

EDINBURGH WAVE ENERGY DEVICE

MOORINGS

CURRENT STATE OF DEVELOPMENT

JANUARY 1980

EDINBURGH - SCOPA - LAING

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## 1 Introduction

These notes are intended to bring together the various aspects of mooring of the Edinburgh Device known at the present time.

Model tests indicate that the mooring forces for the device are substantially below those of other floating devices. That the device can be moored has not been questioned and consequently the manner in which this is to be achieved has not had high priority in the past.

However in order to arrive at an overall scheme cost it has been necessary to produce a mooring system as a reference design and this is described in the following pages.

The next important stage in the development of the moorings will be an input of data on loads and excursions when testing of a long string model in the wide tank commences.

It is considered that results of tests are unlikely to cause substantial changes to the mooring specification and that the system outlined will be viable. Consequently it is intended to seek further advice and comment from informed sources on the manufacture, installation, maintenance and inspection of such a system.

## 2 Loads & Excursions

### 2.1 Mean mooring force

Mooring force is defined for the purpose of this discussion as the mean surge force. The value derived for an irregular sea state will be influenced by the period over which the surge force is averaged. However if a sufficiently long interval is taken it is found that a unique value can be established for any given sea state and this may be defined as the long term mean mooring force. It has been shown by Longuet Higgins that the value of the mooring force  $F = \frac{1}{4}\rho g \left[ (A_{\text{incident}})^2 - (A_{\text{reflected}})^2 - (A_{\text{transmitted}})^2 \right]$  - 1

It is apparent from this relationship that the force is minimised by reducing the amplitude of the reflected wave and increasing the amplitude of the transmitted wave. At sea energy levels below the device power limit an increase in the transmitted wave would represent a loss of energy and up to this level the strategy must be to reduce the amplitude of the reflected wave. At higher energy levels it is still beneficial to reduce the reflected wave but additionally the transmission of excess sea energy will reduce the mooring force.

If it were possible to make the reflected and transmitted wave amplitudes equal zero, i.e. achieve 100% efficiency, then  $F = \frac{1}{4}\rho g \left[ (A_{\text{incident}})^2 \right]$  - 2  
Since to achieve any power extraction  $A_{\text{incident}}$  must be greater than zero then it follows that an external force must be applied to the device.

$$\text{The power level in the sea} = \frac{\rho g^2}{4\pi} (H_{\text{RMS}})^2 T_e \quad - 3$$

If the wave were a regular sine wave equations 2 & 3 give

$$F = \frac{2\pi}{g} \frac{P}{T_e} = 0.64 \frac{P}{T_e} \quad - 4$$

For values of  $P = 7500 \text{ w/m}$  &  $T_e = 10 \text{ secs}$

$$F = 4.8 \text{ KN/m}$$

Hence/

Hence the minimum mean mooring force that one may expect to achieve at a power level of 75KW/m is approximately 5KN/m. The above calculation is obviously a gross simplification of actual conditions. In particular it should be noticed that equation 1 is independent of period, while the power level, equation 3, is not. Hence it is possible to have zero mooring force yet have power extraction (i.e. a lower power level behind the device than in front). To achieve this effect requires that the waves transmitted should be of higher frequency than the incident waves and reflected waves should be minimal.

Experimental work has, to the present time, been conducted in the narrow tank with a few qualitative observations on cylinders in the wide tank. The narrow tank results covered in the 4th year report were derived by averaging the surge force sampled at 10Hz over a period of 51.2 secs (at model scale) at a number of sea states covering the India Scatter Diagram (see Fig 1) and with various duck and rig parameters. A summary of results is shown in Fig 2. The conclusions drawn from these tests (reported p2.73 - 2.79) were (values quoted are for 15m duck):

1. "Mooring force . . . . is much smaller than the sinking force."
2. "Like the sinking force it has a 'leftover' force of 20KN/m at small wave amplitudes." (9KN/m for 10m duck)
3. "Mooring forces are fairly independent of torque limit."
4. "The shorter seas showed somewhat greater mooring forces than the longer seas."
5. "Moving rig tests showed similar results up to Hrms = 1.5m but an actual reduction in mooring force above this level." (1.0m for 10m duck).
6. "All forces in these tests have been in the direction of wave propagation, but some tests on cylinders and observations in our wide tank have shown that mean forces can be in the opposite direction to the wave travel."

## 2.2 Peak mooring force/

## 2.2 Peak mooring force

The peak mooring force is more difficult to define and establish than the mean force primarily because it is a function of the mooring system's characteristics. If the mooring system was infinitely stiff the peak instantaneous loading on the mooring would be the peak instantaneous wave loading. At the other extreme if the the mooring was infinitely compliant there would be no mooring load nor indeed any restraint. Any practicable mooring system must fall between these two extremes and a generalisation can be made that the stiffer the mooring system the higher the loads it will be required to sustain. However it must be recognised that in an irregular sea the device is being subjected to loading variations. The range of variation depends on the period over which the loading is averaged. These fluctuations in loading will produce oscillations in the mooring system. The more compliant the mooring system the more significant these oscillations become and they will cause mooring loads to rise above the long term mean value. Further experiments have been made this year in the narrow tank to provide more data on the variation of mean mooring force with averaging period.

These tests, made on the moving rig for various values of  $H_{rms}$  and two values of  $T_e$ , show the same pattern of long term mean values but also indicate that the range of mean force increases as the averaging period is reduced. Results of these tests are shown in Figs 3 to 12 inclusive.

## 2.3 Relationship of backbone and mooring loads

The foregoing discussion is a consideration of mooring loads on a short length of backbone and it is necessary to consider the effect this has on mooring loads. As in most matters concerning the backbone it is necessary to have a knowledge of the crest length of the seas and this is at present an area of uncertainty. To advance the discussion we must make some assumption and first consider the case of parallel wave crests parallel to the /

the backbone. This is the wave pattern modelled in the narrow tank with the device mounting modelled to represent support from adjacent sections. However the only restraint to backbone movement at any point under the conditions described is that provided by the mooring. This would be extremely unattractive from the point of view of power generation and it is envisaged that action would be taken to avoid this situation by aligning the backbone in a "zigzag" configuration so that at any moment some point on the backbone would be at a wave crest and to both sides of this point the backbone would be in a trough. This would ensure that the wave forces are substantially balanced within the backbone and the mooring is required to provide only the out of balance restraint. This is therefore a situation which may be considered as normal working in small to medium seas when the wave loading is below that required to exceed the yield moment of the joint in the backbone. The wave loadings are averaged within the backbone and the mooring provides the out of balance restraint. At some time either by reason of increased wave height or increased crest length the loads in the backbone will exceed the joint yield moment. Under these conditions the mooring load will rise above the mean value and there will be a local displacement, the relationship between the two depending upon the stiffness of the mooring system and the level of damping applied at the joint.

#### 2.4 Relationships of duck damping and mooring loads

As mentioned previously the magnitude of the mooring force is related to the amplitude of the reflected waves. While it is normally considered undesirable to reflect energy as this is then lost to the device it would be possible to invoke this effect by adjustments to the damping characteristic of the duck should it be necessary to increase the backward force. This may prove of assistance to counteract the observed phenomenon of cylinders drifting into the waves in certain sea states and also in damping oscillations due to the slow variation of mean mooring force.

## 2.5 Longshore effects

The moorings will be required to provide restraint against longshore currents.

Assuming a current of 1½kt (0.77m/sec) acting on the full end area of a duck (53m<sup>2</sup>) the force induced would be

$$\begin{aligned} F &= \frac{1}{2} \cdot \rho \cdot V^2 \cdot A \cdot C_d \\ &= \frac{1}{2} \times 1000 \times 0.77^2 \times 53 \times 1.0 \text{ N/duck} \\ &= 1.6 \text{ tonnes/duck} \end{aligned}$$

As the proposed mooring system has one mooring line to a pair of ducks, each mooring line must provide 3.2 tonnes of sideways force.

Figure 13b shows that a force of 3.2t will produce a sideways displacement of approximately 8m and this is practically constant over a considerable range of fore and aft movement of the spine. At the extremes of movement the offset is reduced.

It is unlikely that all ducks will be equally loaded and the backbone will be required to spread the applied loads to the mooring lines. However these forces are insignificant compared to the assumed design axial force of 1000 tonnes derived from consideration of possible acceleration forces.

Other factors which must be given consideration are additional forces due to skin friction and the effects of currents on the buoys and sinkers of the mooring system.

However these are likely to be small and it appears that the proposed mooring system will adequately resist lateral forces.

3./



### 3. Consideration of Mooring Systems

Two further items influenced the choice of mooring system. The first was the depth of water in which the device was to be moored. This was initially thought to be 60m but due to changes in the transmission system it became economically viable to take advantage of higher power levels further offshore and the water depth now being considered is 100m. The other item was the fatigue characteristic of likely rode materials. The data available showed the substantially increased lifetimes which was obtained by minimising load variations.

Other items which are relevant but have not yet been defined are sea bed conditions and the arrangement for transmission of electricity from the device to the sea bed. The former influences only the type of anchor to be provided and is a matter only of data collection. The latter is of greater consequence since this influences the extent of excursions of the device which can be tolerated. Flexible cables of the voltage envisaged (132 KV) for submarine duty have yet to be developed and there is consequently no guidance from current practice as to what may be achieved. However it is considered that such a cable can be developed and that providing the cable is constrained to always move within defined limits no limitation need be placed on total displacement. This generalisation would need to be reconsidered when more definite costings are available but as it is envisaged that there will be only one connecting point for 68 ducks it is unlikely that mooring design would become limited by electrical connection requirements.

It is possible to envisage a situation where the power level in the sea exceeds the power limit of the duck and for reasons explained earlier the mean surge force is zero. However at lower power levels this will not be the case and it will be necessary to provide moorings to maintain the device within defined limits. However, the mean mooring force to be provided/

provided is low by comparison with other devices due to the low amplitude of reflected waves and the ability of the device to transmit any excess energy. The ducks are maintained in accurate relative location by the backbone so there is no danger of collision between adjacent devices. An electrical connection of unlimited excursion can be designed. The conclusion is that for this device there can be no justification for the provision of a stiff mooring and by so doing incurring high, fluctuating peak loads which give no benefits. Consequently attention has been directed to the provision of a mooring of high compliance which can sustain a maximum mean mooring load of 13KN/m, no limit being placed on excursion.

Two further decisions were made at this stage. One was to determine the spacing between moorings and the second the necessity or otherwise of rear moorings. The value of the former was established from work done on the backbone as being some multiple of 60m (this figure is now amended to 61m). Points considered were the effect of additional bending moment on the backbone and the consequences of the loss of a mooring line. The decision was made to adopt a spacing of 60m but is open to review when loadings and costings are better defined. Regarding rear moorings it was considered that these were essential to prevent oversailing of the front moorings and to provide restraint to forces acting along the line of the backbone. However it was considered that the loading on the rear moorings should be reduced as this also has advantages in reducing the overall spring rate of the system.

Thought has been given so far to two possible systems. The first is essentially an active system utilising a constant tension winch of novel construction, the cylindrical backbone section acting as a base for the coiling of the rode. This concept has not been pursued to conclusion but was suspended when the possible required winding speeds were established. The second system, which forms the moorings in the reference design, is passive being based on buoys and sinkers. The particular configuration proposed is shown in Figs 13 and 14 and was arrived at by fixing  $l_1$  to give 10m/

10m clearance to mean sea level,  $l_2$  to give 10m clearance to sea bed and  $l_3$  to equal  $l_1$  to give maximum range of movement. If  $L = (l_1 + l_2 + l_3)$  then range of movement of one side =  $\sqrt{L^2 - h^2}$

If spacing between anchor points =  $S$ , range of movement for complete system

$$= 2\sqrt{L^2 - h^2} - S^2$$

Thus for maximum range of movement  $S$  must be minimised. However if  $S$  is less than  $\sqrt{L^2 - h^2}$  then the device will oversail an anchor. Hence the distance chosen for  $S$  was 300m to give clearance between buoy and device at maximum travel. For a minimum total weight of buoy and sinkers there should be maximum differential between front and rear sinkers. The limit to this is the need to ensure that the no load position is within the range of travel and tensions in the rear rode are not excessive. A range of 4:1 has been used in the present proposal.

The actual value for the weight of buoys and sinkers was determined from the need to achieve a peak mean load of  $60 \times 1.3 = 78$  tonnes at a displacement which would still allow a reserve for displacement due to individual wave action. The front buoyancy and weights were fixed at 120t each.

Fig 13 shows the characteristics of the mooring system under static displacement. This graph also shows that the vertical component stays fairly constant until large displacements are reached. This will be beneficial by tending to move the duck downwards, away from peak wave action. (Although this load at approx. 1t/m) is well below the reserve buoyancy of approximately 5t/m). The graph also shows that because of the pretensioning effect of the buoy and sinker the tensions in the rodes do not fluctuate markedly over a wide range of displacement.

The spring rate of the system from 0 - 50 tonnes is approximately 100N/m/m and at 80 tonnes is around 300N/m/m.

As/

As mentioned under the section referring to peak mooring forces a compliant mooring system will be subjected to oscillations due to the fluctuation of the mooring force. Using data obtained from this years tests at varying averaging periods some very much simplified calculations have been made to assess possible effects. These calculations show that in small seas the mean forces are insufficient to cause excessive displacements and in large seas although the amplitude of the oscillations is increased the mean point moves towards the front anchor so permitting the displacement to take place within the range of the mooring. However in intermediate seas (particularly  $H_{rms} = 1.1m$   $T_e = 13$  secs) the mean mooring force is high and a large amplitude has to be superimposed on this. Whether this produces overloading of the moorings depends on the assumptions made for the initial conditions of the backbone displacement, velocity and acceleration. For reasonable assumptions the mooring appears satisfactory but under adverse conditions problems will arise. These calculations are inadequate to justify the viability of the system but they serve to indicate possible problem areas.

#### 4 Mooring Components

##### 4.1 General

As the mooring forces for this device are low relative to other devices and are within the range of the properties of available materials it is necessary to decide whether it is more economic to achieve long life in the components or to work materials at higher stresses accepting that maintenance or replacement will be necessary at some stage. In view of the likely circumstances there can be little argument in favour of the latter course of action and attention was directed to ways of achieving the maximum life from the mooring.

##### 4.2 Rodes

Initially consideration was directed to the use of wire rope. Discussions took place with British Ropes, who were in general terms surprised to find how small the loads were which were being specified. These discussions took/

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took place at an early stage in the formulation of the design but the loading has not changed appreciably. A peak tension of 150 tonnes was quoted to British Ropes together with some assessment of the number of occurrences of loads of lesser value. British Ropes recommendation was for a rope of 750t minimum breaking load which would be 108 mm diameter. They expressed the view that tension fatigue would not be a problem but potential bend fatigue situations would have to be designed out of the system to achieve a fatigue life of 25 years. They further stated that a wire rope suitably protected will have a corrosion life of 25 years. The possibility of using novel forms of rope construction and materials was discussed with a view to achieving increased life but British Ropes saw little merit in following this as in their opinion present materials were adequate. These comments while encouraging are not necessarily generally accepted as there is very little data available on the combined effect of corrosion and fatigue loading on wire rope.

An alternative material which appears to avoid corrosion problems is the use of manmade fibres. In view of the data available on fatigue properties of these fibres in various forms of rope construction a parallel fibre rope of the "Parafil" type would be the preferred construction.

#### 4.3 Terminations

Whichever of these alternatives is selected it will be necessary to provide terminations to the rodes. These terminations must be capable of developing the required loads and also be of a form to provide adequate bend fatigue life by limiting the radius of curvature. Termination of wire rodes can be readily achieved by a potting technique but the termination of man made fibre ropes is more difficult to accomplish and they are a substantial cost item. Consequently the idea was evolved of using a continuous rode between anchors, the buoys and sinkers being clamped to the rodes at pre-determined points. This procedure would also allow greater freedom in the choice of distance between the various buoys, sinkers and anchors. A saddle would 7

would be provided common to all buoys and sinkers which would comprise a cable clamp and bend radius limiter shown on Fig 16.

#### 4.4 Sinkers

The primary requirement for the sinkers is that the material used should have minimum cost per unit of submerged weight. From prices quoted verbally by British Steel it would seem that advantage could be taken of the scrap steel resulting from the preparation of gyro discs from square plates and a possible arrangement has been shown on Fig 18 for the 120t and 80t weights. The scrap elements are bonded in an outer skin of reinforced concrete to provide corrosion protection, a smooth external profile and an attachment point to the cable saddle.

#### 4.5 Buoys

Buoys are required with net buoyancy of 120t and 30t to work at depths up to 40m. Early thoughts were directed to the use of a foam material supporting a thin external skin. However, discussions with manufacturers indicate that in the pressure range being considered the skin must be of a special flexible construction to conform to the elastic deformation of the foam, or alternatively the foam must be sufficiently rigid to prevent damaging skin deflections. The net result of these criteria is that the costs of buoys in this form of construction are at present prohibitively high. The preferred alternative is to fabricate the buoys in mild steel. These have not been designed as yet but Figs 16 and 17 show the salient features to be achieved in any design and indicate appropriately the dimensions which may be required.

#### 4.6 Anchors

The "anchors" are required to resist vertical forces arising from the geometry of the system in addition to the horizontal mooring forces. Initially it was thought that a deadweight anchor while being a solution lacking any pretence of elegance would be satisfactory for any sea bed condition hence permitting a mooring for all locations. However it was soon/

soon realised that this procedure would impose severe financial penalties both in fabrication and installation costs. Consequently a less versatile but more cost effective solution was sought. To refine the anchor design it is necessary to know the sea bed composition but very little data is at present available. The alternative approach is to define what anchorage would be used if the sea bed were of some particular type. Generalised costings produced by NEL for TAG 4 indicate that the costs of a drag anchor system are approximately one tenth that of a deadweight anchor system (WESC(78)MA.P14). To make use of a drag anchor it is necessary to ensure that the applied force is essentially horizontal. The Bruce anchor is perhaps most tolerant in this respect in that it will perform satisfactorily with the direction of pull at up to  $30^{\circ}$  from the horizontal. This anchor also has the merit of retaining its efficiency i.e. holding power over dry weight, even when turned full circle. The angle of  $30^{\circ}$  is below that required for the operation of the system envisaged. To satisfy the requirements of the anchor and the buoy/sinker system a further weight is introduced. A submerged weight of 80t is suggested at present with a distance from drag anchor to weight of 100m. More detailed appraisal may show that this weight can be reduced particularly if the distance from anchor to weight is increased to permit a greater lift of this weight for the same angle to the horizontal. Provided the weight is adequate to ensure it is not lifted until the device nears the end of its travel the maximum angle to the horizontal that the rode can attain is limited to approx  $16^{\circ}$ . The holding efficiency of an 8.8t Bruce anchor is reported as 44 in sand and 39 in mud. The load factor to be applied to the anchor is not laid down and there is a powerful argument to keep this at a low value so that a "failure" of the system manifests itself in dragging of the anchor, the other components of the system remaining intact. If this is accepted the anchor referred to above would appear to be of approximately the right magnitude./

magnitude. The use of a Bruce anchor in clay appears to be more problematical possibly due to some confusion in the description of soils. It may be necessary to modify the anchor details or utilise an anchor of different configuration in these conditions.

In the case of rock, drag anchors are not a feasible proposition and the only apparently viable solution would be the use of a piled anchor. Recent information indicates that the costs of such an anchor may be less than had previously been quoted but the pricing will be very much influenced by the scale of operation envisaged. Careful attention will require to be given to the pile/rode connection to achieve long life.

#### 4.7 Duck to mooring connection

It is currently envisaged that the mooring will be installed prior to the arrival of the duck on station. The rode is located and raised either by attendant barges or equipment mounted on the backbone and is clamped to the backbone at a point between the two ducks on that section. It will be necessary to ensure that the rode will not come into contact with duck as the duck rotates about the backbone due allowance being made for the offset of the rode from the perpendicular due to longitudinal forces or backbone deflections. This may require a fin or inverted mast to be suspended beneath the backbone. More investigation is required before a decision is reached.

#### 5 Installation

As mentioned in the previous item it is envisaged that moorings will be installed prior to the arrival of the device. The configuration of the anchors will depend on the final alignment selected for the spine. Two alternatives are shown on Fig 14. It is assumed that at the time of installation the sea bed conditions will be known and a suitable anchor will have been determined. If the anchor to be used is a/



is a drag anchor it will be necessary to establish this by pulling-in with a force at least equal to the normal working force to give reasonably accurate location and assurance of satisfactory performance. The presence of buoys and sinkers could present handling problems during this operation and it may be preferred to install the anchors with the rode clear of all attachments. The buoys would next be clamped to the rode followed by the duck attachment point, the 80t sinkers, the 30 ton sinker and finally the 120t sinker. Another possibility is to affix the 80t sinkers before lowering the anchor and pulling in via these sinkers.

These procedures must depend to some extent on the equipment required to handle the rodes and details such as clamps and anchors, nor would they be appropriate to pile anchors. These points will require to be resolved before the installation procedure can be finalised but they serve to indicate current thinking on the subject.

## 6 Inspection and Maintenance

The objective is to eliminate the necessity for human presence either to inspect or maintain any aspect of the device including moorings. This has not previously been defined as a specific objective. The requirement for permanent mooring of unmanned floating objects has only recently arisen and although such objects do now exist e.g. SPM devices the need for long life has been relegated by higher priorities.

Attention must be concentrated on time dependant properties of materials which may be identified as creep, fatigue (since load varies with time), corrosion, erosion and fouling. It must be remembered that these in combination may produce worse conditions than when acting seperately.

The actual details of how long life is to be achieved are still being considered but as has been indicated in the component sections the relatively/

the relatively low forces permit the use of high load factors while still utilising available materials. However high load factors represent the payment of a premium to cover areas of ignorance and it would be preferable to have more data available to reduce the extent of unknowns.

Fig.

1. 4th Year Report Scatter Diagram Tests Performed
2. " " " Mooring Force, Mooring Rig 1MNm/m Torque Limit
3. Surge Forces Time Averaged  $T_e = 10$  secs Hrms = 0.4m
4. " " " " " " " " Hrms = 0.6m
5. " " " " " " " " Hrms = 0.8m
6. " " " " " " " " Hrms = 1.0m
7. " " " " " " " " Hrms = 1.15m
8. " " " " " " " " Hrms = 1.25m
9. " " " " " " " " Hrms = 1.66m
10. " " " "  $T_e = 13$  secs Hrms = 0.7m
11. " " " " " " " " Hrms = 1.16m
12. " " " " " " " " Hrms = 1.4m
13. Mooring Configuration Rode Tensions and Mooring Force  
Against Displacement
14. Mooring General Arrangement - Plan
15. " " " - Elevation
16. " 120t Buoy
17. " 30t Buoy
18. " 120t and 80t Sinkers

STATION INDIA SCATTER DIAGRAM  
SHOWING TESTS PERFORMED

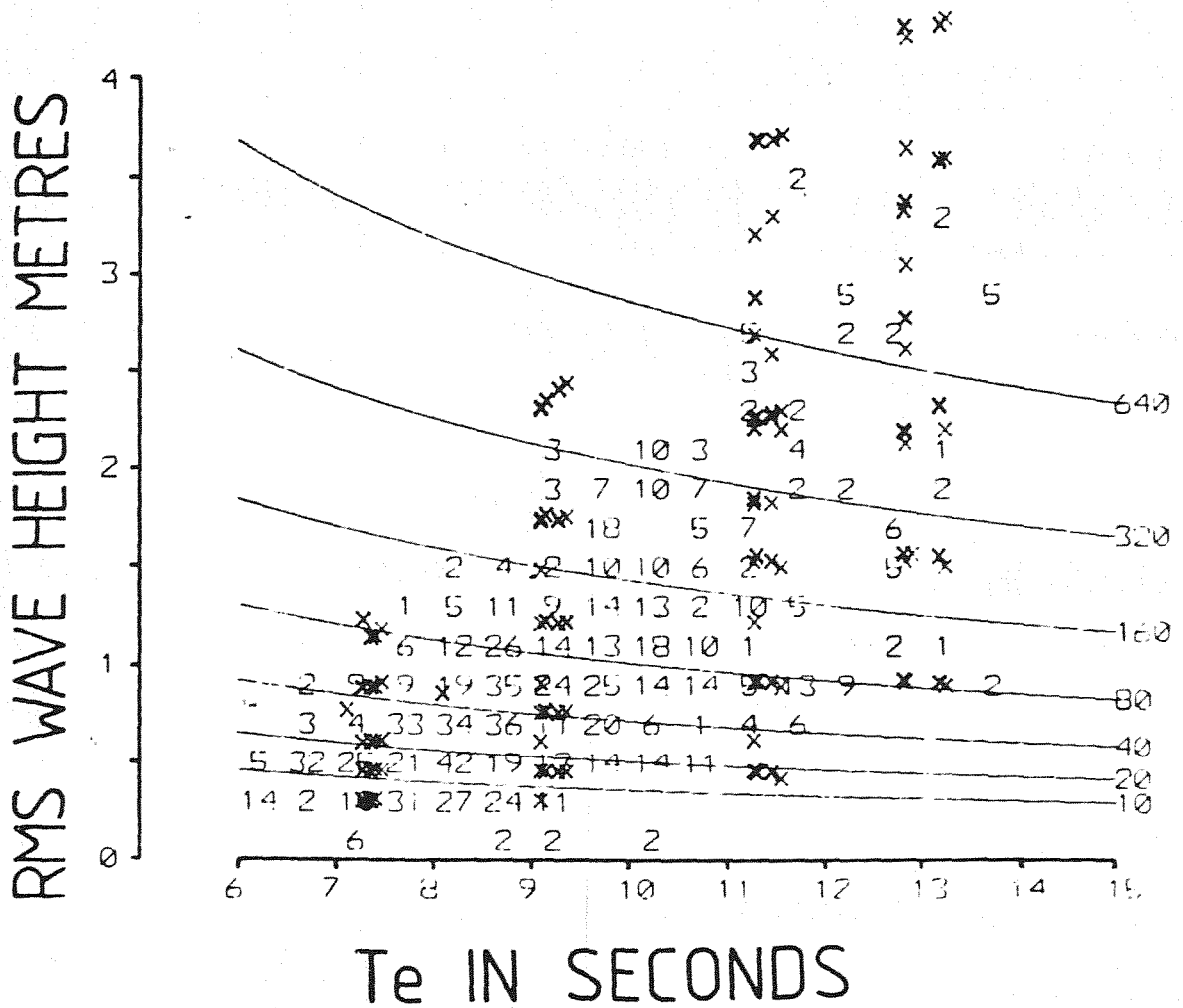


Fig. 1.

# Scatter Diagram Tests

## MOORING FORCE, MOVING RIG, 1MNm/m

### TORQUE LIMIT

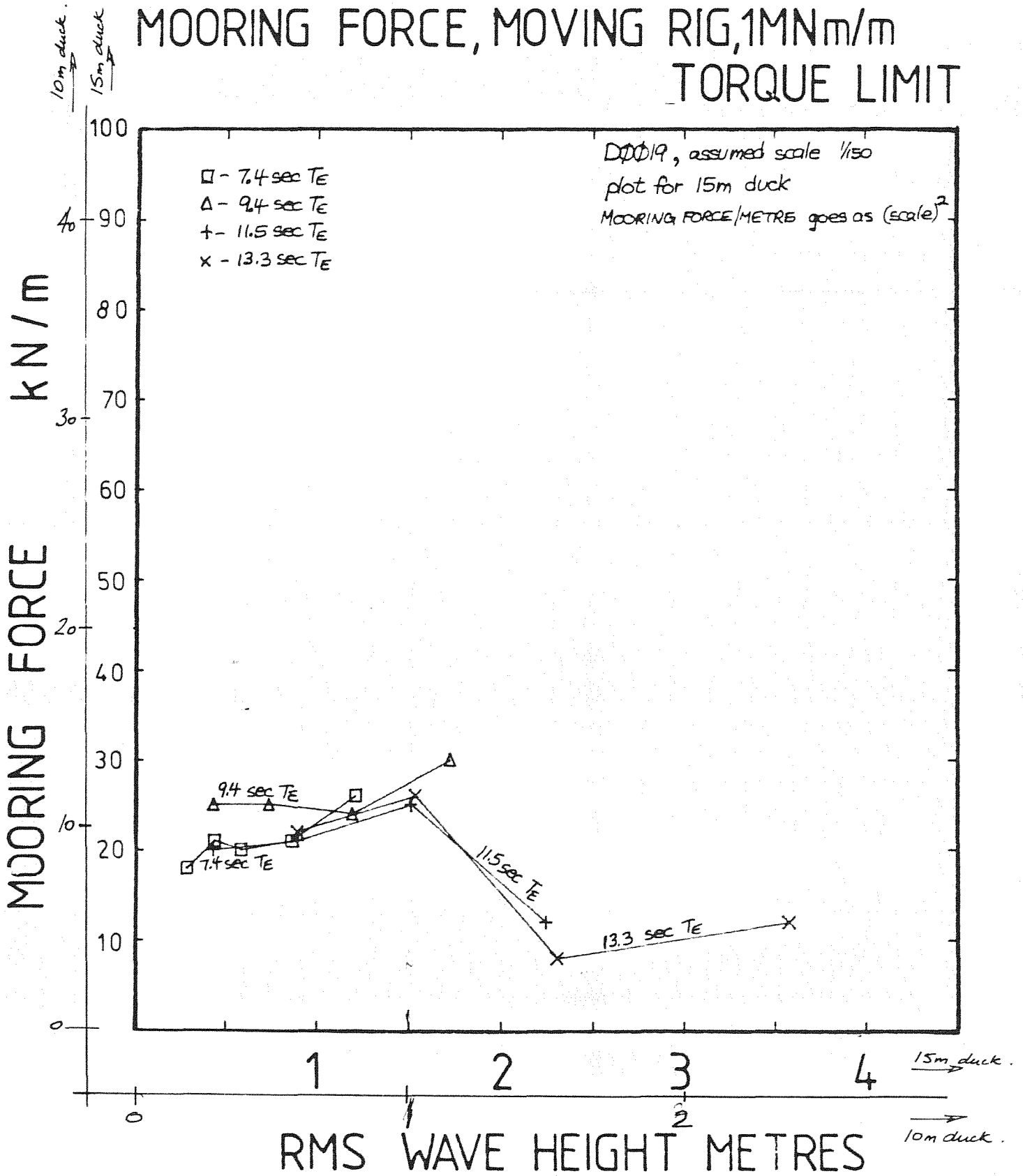
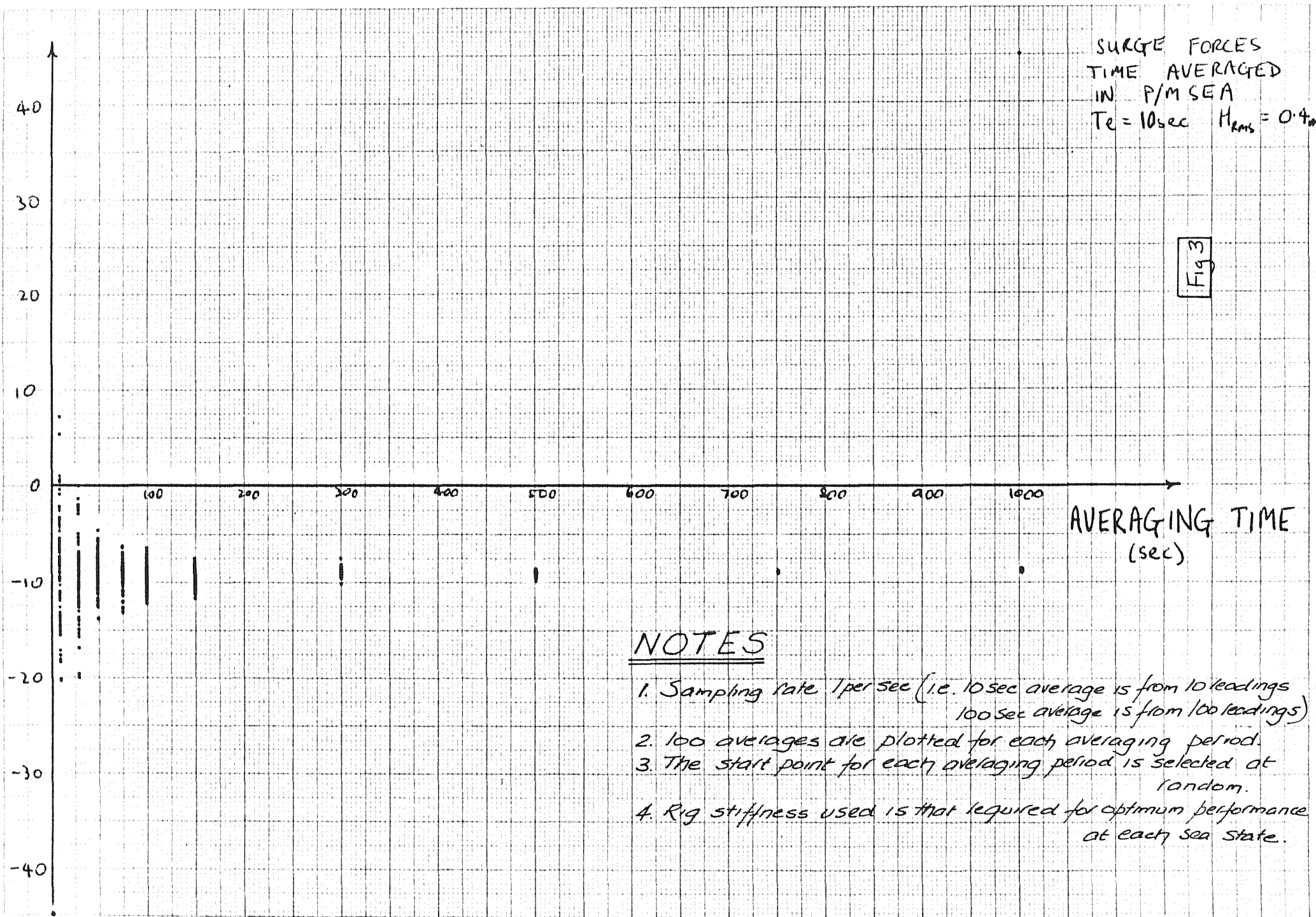


Fig. 2

SURGE WAVE HEIGHT (m/m)

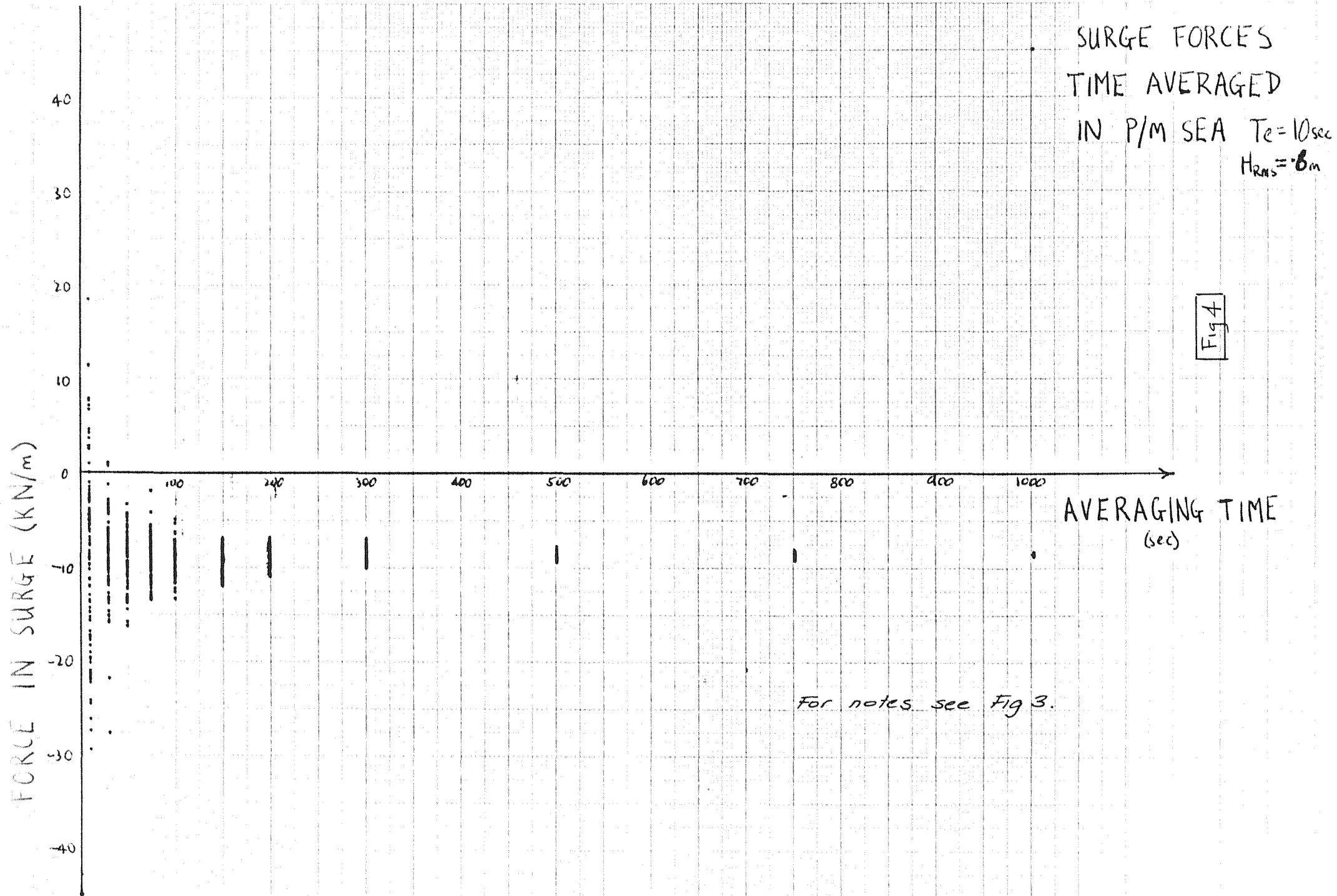


SURGE FORCES  
TIME AVERAGED  
IN P/M SEA  
 $T_e = 10\text{sec}$   $H_{rms} = 0.4\text{m}$

Fig 3

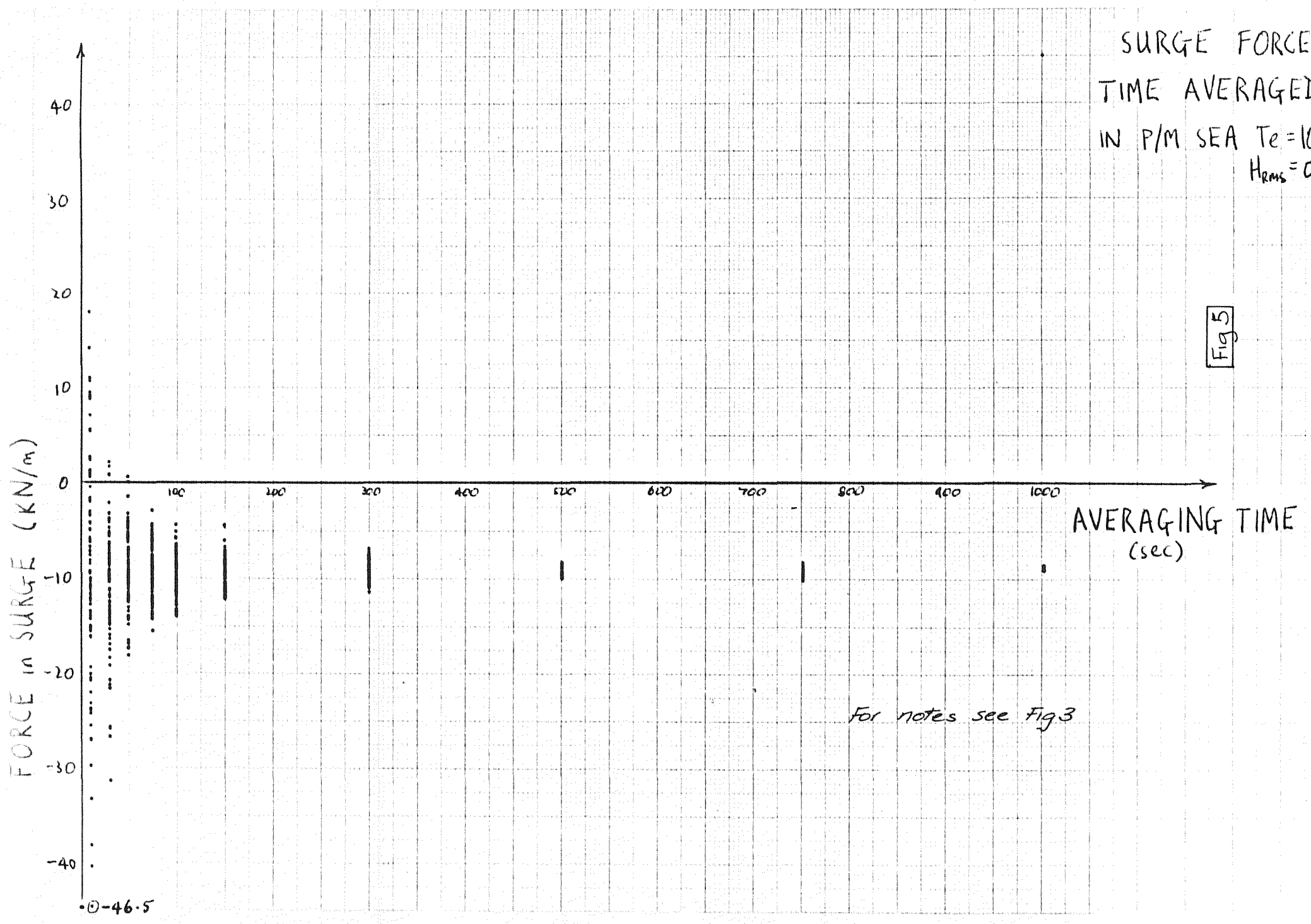
### NOTES

1. Sampling rate 1 per sec (i.e. 10 sec average is from 10 readings  
100 sec average is from 100 readings)
2. 100 averages are plotted for each averaging period.
3. The start point for each averaging period is selected at random.
4. Rig stiffness used is that required for optimum performance at each sea state.



SURGE FORCES  
TIME AVERAGED  
IN P/M SEA  $T_e = 10 \text{ sec}$   
 $H_{rms} = 0.8 \text{ m}$

Fig 5



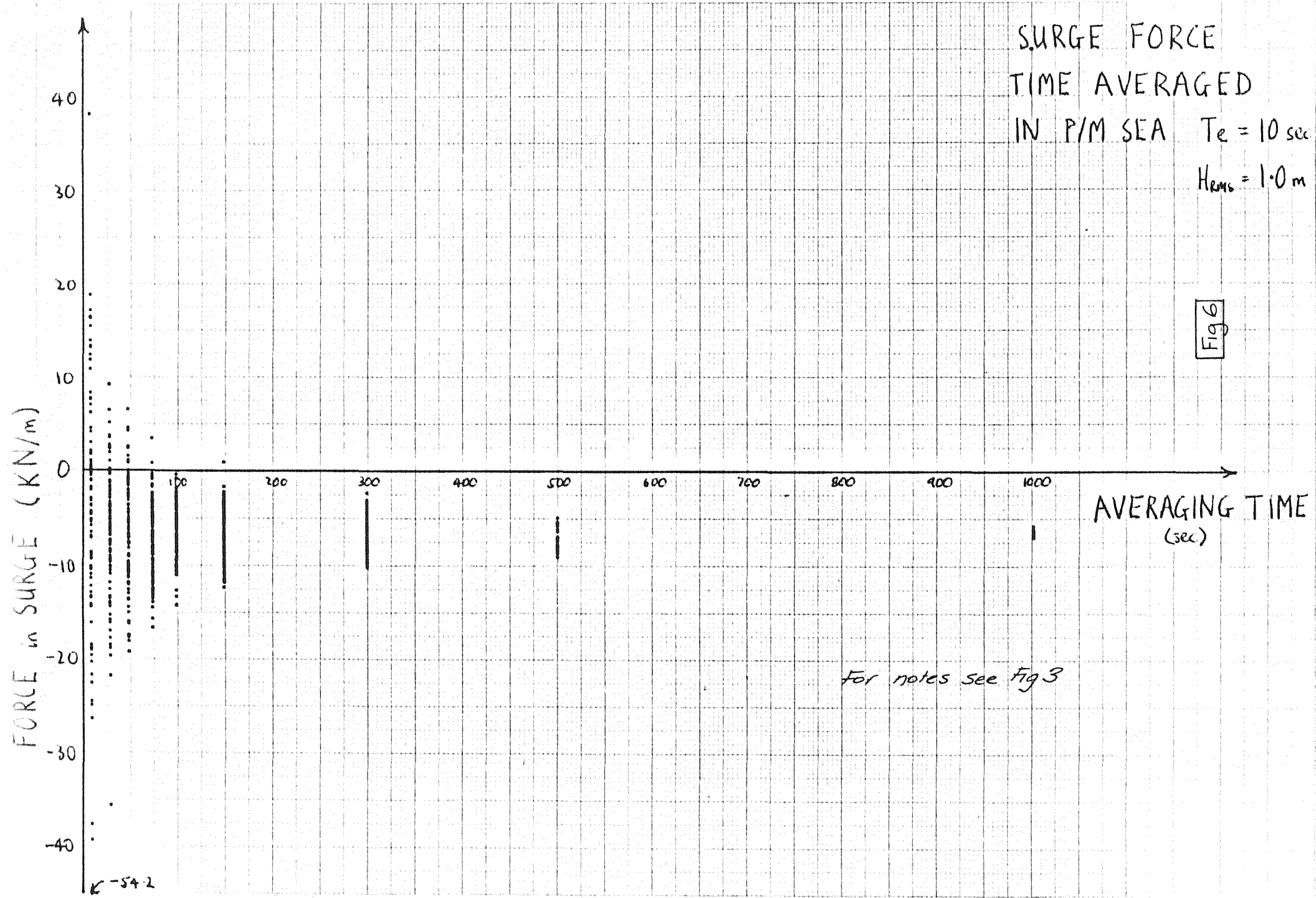
For notes see Fig 3

①-46.5



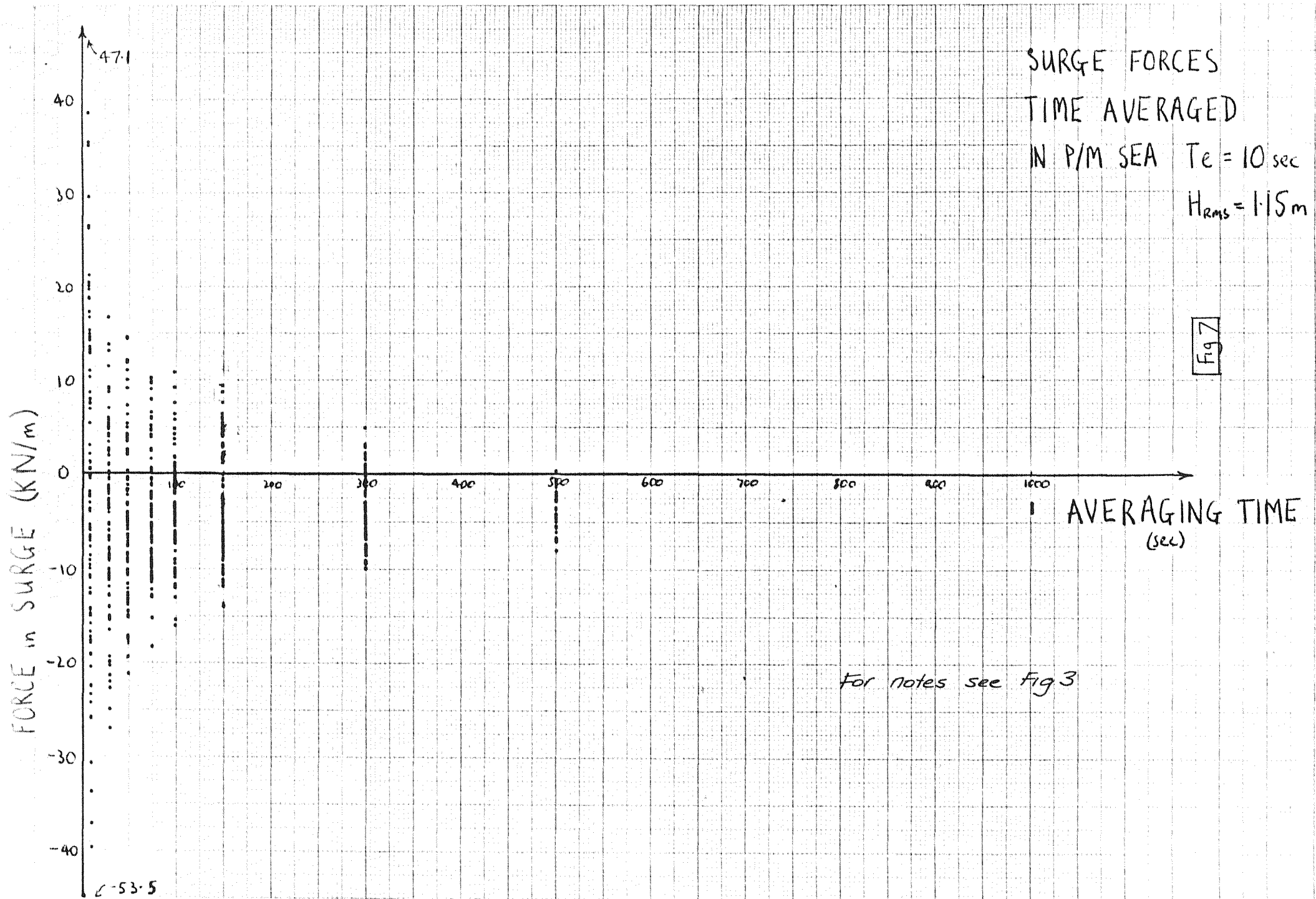
SURGE FORCE  
TIME AVERAGED  
IN P/M SEA  $T_e = 10 \text{ sec}$   
 $H_{rms} = 1.0 \text{ m}$

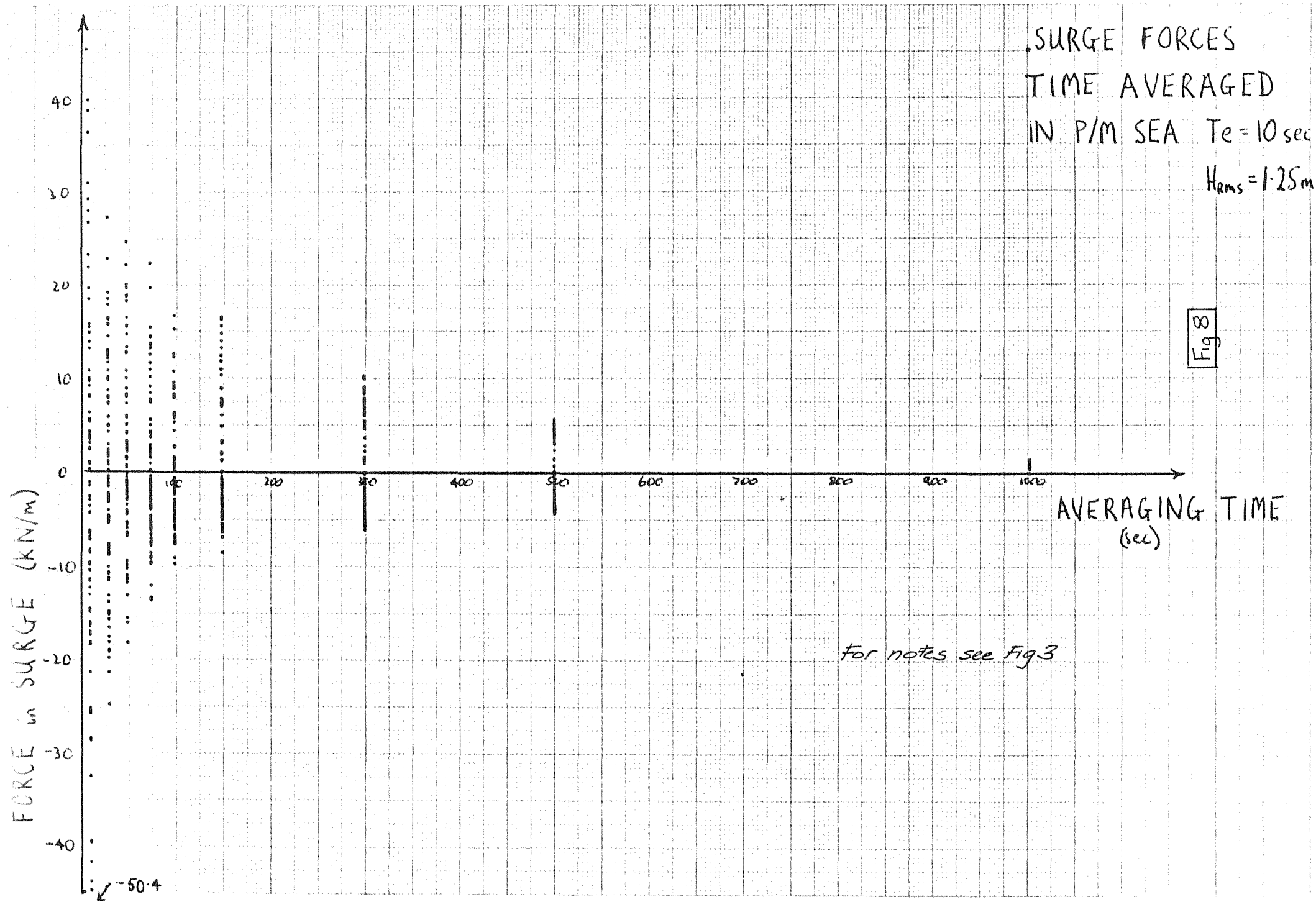
Fig 6



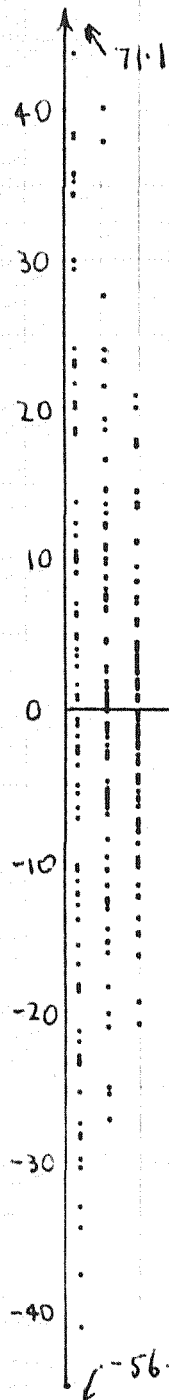
For notes see Fig 3

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FORCE IN SURGE (KN/M)



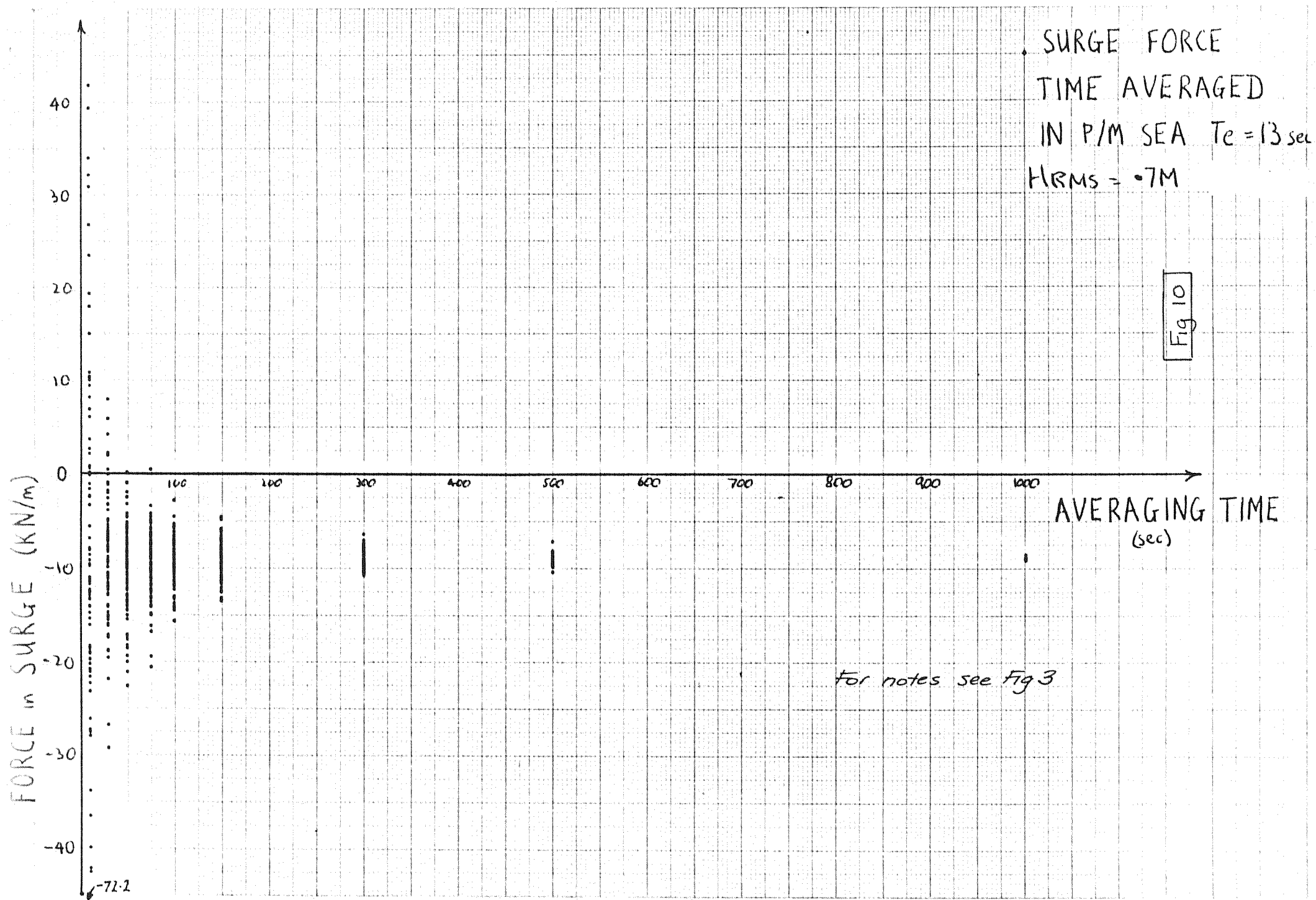
SURGE FORCES  
TIME AVERAGED  
IN P/M SEA  $T_e = 10 \text{ sec}$   
 $H_{rms} = 1.66 \text{ m}$

[endstop encountered once  
during test]

Fig 9

AVERAGING TIME  
(sec)

For notes see Fig 3



FORCE IN SURGE (KN/m)

40  
30  
20  
10  
0  
-10  
-20  
-30  
-40

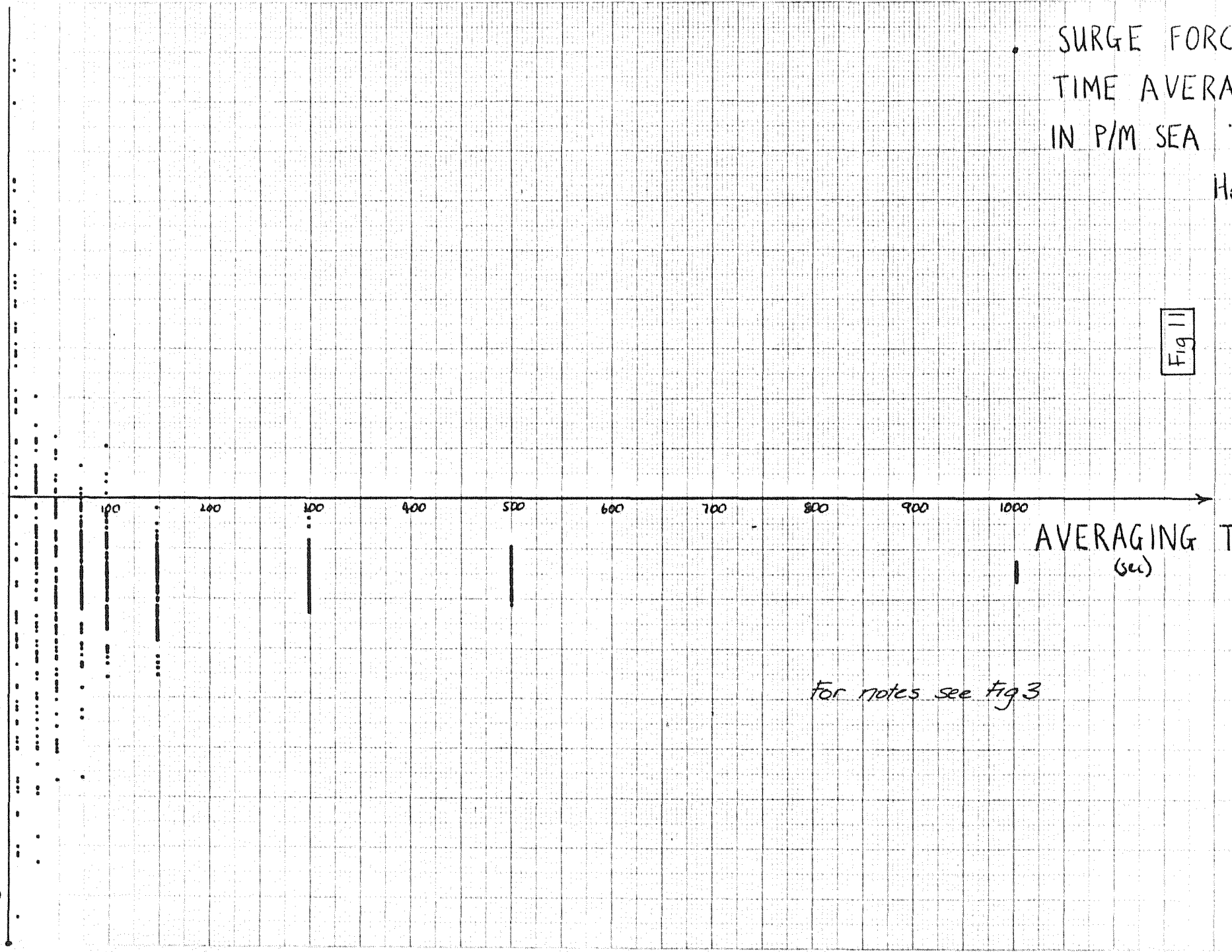
SURGE FORCE  
TIME AVERAGED  
IN P/M SEA  $T_e = 13 \text{ sec}$   
 $H_{rms} = 1.16 \text{ m}$

Fig 11

AVERAGING TIME  
(sec)

100 200 300 400 500 600 700 800 900 1000

For notes see fig 3



SURGE FORCE  
TIME AVERAGED  
IN P/M SEA  $T_e = 13$  sec  
 $H_{rms} = 1.4$  m

FORCE IN SURGE (KN/m)

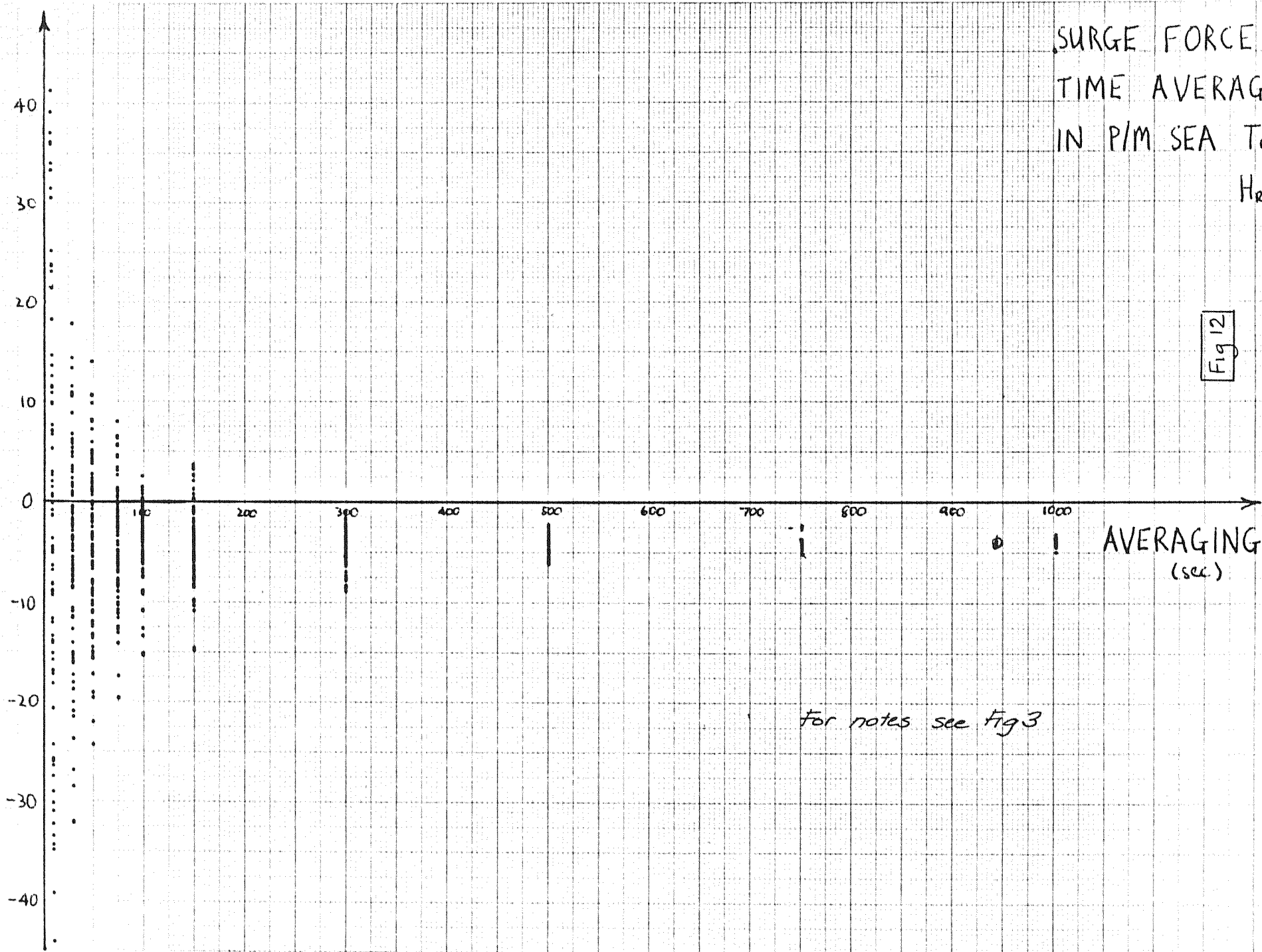
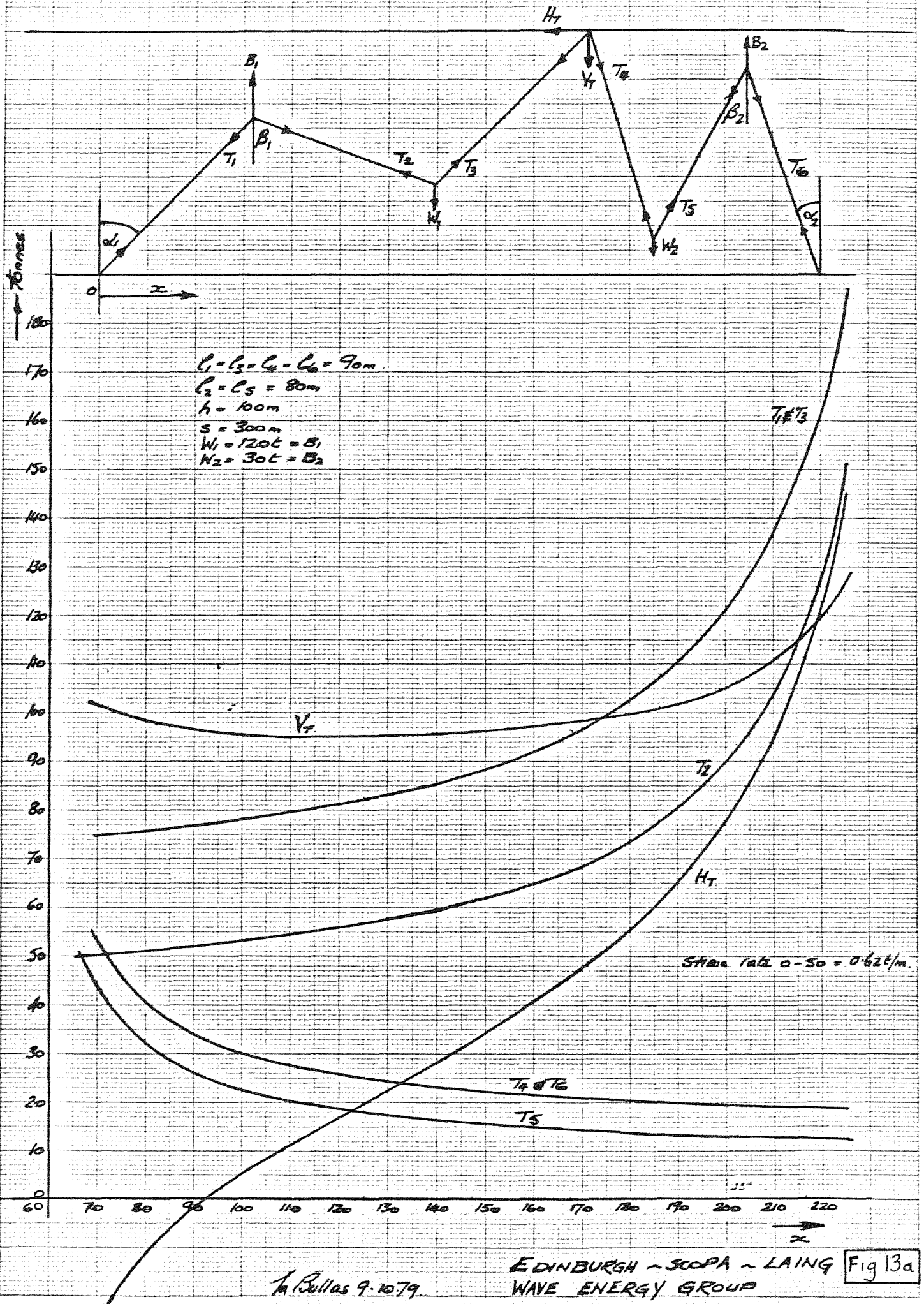


Fig 12

AVERAGING TIME  
(sec)

For notes see Fig 3

MOORING CONFIGURATION  
ROPE TENSIONS & MOORING FORCES AGAINST DISPLACEMENT.





LATERAL FORCE ( $H_L$ ) AGAINST DISPLACEMENT FOR GIVEN OFFSETS

$l_1 = l_3 = l_4 = l_6 = 90m$   
 $l_2 = l_5 = 80m$   
 $h = 100m$   
 $S = 300m$   
 $W_1 = B_1 = 120t.$   
 $W_2 = B_2 = 30t.$

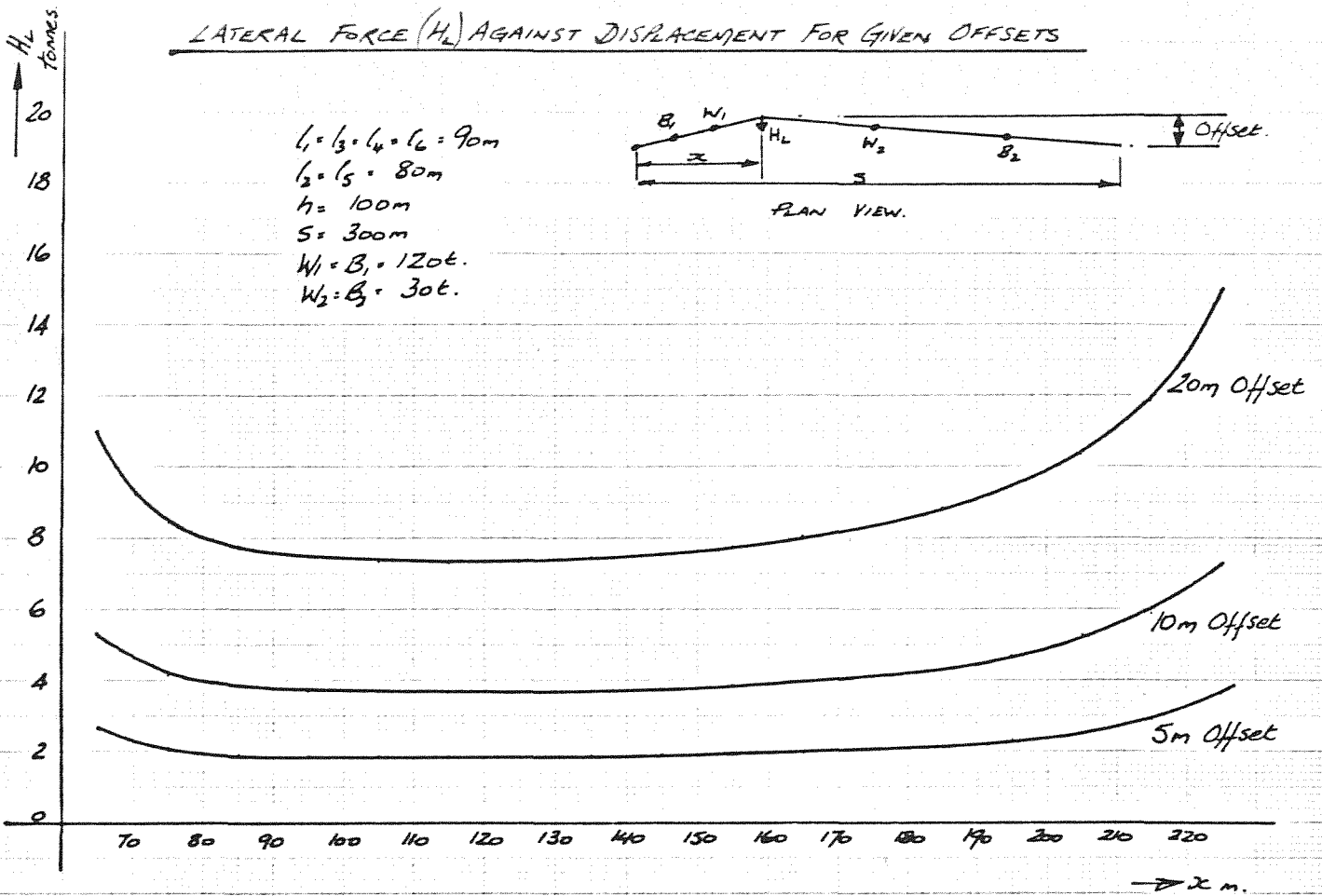
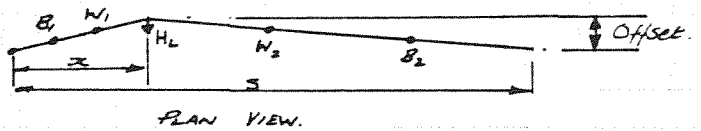


Fig 13b

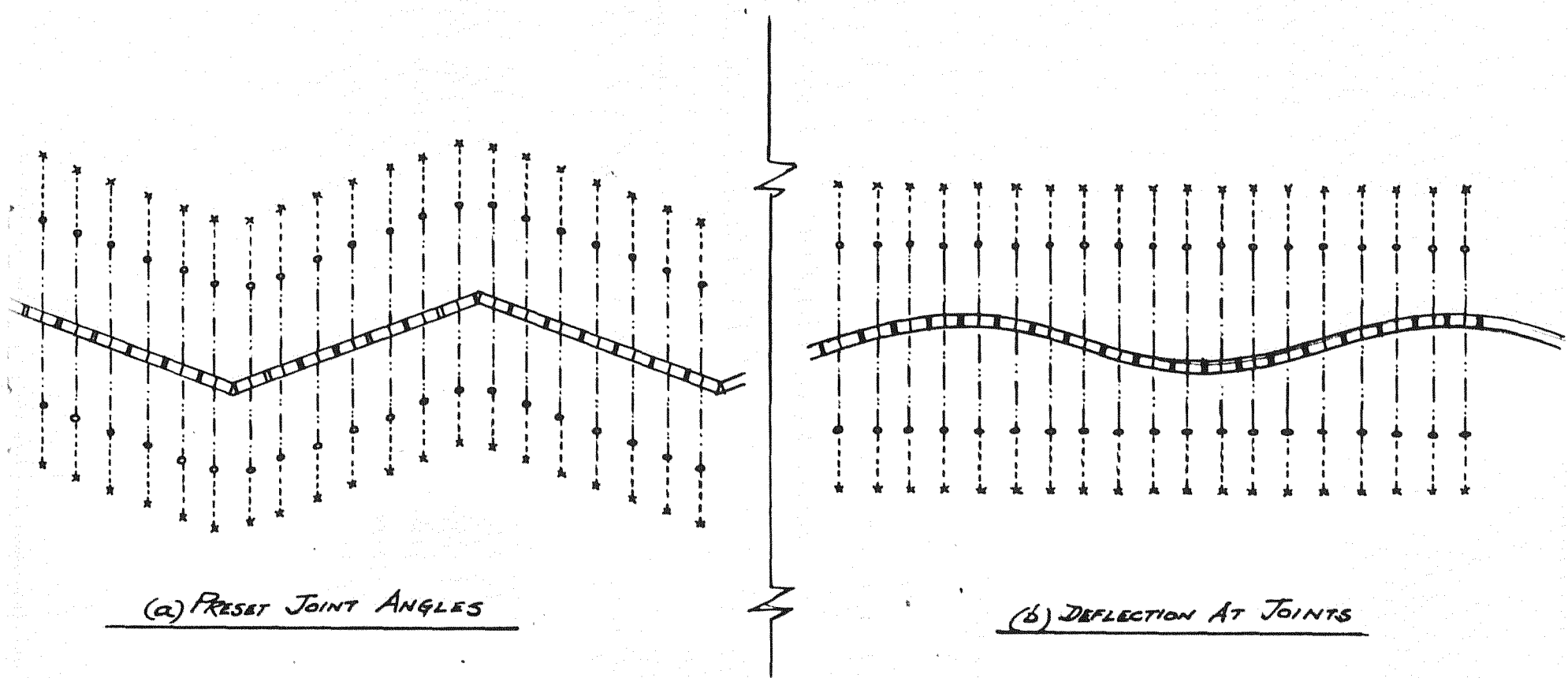
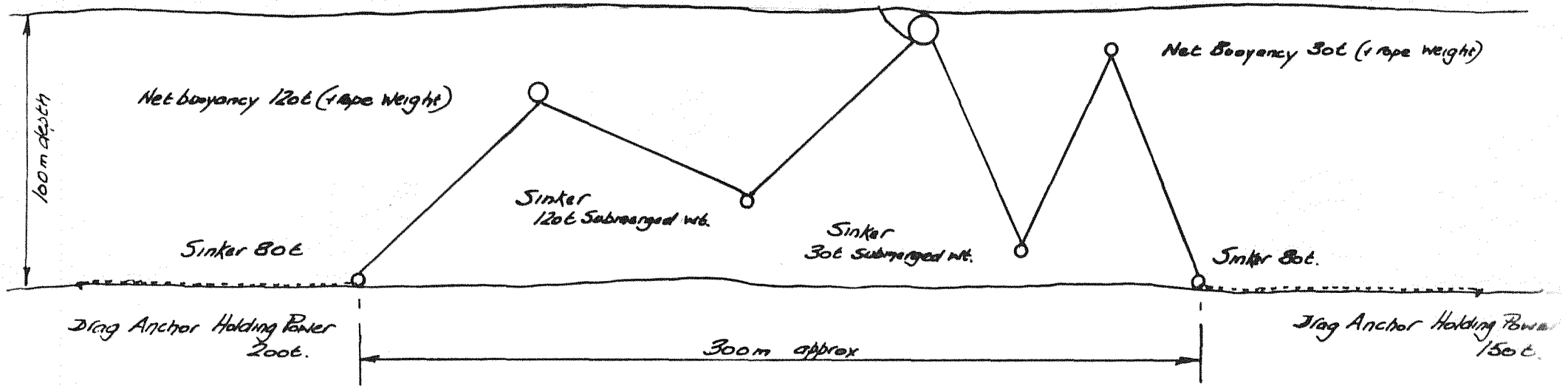


Fig 14

MOORING - GENERAL ARRANGEMENT - PLAN

EDINBURGH - SCOPA - LAING  
SKETCH 1 20.11.79



MOORING GENERAL ARRANGEMENT ~ ELEVATION

Fig 15

EDINBURGH - SCOPA - LANG.

SKETCH 2

20.11.79 (Rev 3)

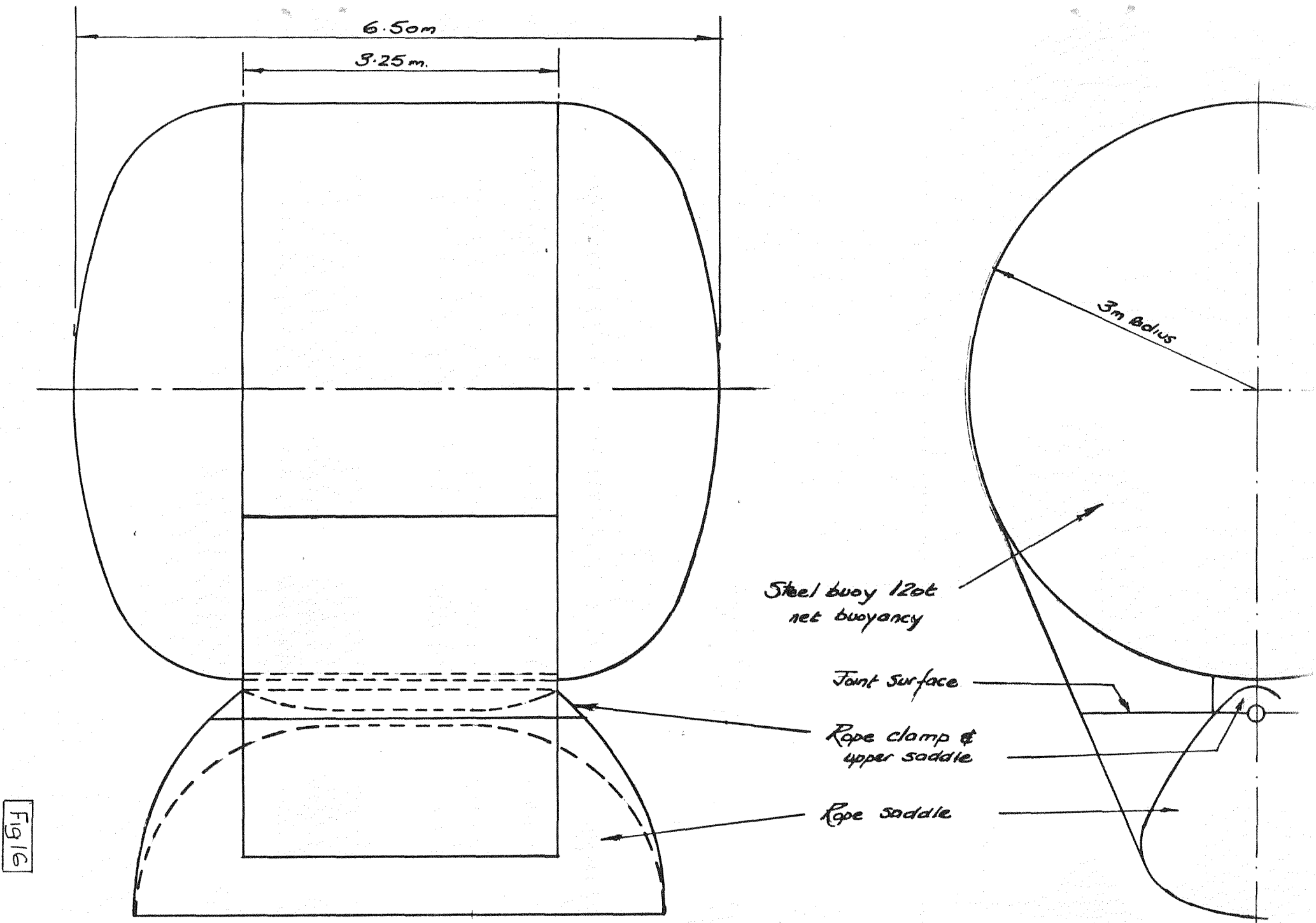


FIG 16

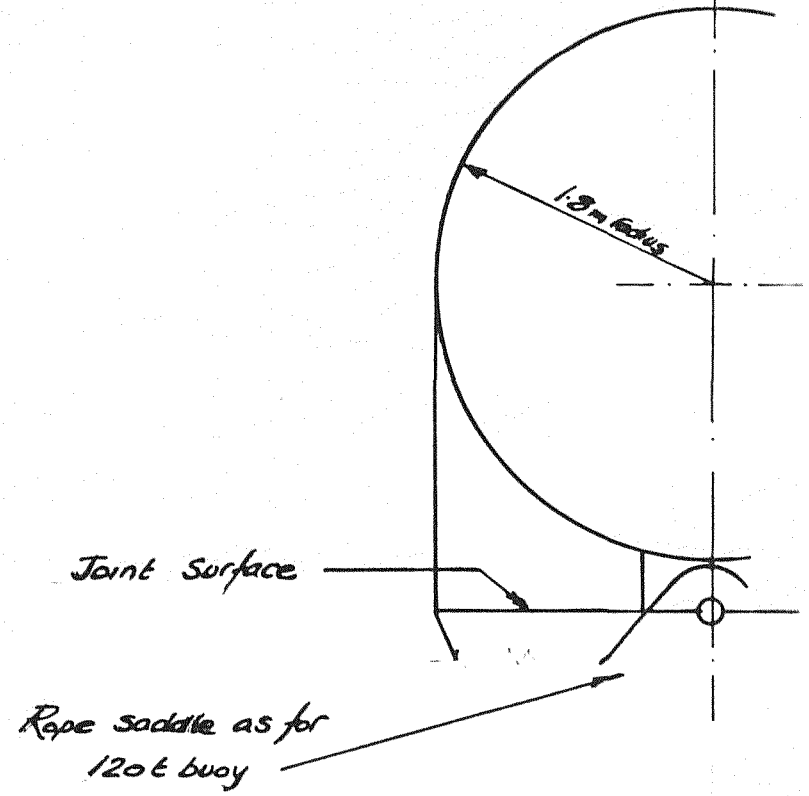
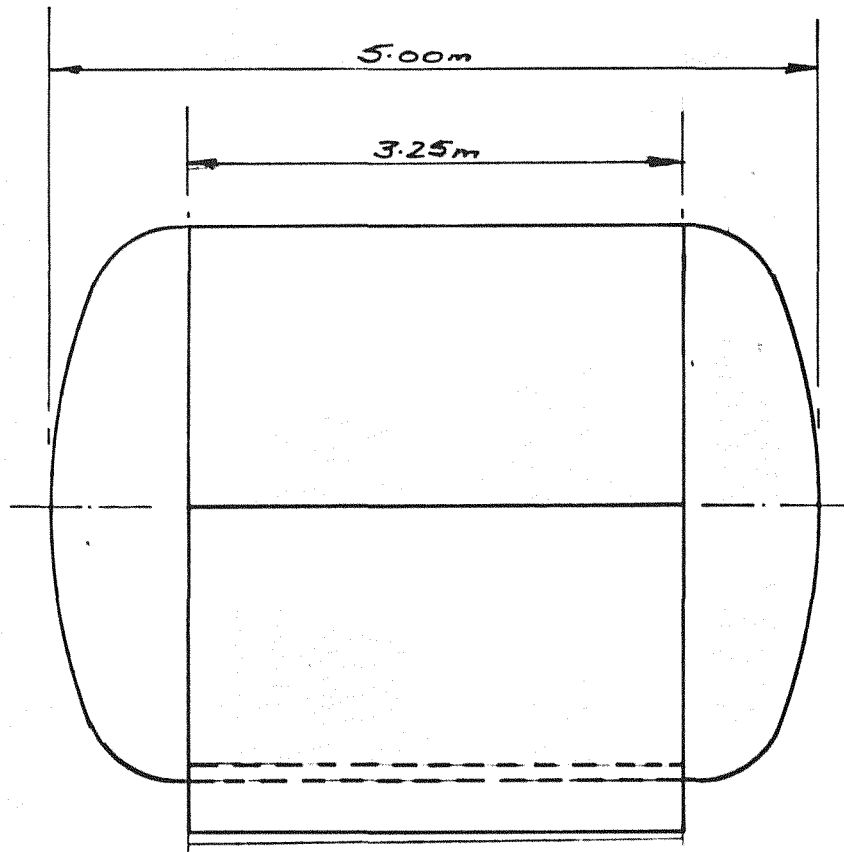


Fig 17

MOORING 30t BUOY

EDINBURGH - SCOPA - LAING

SKETCH A.

30. 11. 76

MOORING ~ 120t & 80t SINKERS

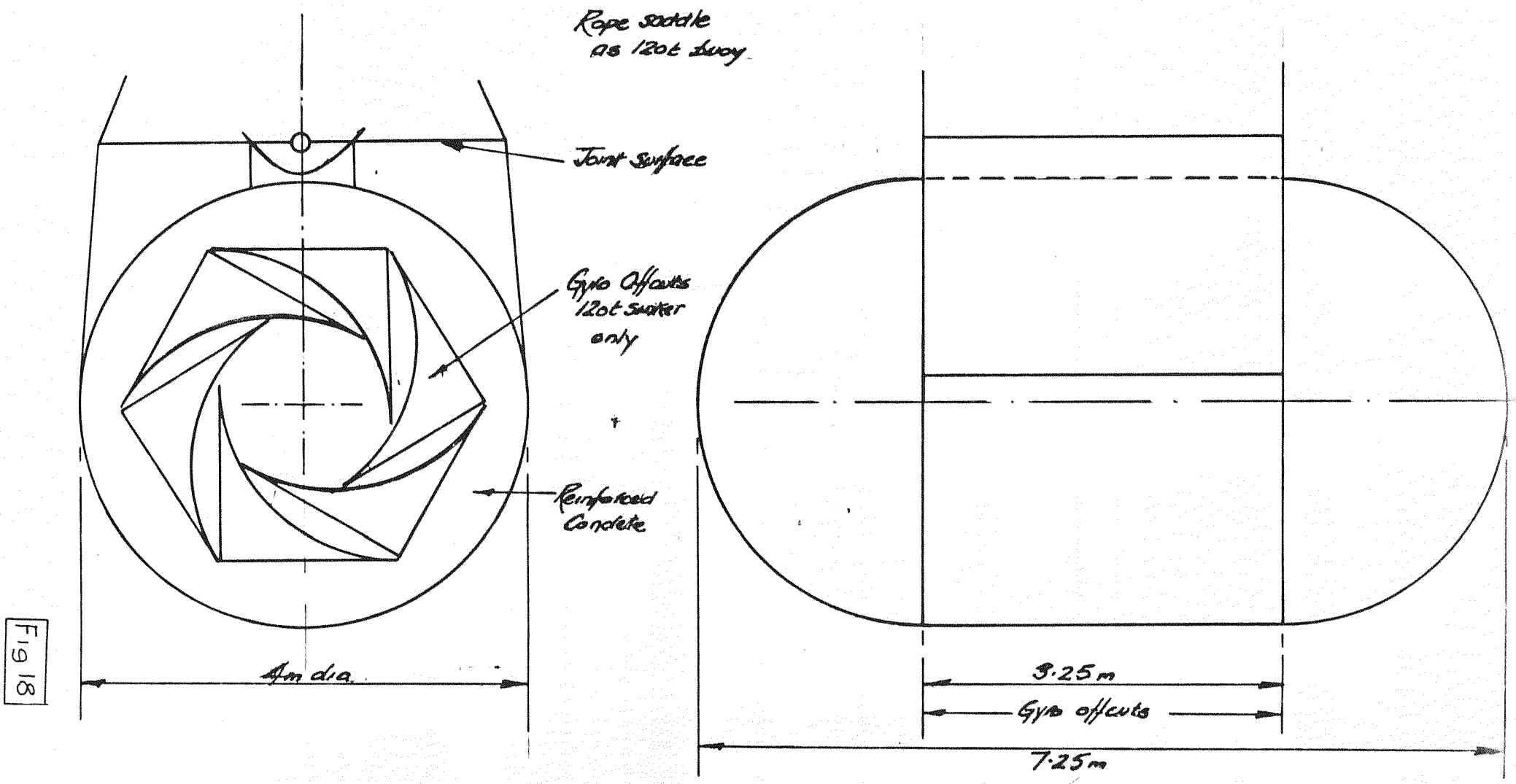


Fig 18