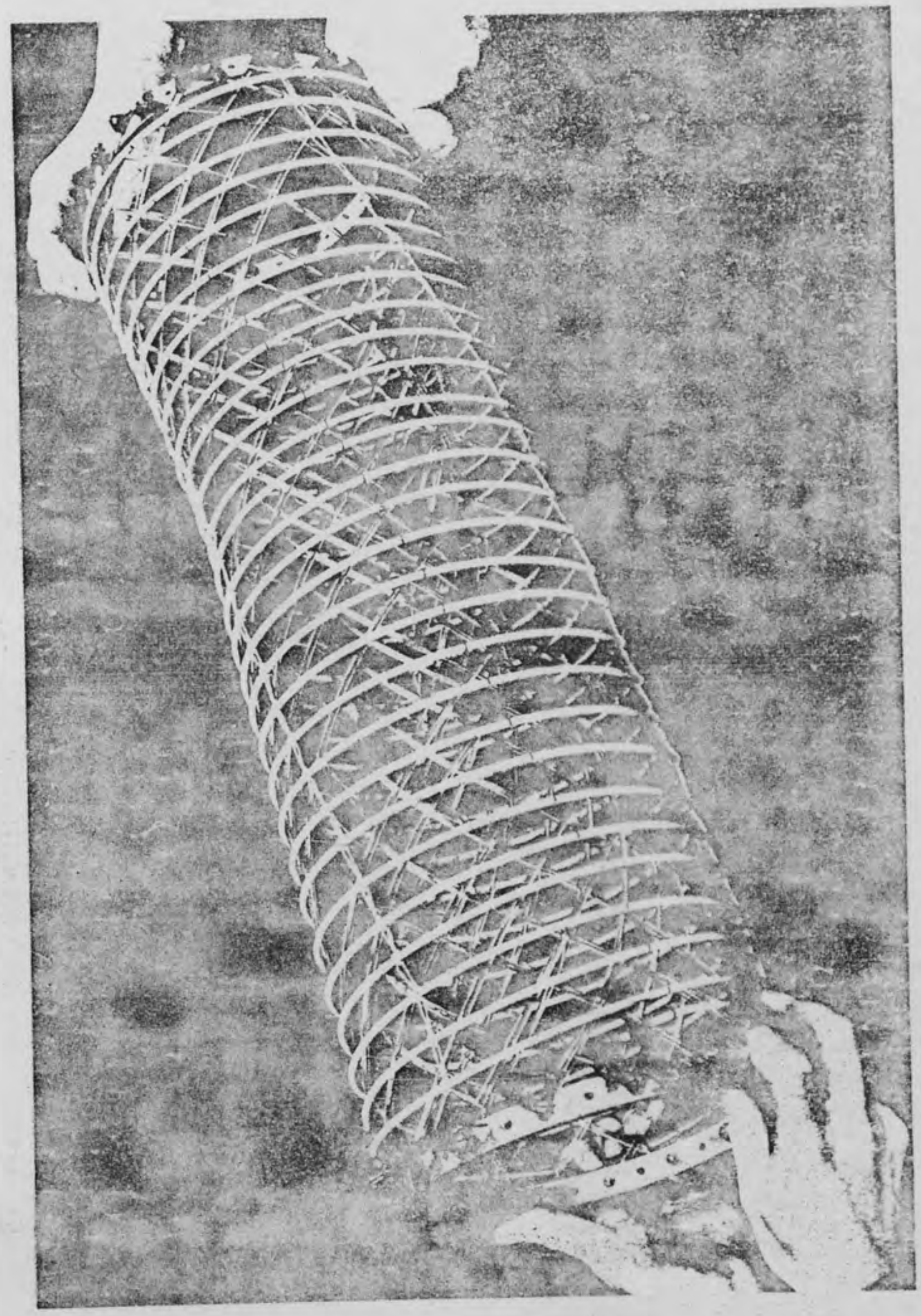
VIEW OF CONVEX SIDE OF BENT HOSE MODEL

FIGURE 119



VIEW OF CONCAVE SIDE OF BENT HOSE MODEL

FIGURE 120



RELATIVE DISPLACEMENT OF REINFORCEMENT WIRES IN HOSE MODEL UNDER TORSION