A PRELIMINARY DESIGN STUDY
OF THE TENSION LEG PLATFORM

by

HIROSHI IWASAKI

B.Eng., The University of Tokyo, Japan
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Signature of Author________________________________
Department of Ocean Engineering
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Certified by________________________________
Michael Stefanos Triantafyllou
Thesis Supervisor

Accepted by________________________________
Chairman, Departmental Committee
on Graduate Studies

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ABSTRACT

This thesis focuses on the preliminary design of the TLP which is considered an optimal production platform for water depths between 2,000ft and 3,000ft. First, published data are analyzed and useful information for an actual preliminary design is presented. Then, computer programs are developed for two alternative simple models for preliminary design. Finally, based on the analysis of the computer calculation, a preliminary design procedure is proposed.

Thesis Supervisor: Michael Stefanos Triantafyllou

Title: Assistant Professor of Ocean Engineering

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Also, I would like to express my appreciation for financial support of Mitsui Engineering and Shipbuilding Co.,Ltd., during my graduate study at M.I.T.

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LIST OF SYMBOLS

a; Wave amplitude

A; Area of object

A_T; Cross sectional area of the mooring line

b; $=d_0-H$

B; Deck width

BM; The distance from the center of buoyancy to the metacenter

BM'; Equivalent BM

c; Coefficient used for production equipment weight estimate

c'; Coefficient used for jacket weight estimate

C_1, C_2; Height coefficients

C_D; Drag coefficient

C_s; Shape coefficient

d; Draft

D; Deck elevation

d_0; Towing draft

E; Young's modulus of elasticity

f; Freeboard

F_c; Current force

f_e; Estimated freeboard

$F(t)$; External surge force

F_v; Vertical dynamic force

F_w; Wind force

g; Acceleration of gravity

GM; The distance from the center of gravity to the metacenter

GM'; Equivalent GM

h; Submerged column height

H; Height of enlarged section

h_d; Deck height

H_w; Max wave height

i_θ; Added pitching moment of inertia

I_θ;Pitching moment of inertia

i_ψ; Added yawing moment of inertia

I_ψ; Yawing moment of inertia

k; Wave number

K; Spring constant

KB; The distance from keel to the center of buoyancy

KB'; Equivalent KB

KG; The distance from keel to the center of gravity

KG'; Equivalent KG

l; Shorter column center spacing

L; Deck length

L_b; Lower hull width

L_h; Lower hull height

L_{wd}; water depth

M_1; Quantity for model 1

M_2; Quantity for model 2

M; Mass

M_c; Pitching moment due to current

M_h; Heeling moment

M_p; Pitching moment due to wave

M_r; Righting moment

M_w; Pitching moment due to wind

$1/n$; An exponent used for wind profile

P_b; Lower hull width

P_c; Production capacity

P_h; Lower hull height

r; Radius of column

R; Radius of enlarged section

R; Damping coefficient

S_a; Surface area of columns

t; Time

T; Wave period

T_0; Pretension

T_h; Horizontal component of the mooring tension

T_v; Vertical component of the mooring tension

v_c; Current velocity

V_H; The wind velocity at a reference height H, usually 10m above a reference water depth

V_j; Jacket volume

v_w; Wind velocity

V_y; The wind velocity at height y

W_1; Deck weight + deck load

W_a; Auxiliary equipment weight

W_d; Deck weight

W_e; Equipment weight

W_{js}; Jacket steel weight

W_{jt}; Jacket total weight

W_p; Weight of production and/or drilling system

W_r; Weight of riser & mooring system

W_{ss}; Steel structure weight

x; Horizontal excursion of the TLP

x_{all}; Allowable horizontal excursion.

x_{max}; Maximum horizontal excursion

x_{stat}; Static excursion

Δ; Displacement

Δ_l; Light weight

ΔF_v; Maximum vertical dynamic force variation

∇; Displacement volume

θ; Angle between tension line and vertical line

ρ; Density of sea water

ϕ; Helling angle

ω; Wave frequency

ω_h; Natural frequency of heave

ω_{pr}; Natural frequency of pitch and roll

ω_{ss}; Natural frequency of surge and sway

ω_y; Natural frequency of yaw

LIST OF EQUATIONS

$$W_r=0.44L_{wd}\,\Delta/1{,}000 \qquad \text{(Eq.6-1)}$$

$$W_p=cP_c \qquad \text{(Eq.6-2)}$$

$$W_a=93.3\,\Delta/1{,}000 \qquad \text{(Eq.6-3)}$$

$$W_e=(93.3+0.44L_{wd})\,\Delta/1{,}000\ +cP_c \qquad \text{(Eq.6-4)}$$

$$W_{ss}=0.37\,\Delta \qquad \text{(Eq.6-5)}$$

$$W_d=(0.74\ \pm\ 0.16)LB \qquad \text{(Eq.6-6)}$$

$$f_e=H_w/2+4+0.0113L_{wd} \qquad \text{(Eq.6-7)}$$

$$f_e=H_w/2+1.52 \qquad \text{(Eq.6-8)}$$

$$f_e \leq f \leq f_e+5.5 \qquad \text{(Eq.6-9)}$$

$$W_{jt}=0.23V_j \qquad \text{(Eq.6-10)}$$

$$\Delta_1=cP_c+(0.4633+0.44L_{wd}/3{,}000)\,\Delta \qquad \text{(Eq.6-11)}$$

$$\Delta_1=0.23V_j+0.74LB+cP_c+(93.3+0.44L_{wd})\,\Delta/3{,}000 \qquad \text{(Eq.6-12)}$$

$$F_v=(\,\nabla e^{-\frac{\omega^2}{g}\frac{d}{2}}\ +16\pi R^3/3*e^{-\frac{\omega^2}{g}d})\frac{\omega^2}{g}\ a*\cos\omega t-4\pi r^2\rho a*\cos\omega t \qquad \text{(Eq.7-1)}$$

$$\sqrt{6.5H_w}<T<\sqrt{15H_w} \qquad \text{(Eq.7-2)}$$

$$\omega_0{}^2=\frac{\omega_{max}^2+\omega_{min}^2}{2}=4.3527/H_w \qquad \text{(Eq.7-3)}$$

$$\Delta F_v=5.093r^2a \qquad \text{(Eq.7-4)}$$

$$r^2=\omega_0{}^2(\,\nabla e^{-\frac{\omega_0^2}{g}\frac{d}{2}}\ +16\pi R^3e^{-\frac{\omega_0^2}{g}d})/(4\pi g) \qquad \text{(Eq.7-5)}$$

$$F_v=(\nabla e^{-\frac{\omega^2}{g}\frac{d}{2}}+2LP_b{}^2e^{-\frac{\omega^2}{g}d})\frac{\omega^2\rho}{g}a\cos\omega t-S_a\rho a\cos\omega t$$

(Eq.7-6)

$$S_a=\frac{\omega_o^2}{g}(\nabla e^{-\frac{\omega_o^2}{g}\frac{d}{2}}+2LP_b{}^2e^{-\frac{\omega_o^2}{g}d}) \quad \text{(Eq.7-7)}$$

$$\Delta F_v=0.4053*S_a a \quad \text{(Eq.7-8)}$$

$$M\ddot{x}+R\dot{x}+Kx=F(t) \quad \text{(Eq.7-9)}$$

$$x_{max}=\frac{2\nabla\rho a\,\omega^2\cos\frac{kl}{2}e^{-k\frac{d}{2}}}{\left[\{T_0 g/(L_{wd}-d)-\omega^2(\Delta_1+\Delta)\}^2+4T_0\frac{g}{L_{wd}-d}(\Delta_1+\Delta)\zeta^2\omega^2\right]^{1/2}}$$

(Eq.7-10)

$$F_w=0.06255v_w{}^2(1.65Lh_d+4fr) \quad \text{(Eq.7-11)}$$

$$F_c=0.4188v_c{}^2[r(d-H)+RH] \quad \text{(Eq.7-12)}$$

$$F_c=0.4188v_c[r(d-P_h)+0.5P_hL] \quad \text{(Eq.7-13)}$$

$$x_{stat}=(L_{wd}-d)\sin \quad \text{(Eq.7-14A)}$$

$$T_h=T_v\tan\theta \quad \text{(Eq.7-14B)}$$

$$T_v=T_0+4\pi r^2(1-\cos\theta)(L_{wd}-d)\rho \quad \text{(Eq.7-14C)}$$

$$M_r=\Delta*BM\sin\phi\frac{1+\sec^2\phi}{2}-\Delta(KG-KB)\sin\phi \quad \text{(Eq.7-15)}$$

$$M_h=[1.65(h_d\cos\phi+B\sin\phi)L(d+f+h_d/2-KG)\cos\phi$$
$$+4r\{f(d+f/2-KG)\cos^2\phi+\frac{2}{8}\sin^2\phi\}]*0.06255v^2 \quad \text{(Eq.7-16)}$$

$$M_p=\Delta_1x_{max}\frac{\omega^2BG}{g}+T_0x_{max}KG/(L_{wd}-d) \quad \text{(Eq.7-17)}$$

$$M_w=0.06255v_w{}^2\{1.65h_dL(f+d+h_d/2-KG)+4rf(d+f/2-KG)\}$$

(Eq.7-18)

$$M_c=0.41878\left[r(d-H)(KG-H-d/2)+RH(KG-H/2)\right]v_c^2 \quad \text{(Eq.7-19A)}$$

$$M_c=0.41878\left[r(d-P_h)(KG-\frac{P_h-d}{2})+0.5P_hL(KG-P_h/2)\right]v_c^2 \quad \text{(Eq.7-19B)}$$

$$\omega_{ss}=\sqrt{\frac{\Delta-\Delta_1}{\Delta+\Delta_1}\,\frac{g}{L_{wd}-d}} \quad \text{(Eq.7-20)}$$

$$\omega_h=\sqrt{\frac{EA_Tg/(L_{wd}-d)+4\pi r^2\rho g}{\Delta_1+16\pi R^3\rho/3}} \quad \text{(Eq.7-21)}$$

$$\omega_h=\sqrt{\frac{EA_Tg/(L_{wd}-d)+S_a\rho g}{\Delta_1+2P_b{}^2L\rho}} \quad \text{(Eq.7-22)}$$

$$\omega_{pr}\geqq\sqrt{\frac{g\{EA_T/(L_{wd}-d)+4\pi r^2\rho}{\Delta+\Delta_1}} \quad \text{(Eq.7-23)}$$

$$\omega_{pr}\geqq\sqrt{\frac{g\{EA_T/(L_{wd}-d)+S_a\rho\}}{\Delta+\Delta_1}} \quad \text{(Eq.7-24)}$$

$$\omega_y\leqq\sqrt{\frac{12T_0}{(L_{wd}-d)(\Delta+\Delta_1)}} \quad \text{(Eq.7-25)}$$

$$M_1=\frac{(v_c{}^2+v_w{}^2/1{,}000)L_{wd}}{H_w} \quad \text{(Eq.8-1)}$$

$$M_2=\frac{(2v_c{}^2+v_w{}^2/1{,}000)L_{wd}}{H_w} \quad \text{(Eq.8-2)}$$

$$W_r=L_{wd}\Delta/1{,}000+9L_{wd}(1+cP_c/1{,}000) \quad \text{(Eq.8-3)}$$

$$x_{dyn} \cong \frac{0.725\,\Delta}{(\Delta_1 + \Delta)/H_w - (\Delta - \Delta_1) * 2.25/(L_{wd} - 30)} \qquad \text{(Eq.8-4)}$$

$$x_{stat} \cong \frac{(77 + 1.15\,\Delta/1{,}000)(v_c^{\,2} + v_w^{\,2}/1{,}000)}{(\Delta - \Delta_1)}(L_{wd} - 30)$$

(Eq.8-5)

OVERVIEW

Offshore platforms have been installed in increasingly deeper water. However, some studies indicate that the Cognac platform installed in 1025 ft of water is probably the practical upper limit of the fixed offshore platform due to economic and engineering problems. There is a consensus today that the fixed platform is optimal up to 1,000 ft, the guyed tower between 1,000 and 2,000 ft and the TLP between 2,000 and 3,000 ft.

In Chapter 3 a description of the TLP is given. First, the total TLP system is overviewed followed by a relatively detailed description of the Tension Leg Platform itself. Next the installation is described Finally, the advantages of the TLP are discussed.

In Chapter 4 some previous work is reviewed. First, research and experimental results are reviewed to indicate the available engineering level of the TLP. Then 5 proposed designs are reviewed.

Chapter 5 presents the difficulties associated with the actual design of the TLP and explains the purpose and goals of this thesis.

In Chapter 6 published data are analyzed to get useful information for an actual preliminary design.

This data include proposed TLP designs, actual designs of semisubmersibles, and designs of fixed platforms. First, equations to estimate equipment weight are presented. These equations enable us to estimate the equipment weight from production capacity, water depth and displacement and are very important at the preliminary design stage.

Next, relations among parameters are derived. They include: (1) Steel structure weight vs. displacement, (2) Deck size vs. deck load, (3) Deck size vs. deck weight, and (4) Displacement vs. draft.

Section 6.3, Estimate of Freeboard, presents a method to predict maximum and minimum freeboard required from environmental data.

Section 6.4 presents a method to estimate the jacket weight from the jacket volume. This method is used in the computer program to calculate the light weight. Finally, Section 6.5 presents a method to calculate the light weight without using the computer. This is very useful to find an initial value of the displacement to start the computer iterations.

Chapter 7 introduces two simple models for analysis and reviews the various design parameters and requirements. Section 7.2 presents a method to minimize the dynamic tension variation for a given range of wave period. Section 7.3 discusses the stability. Two conditions are considered. The one is the towing-out condition, and the other is the operating condition with one leg totally lost. Section 7.4 presents the evaluation of the dynamic excursion, Section 7.5 discusses the wind and current forces. Section 7.6 presents a method to calculate the static excursion. In Section 7.7 the dynamic stability is discussed. Section 7.8 describes the pitching moment effect on mooring tension variation. Finally, in Section 7.9 natural frequencies are discussed. The computer program is developed based on these design considerations.

Chapter 8 presents the results of the computer calculation and the analysis of these results. First, displacement and deck size are discussed. A method to check the relative importance of the dynamic effects against static effects is introduced. This method is used to explain the correlations between displacement and deck size and horizontal excursion.

In Section 8.2, the relation between the ratio of light weight to displacement and positive tension restriction is presented. The influence of draft is discussed in Section 8.3. In Section 8.4, natural frequencies are discussed. It is shown that there is no problem with surge and yaw but there can be a problem with heave and pitch for deep water. In Section 8.5, it is shown that dynamic stability is not critical as long as the TLP has a reasonable static stability.

In Section 8.6, the influence of water depth is discussed. Another method to estimate riser and mooring system weight is presented. The minimum required displacement, together with the natural heave period, are calculated for various water depths based on 2 different assumptions of weight estimate. It has been shown that the practical limit of the TLP is at most 1,000m. Due to limited data available, it can not be determined which assumption is closer to the truth. Some other results are discussed and finally an approximate yet very simple method to estimate the horizontal excursion is presented.

Based on the analysis presented in Chapter 8, a design procedure of the TLP is developed in Chapter 9. Finally, in Chapter 10, this investigation is summarized and recommendations for future research are outlined.

CHAPTER I

INTRODUCTION

The last decade has seen tremendous advances in the development of deep water technology in the field of offshore drilling. A number of exploratory wells have been drilled in water depths in excess of 3,000ft. However, development and production technique have not advanced to the same extent.

As the search for hydrocarbons continues, offshore platforms are being installed in deeper and deeper water. These platforms have primarily been of the fixed type. The Cognac platform installed offshore Louisiana in 1,025 feet of water is the deepest existing rigid platform. However it is expected that the application of the fixed offshore platform for deeper water will encounter serious technical and economical problems.

The cost of fixed structures increases beyond a certain point, exponentially with water depth and the severity of the environmental forces. Only large and prolific reservoirs can be considered for development using a fixed platform in deep water. In addition, as the water depths increases, the first natural periods of vibration of these platform increase up to $4 \sim 5$ seconds. The wave forces have significant energy at this level and dynamic amplification becomes significant. (see Fig.1) Such structures may thus be subjected to a very large high number of cycles at a significant stress level so that structural fatigue becomes a significant problem.

Another limitation of the fixed type platform is the fabrication capability. Single-section jackets are necessarily constructed in a horizontal mode in order for cranes to roll and lift the jacket sections into position. The highest lift used in fixed platform fabrication to date was a 316 ft crane for the Cognac project. According to the study (14) the base width in the short dimension is generally one-third of the water depth, making 1,000 ft water depths the limits of conventional technology, at least until cranes with a higher capacity are developed.

Consequently, development of relatively small reservoirs or those located in water depths beyond the economic limit of a fixed platform require alternate production concepts and the need for such production facilities will also arise soon.

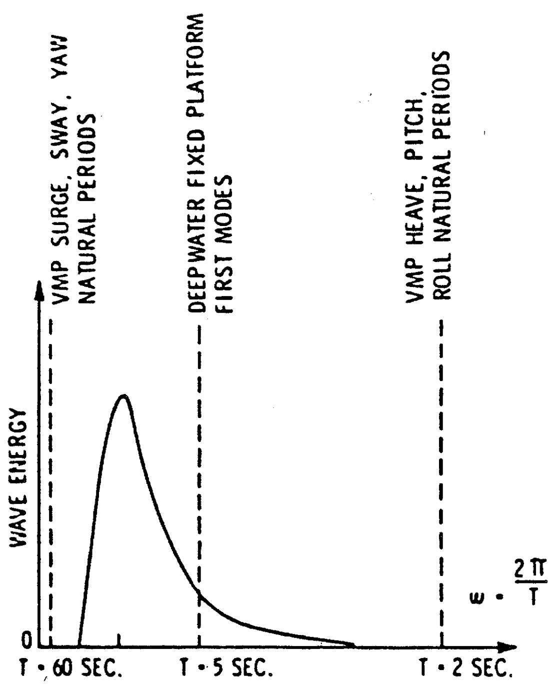


FIG. 1 - WAVE ENERGY SPECTRUM.

CHAPTER II

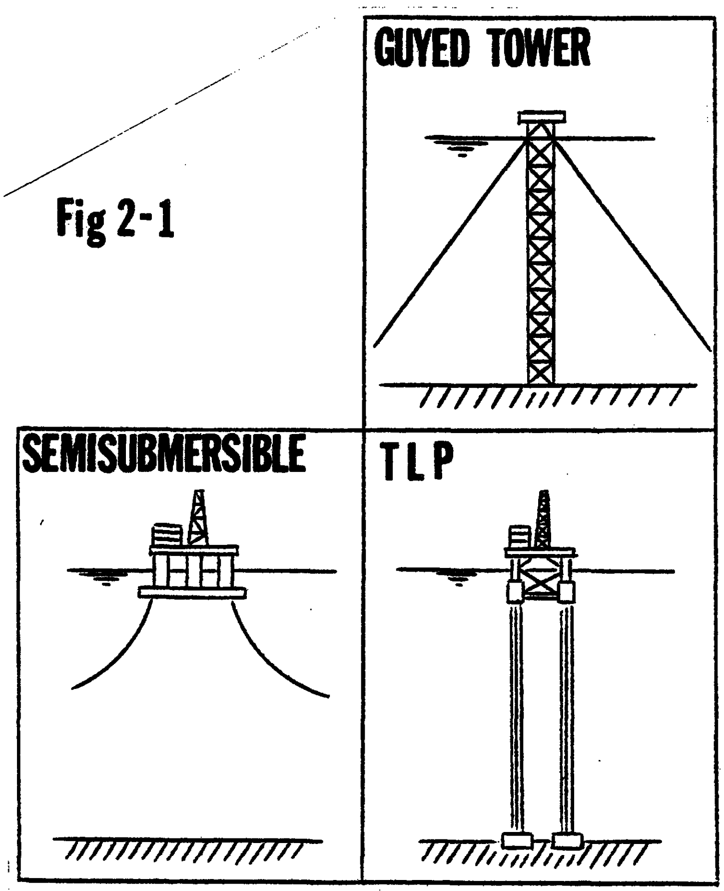
COMPARISION OF ALTERNATIVES

2.1 Description of Each System

Due to the economic and engineering problems discussed in the previous section, considerable interest has developed in the use of compliant structures. Compliant structures, by definition, allow motion under wave loading and therefore experience reduced stresses. In other words, these structures avoid severe dynamic problem by making the structures more flexible and moving their natural frequencies to the lower frequency side of the wave spectrum.(see Fig.1) Though there are numerous variations of compliant structures, 3 distinct types TLP, guyed tower and semi-submersible as shown in Fig.2-1 are regarded as the more practical concepts.

The guyed tower is a relatively slender symmetrical structure supported on the sea bed and held upright by spread-moored guy lines. A guyed tower, particularly for water depths greater than 1,000 ft, could be fabricated at a smaller cost than a conventional fixed platform because of uniform cross section and greatly reduced steel tonnage. The concept is quite attractive for light deck loads and a mild environment. The cross-section of a guyed tower is generally about one-tenth of the water depth. The study projected that guyed towers will not likely be installed in depths greater than 2,000 ft, because the size at such depths becomes difficult to fabricate. Although no full-scale guyed tower exists today, Exxon has installed a 1/5 scale model in 300-ft water depth.

Fig 2-1



A second alternative which takes advantage of readily available semi-submersible drilling platforms and much tested equipment and techniques is the floating platform production system. The first such installation is Sedco-Hamilton's Floating Production Facility (FPF) on the Argyll Field in the North Sea. Prudent operation of such a conventionally moored facility requires that the wells be shut-in and risers be cleared of oil and retrieved well in advance of weather conditions which would impede such operations. The result is significant downtime and therefore a less cost-effective program. Also, when production is resumed, flow rates are often lower.

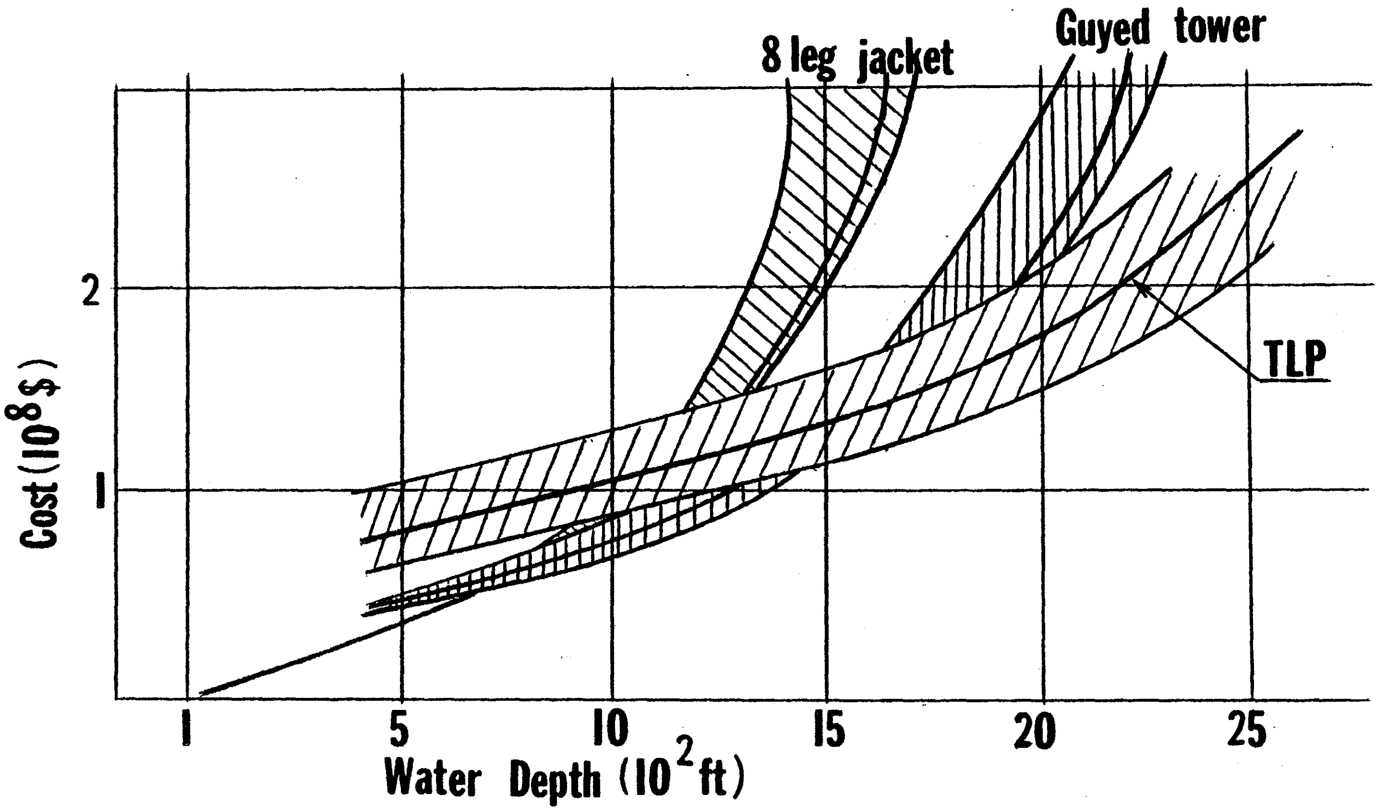
The tension-leg platform is basically a large semisubmersible drilling vessel with vertically anchored lines at high tension. In 1976, the 1/3-scale version of a TLP has been installed and tested off Southern California.(16,17,18) Unlike the fixed platform or guyed tower, the tension-leg platform requires a subsea wellhead system or template. Placement of the template, wellhead assembly and anchor blocks, and connection of the production riser and mooring systems (cables or structural riser pipes) make the tension-leg platform more expensive to install. However its relative insensitivity to water depth makes it very attractive for deep water. The basic description and variations of this type of structure are presented in Chapter 3.

2.2 Optimal Depth for Each Concept

Technical and economical comparisions of these alternatives show that each has a certain optimal water depth range.(see Fig.2-2) The semisubmersible can be deployed at virtually any water depth but the prospect of substantial down-time for weather and maintenance makes it a lesser choice. Fixed steel platform are suited for shallow water applications up to a depth range of 800 to 1,000 ft of water, at which point the guyed tower becomes the most economic application. The guyed tower is expected to be the least expensive unit between 1,000 and 1,800 ft of water. Beyond 2,000 ft the guyed tower becomes too massive for practical use and the tension-leg platform becomes the most economical application. For ultra-deep discoveries past the 3,000 ft depth, the natural period and heave of the tension-leg platform becomes too large. Platform designs for production in water depths beyond 3,000 ft have not been seriously evaluated, but the semisubmersible production facility linked to a future version of the existing subsea systems will be the likely choice.

Fig 2-2 (14)

Cost comparison for deepwater structures in the Gulf of Mexico



CHAPTER III

DESCRIPTION OF THE TLP

3.1 Subdivision of the System

The TLP production system consists of (see Fig.3-1)

(1) A semi-submersible type floating structure moored with vertical tension cables, or structural riser pipes to the seabed, which carries the usual processing and/or drilling equipment.

(2) A production riser system for flowing fluids between the seabed and the TLP and for servicing the wells and reservoir.

(3) A sub-sea wellhead system consists of a multi-well seafloor template and a comparatively simple safety block valve for each well. (Fig.3-2～3-4)

(4) An offshore tanker loading oil export system or pipeline to shore.

Produced crude oil is processed on the platform and transferred to shore through a subsea pipeline or a tanker loading/shuttle tanker system. A variation of the tanker loading system would include an undersea storage tank to make production capability less sensitive to tanker availability.

Fig 3-1

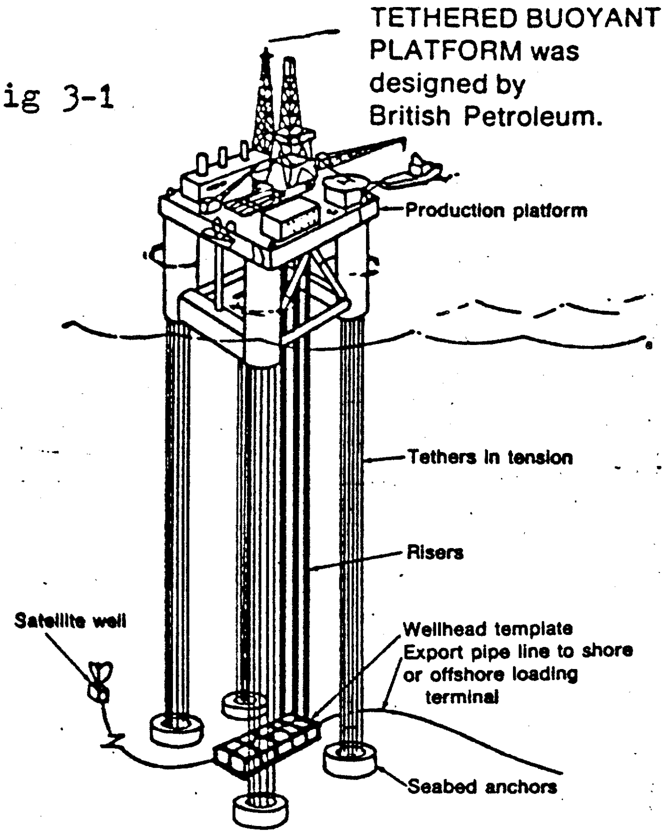

TETHERED BUOYANT PLATFORM was designed by British Petroleum.
Production platform
Tethers in tension
Risers
Satellite well
Wellhead template
Export pipe line to shore or offshore loading terminal
Seabed anchors

Fig 3-2

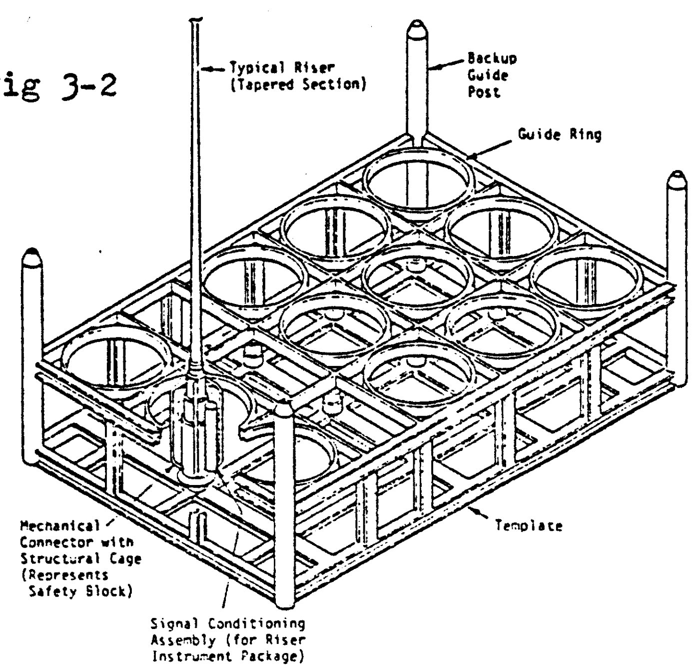

Typical Riser (Tapered Section)
Backup Guide Post
Guide Ring
Template
Mechanical Connector with Structural Cage (Represents Safety Block)
Signal Conditioning Assembly (for Riser Instrument Package)

Fig 3-3

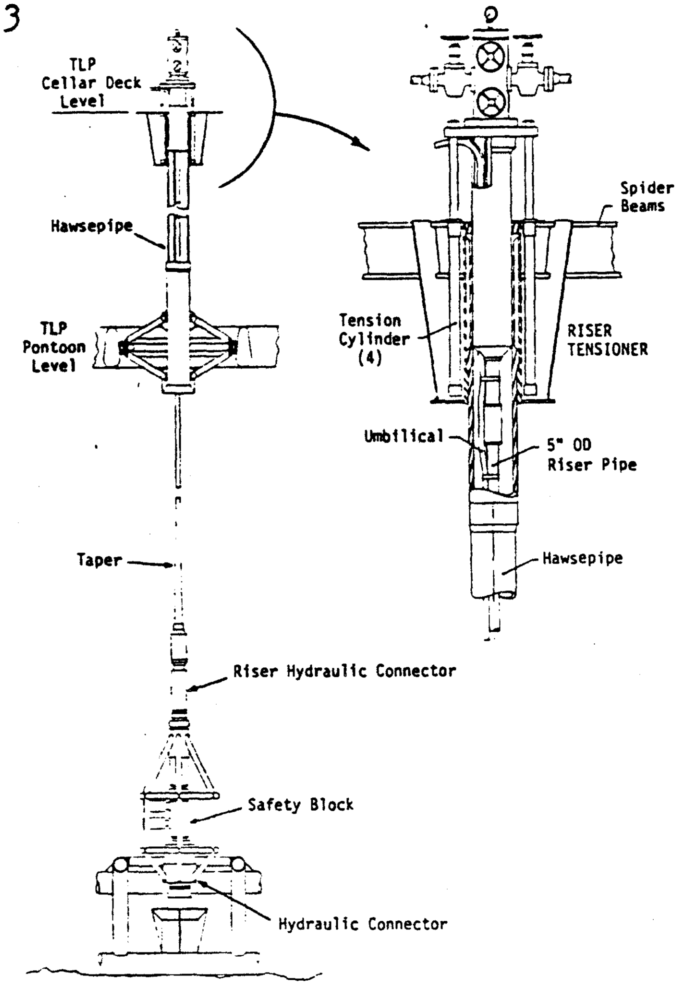
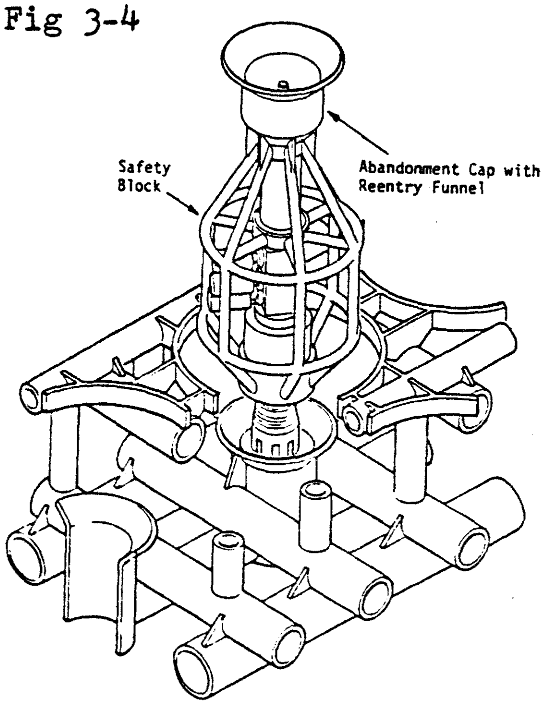

TLP
Cellar Deck
Level
Hawsepipe
TLP
Pontoon
Level
Taper
Riser Hydraulic Connector
Safety Block
Hydraulic Connector
Spider
Beams
Tension
Cylinder
(4)
RISER
TENSIONER
Umbilical
5" OD
Riser Pipe
Hawsepipe

Fig 3-4


Safety
Block
Abandonment Cap with
Reentry Funnel

3.2 Description of the TLP

The Tension Leg Platform is a floating structure similar to ordinary semi-submersibles. The principal characteristic of this type of platform is that its buoyancy exceeds the weight, and the supplementary downward force is supplied by tensioned vertical anchor cables or by risers.

The basic motion is similar to that of an inverted pendulum; it is very flexible horizontally and rigid vertically. The effect of this mooring system is to eliminate vertical motion while permitting limited horizontal excursion of the platform.

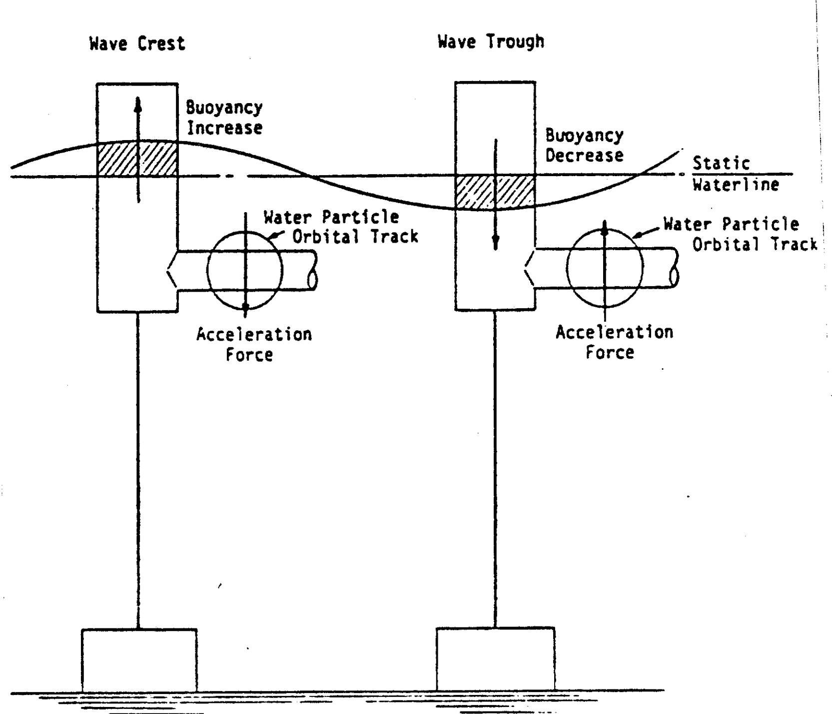
The mooring system provides the necessary resiliency to absorb horizontal forces developed by wind, wave or current, while providing the restoring force to keep the platform on station. As the platform is forced off station, the horizontal component of the mooring line tension counteracts the offsetting force. The draft of the platform increases with offset and this enhances the restoring force effect.

The maximum offset is limited by design to insure that riser stresses remain within acceptable limits. Platform buoyancy and anchor size are selected to insure that mooring lines remain in tension and the anchors can stand maximum expected mooring tension.

The vertical dynamic response of the platform due to wave motion is minimized by carefully designing the vertical and horizontal members, to take full advantage of the principle of wave force cancellation. As the wave crest passes along the platform (see Fig. 3-5), the buoyancy of the platform increases. At the same instant, downward forces are caused by the passing wave through the associated water particle accelerations. As the wave trough passes the platform, the vertical forces are reversed.

Thus the buoyancy and acceleration forces resulting from waves tend to cancel each other at all times. The optimum sizing and distribution of the buoyant members maximizes the wave force cancellation effect and minimizes the cyclic response of the platform. The result is a smaller (less costly) and more efficient mooring system.

Fig 3-5


Wave Crest
Wave Trough
Buoyancy
Increase
Buoyancy
Decrease
Static
Waterline
Water Particle
Orbital Track
Water Particle
Orbital Track
Acceleration
Force
Acceleration
Force

3.3 Installation

The wellhead template and the anchor templates will be fabricated and installed at least one year ahead of the platform to achieve an early start to production drilling. The platform installation will be assisted by a number of tugs and a temporary caterary mooring system to position the TLP over the anchors. The TLP is ballasted with seawater to float freely when finally installed. Some designs require temporary auxiliary bouyancy tanks at this stage. The mooring tethers are run simultaneously and each is stabbed-in and connected to the appropriate anchor post at about the same time. At this stage the tethers are slack, and the next step is to apply tension by a combination of deballasting and the hydraulic tension mechanism. Finally the remaining tethers are installed and the correct tensions applied.

3.4 Advantages of the TLP

The following is a summary of the major advantages of a TLP production system as compared to other alternatives.

(1) Unlike conventional fixed type offshore platforms, TLP costs are relatively insensitive to water depth increases.

(2) Since there is virtually no movement of the wellhead relative to the deck structure, constant tensioning devices or flexible piping can be eliminated. Thus essentially conventional fixed offshore platform drilling, completion, production and workover operations are conducted from the TLP deck.

(3) TLP can be retrieved and used again at other locations, an attractive feature allowing flexibility in developing a field. The much lower salvage costs plus the possible reuse value are facts that make the overall economics of the TLP very attractive.

(4) The construction procedure allows the bulk of the production facilities and platform equipment to be installed at the fabrication yard, effectively reducing the offshore construction costs and installation time.

CHAPTER IV

REVIEW OF PREVIOUS WORK

4.1 Review of Research

A number of companies and investigators have carried out research on tension-type platforms. Paulling et al developed a method of predicting the TLP motions and the forces in the mooring legs using a linearized hydrodynamic synthesis technique.(20,21)

The actual application of the TLP concept was carried out by Earl and Wright on a "screen barge" in 1966. Gravel for the transbay tube in San Francisco Bay was placed by this screen barge. Earl and Wright carried out feasibility and parametric studies, as well as an actual initial design of a TLP system.(13)

Another structure that has actually been fabricated, installed(1976) and tested is the 1/3-scale version of a TLP by Deep Oil Technology off Southern California.(16,17,18) The platform is triangular in shape, 130 ft on each side, and 66 ft in height from deck to lower horizontal pontoon. The test site was in 200 ft of water on the seaward side of Catalina Island. Many field tests which simulate actual installation and operation have been and still are being carried out.

4.2 Review of the Proposed Design

Most major oil companies are developing variations of the tension leg platform. Conoco intends to install a TLP in the North Sea over the Hutton field.(1,2,3) The proposed 8-column TLP with rectangular deck will be installed in 485 ft deep water by 1984. This project is considered to be a test of TLP system in an actual production mode, similar to the non-production TLP test off the coast of southern California.

B.P. designed a four-column TLP with square deck for 183 meter deep Magnus field in the North Sea.(4,5,6) B.P. also studied 2 types of tensioning system-spiral strand wire and tubular steel pipe. It was concluded that while a TLP system was technically feasible for Magnus, it was not the most economic solution.

The Aker group has proposed its own Aker TPP41 tethered production platform.(11) The hexagonal hull configuration was developed by Gulf Research and Development as the least sensitive to wind and wave direction. (19)

Amoco and Standard oil proposed a four-column TLP with square deck.(6,7,8,9,10) They also conducted fatigue analysis of the structural risers. Tecnomare also designed a four- column TLP with square deck for

600 m deep water.(12) Tables 4-1 through 4-5 summarize technical data of 5 proposed TLPs.

Table 4-1

Conoco	
References	(1,2,3)
Production Capacity	120,000(b/d)
Water Depth	147m
Site	Hutton field in the North Sea

Environmental Condition	
Max. wave height	30m period 17sec
Max. wind	44m/s(1min mean)
Max. current	1.13m/s
Max. tide	+2m

Dimensions	
Deck length	78m
Deck width	74m
Draft	30m
Freeboard	23.7m
Deck height	10.8m
# of column	8m
Column diameter	15m
# of mooring tethers	12 steel pipes

Load Summary(tons)	
Facilities	15,000
Structure Steel	20,000
Riser mooring system etc.	3,400
Ballast	1,800
Pretension	11,500
Displacement	51,700

Motion	
Wind and current	10m
Wave	14m
Total	24m

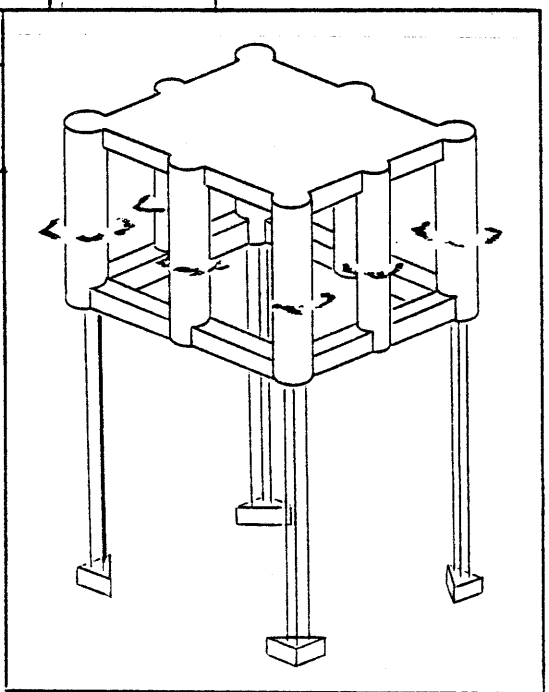


Table 4-2

British Petroleum	
References	(4,5,6)
Production Capacity	80,000(b/d)
Water Depth	150-600m
Site	Magnus(183m) in the North Sea

Environmental Condition	
Max. wave height	not available
Max. wind	
Max. current	
Max. tide	

Dimensions	
Deck length	85m
Deck width	85m
Draft	30m
Freeboard	33m
Deck height	?
# of column	4
Column diameter	?
# of mooring tethers	24

Load Summary (tons)	
Facilities	?
Structure Steel	13,000
Riser mooring system etc.	?
Ballast	?
Pretention	8,000
Displacement	30,000

Motion (200m water depth)	
Max. excursion	16m

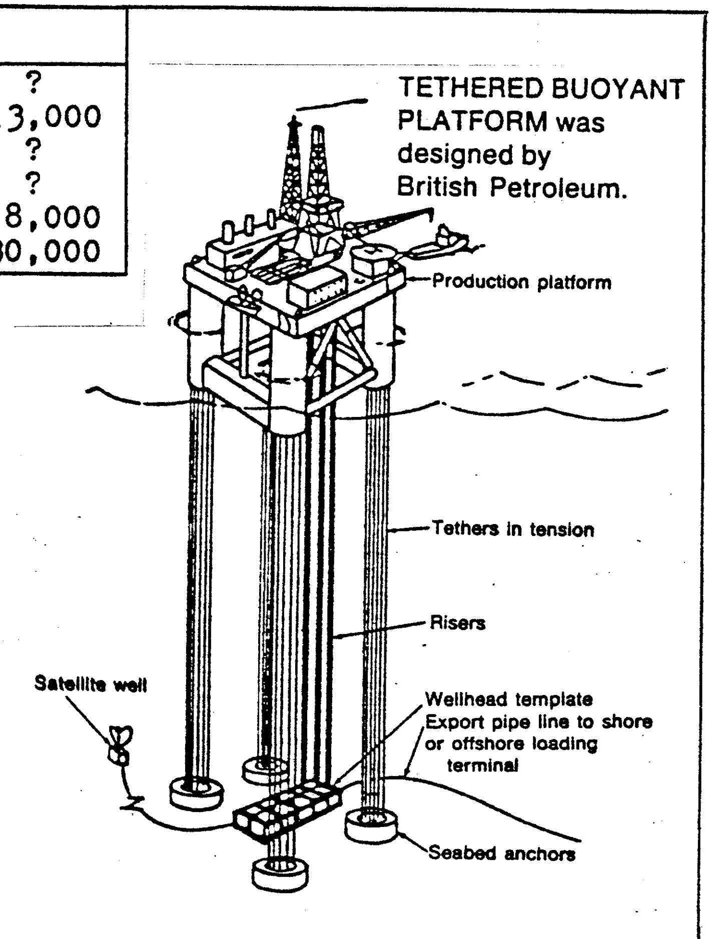


Table 4-3

Amoco	
Referencies	(6,7,8,9,10)
Production Capacity	26,000(b/d)
Water Depth	264m
Site	Gulf of Mexico

Environmental Condition	
Max. wave height	26m period 13-16sec
Max. wind	67m/s (1min duration)
Max. current	2.7m/s
Max. tide	?

Dimensions	
Deck length	61m
Deck width	58m
Draft	36m
Freeboard	19.5m
Deck height	9m
# of column	4
Column diameter	9m
# of mooring tethers	24

Load Summary (tons)	
Facilities	4,350
Structure steel	11,500
Riser mooring system etc.	3,474
Ballast	1,150
Pretension	9,526
Displacement	30,000

Motion (1,000ft deep)	
Wind & current	130ft
Wave	±25ft

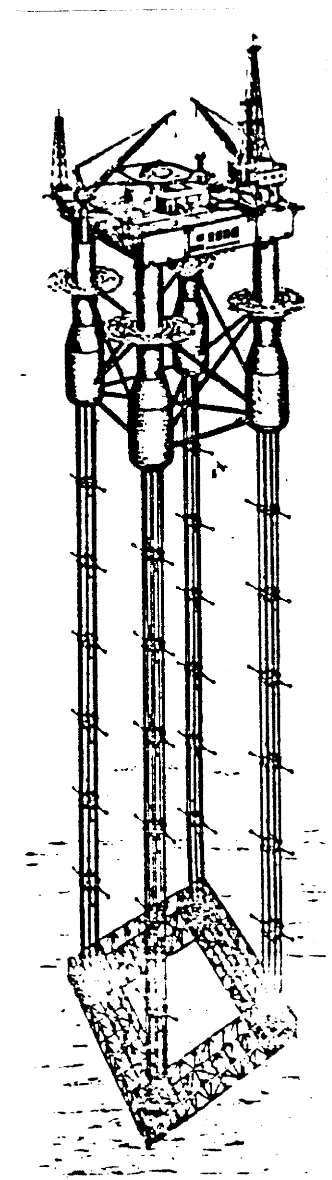


Table 4-4

Aker	
Referencies	(11)
Production Capacity	150,000(b/d)
Water Depth	150m
Site	northern North Sea

Environmental Condition	
Max. wave height	30m
Max. wind	56m/s
Max. current	1.35m/s
Max. tide	2.75m

Dimensions	
Deck length	86m
Deck width	86m
Draft	32m
Freeboard	25.6m
Deck height	9m
# of column	4
Column diameter	16m
Mooring tethere	Cables

Load Summary (tons)	
Facilities	?
Steel Structure	15,000
Riser mooring system etc.	?
Ballast	2,100
Pretension	10,000
Displacement	41,100
Payload	14,000

Motion
15% of the water depth for 150m water depth
12.5% of the water depth for 300m water depth

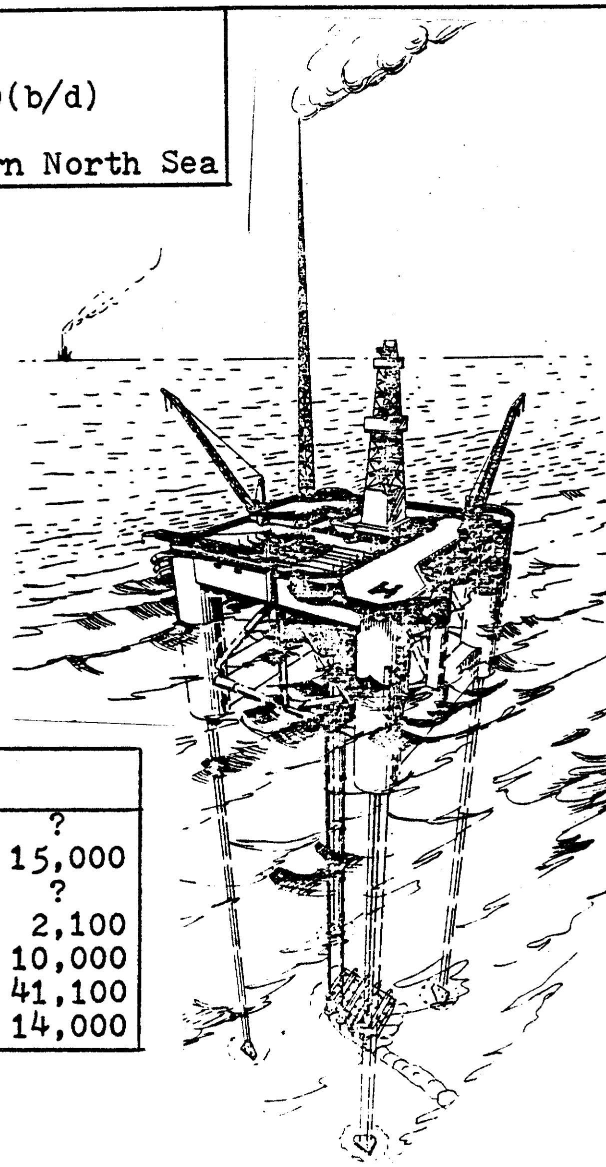


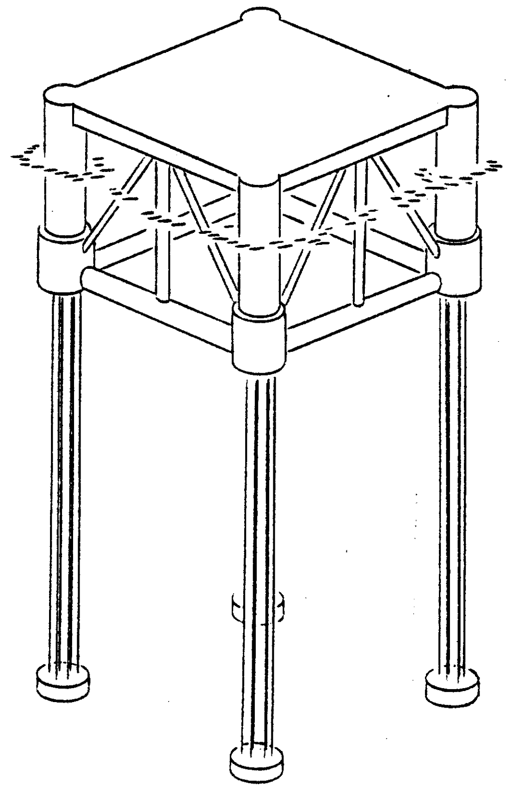
Table 4-5

Tecnomare	
Reference	(12)
Production Capacity	?
Water Depth	600m
Site	the North Sea

Environmental Condition	
Max. wave height	30m
Max. wind	56m/s
Max. current	1.34m/s
Max. tide	2.75m

Dimensions	
Deck length	96m
Deck width	96m
Draft	35m
Freeboard	?
Deck height	?
# of column	4
Column diameter	?
# of mooring tethere	24

Load Summary (tons)	
Facilities	?
Structure Steel	23,500
Riser mooring system etc.	?
Ballast	?
Pretention	15,000
Displacement	64,500
Payload	26,000



CHAPTER V

PURPOSE

The design of a TLP is rater complicated, because it is required to satisfy many requirements which may be conflicting with each other.

The following is a summary of the major design requirements.

(1) Fabrication requirements
Most of the TLP dimensions are controlled by the available capacity of graving docks, especially its overall width.

(2) Dynamic requirements
Its natural frequencies must be kept outside the wave frequency range and the vertical force variation must be minimized. These two requirements govern both the TLP dimensions(esp. column diameter & lower hull size) and the vertical mooring configuration.

(3) Structural requirements
Maintaining a positive minimum mooring tension requires careful selection of geometric parameters and appropriate pretention.

(4) Material requirements

The vertical mooring system must be designed so that it can stands both the maximum tension and life-time fatigue damage.

(5) Operational requirements

Operational requirements dictate that geometric parameters shoud be chosen so that the maximum horizontal excursion is less than the allowable one and that the TLP has a sufficient freeboard at the maximum offset position.

(6) Stability requirements

The TLP must maintain positive stability in the floating mode(tow-out condition) To avoid capsizing or severe damage when an anchor fails, it is necessary to have positive stability in the operating mode as well.

(7) Economical requirements

Cost should be minimized.

Although many studies have been carried out (which allow us to predict the TLP's performance reasonably well) and several designs have been developed based on these results, these sophisticated methods are costly and require time and accurate data. Thus these methods are not appropriate for the preliminary design, which requires simple estimates without detailed data.

It is quite rare that accurate long term wave data are given at the preliminary stage. Often times engineers are required to design based on the guesstimate of the maximum wave height. The purpose of this thesis is to present a simple and practical design procedure, which enables fast estimate at the preliminary design stage.

CHAPTER VI

ANALYSIS OF PUBLISHED DATA

6.1 Estimate of Equipment Weight

The first step of this analysis is to estimate equipment weight. Equipment weight can be classified into 3 categories.

(1) Riser & Mooring System; W_r

production risers, structural risers, winches etc. (excluding pretension)

(2) Production and/or Drilling System; W_p

all the equipments directly related to production and/or drilling

(3) Auxiliary Equipment; W_a

outfitting, machinery, electrical, piping, ventilation, bilge & ballast and accommodation.

Subsequently the purpose is to establish reasonable correlation between equipment weight and important parameters. For this purpose data were collected and the analysis of this data indicates that water depth, displacement of the TLP and production capacity are the dominant factors.

Table 6-1 shows weight data of proposed designs. First, let us consider the riser & mooring system.

Assuming that the weight is a function of displacement and water depth, and using design data of Conoco and Amoco, the following equation is derived.

$$W_r = 0.44 * L_{wd} * \Delta / 1,000 \qquad \text{(Eq.6-1)}$$

W_r; Weight of riser & mooring system (ton)

L_{wd}; Water depth (m)

Δ ; Displacement (ton)

Table 6-2 shows production & auxiliary equipment weight of several production platforms. Note Eq.6-1 is used to estimate the weight for several TLPs. In Fig.6-1 production & auxiliary equipment weight data are plotted against production capacity. It is clearly seen that production & auxiliary equipment weight is strongly related to production capacity.

An article "Cost correlated for N. Sea platforms" proposes two equations for superstructure cost.(29)

$ce = 0.415 * P_c + 26.32$ for low gas and oil ratio

$ce = 1.4064 * P_c$ for high gas and oil ratio

ce; Superstructure cost ($million,1976)

Another article "The Brent Oil-Field" (28) gives the following information. $P_c = 550,000$ b/d and equipment weight is 82,680 tons for the Brent Field. Assuming that superstructure cost is proportional to equipment weight, the following equations are derived.

Equipment weight=$44.4*P_c+2,814$ (tons) for low GOR

Equipment weight=$150.3*P_c$ (tons) for high GOR

These two lines are also drawn in Fig.6-1.

Considering these two lines and all other plotted data, the following equations are derived.

$$W_p=cP_c \quad \text{(Eq.6-2)}$$

$$W_a=93.3*\Delta/1,000 \quad \text{(Eq.6-3)}$$

W_p; Weight of production and/or drilling system (ton)

W_a; Auxiliary equipment weight (ton)

P_c; Production capacity (10^3b/d)

c; Coefficient $44.4 \leqq c \leqq 88.8$

The reason why c varies from 44.4 to 88.8 can be explained by 2 uncertainties.

(1) The characteristics of the fluids produced-some crude oils require more processing equipment, while others do not.

(2) The function assigned to the TLP operation. It can be designed solely for production or both production and drilling.

Table 6-4 compares Amoco design to estimated values by Eq.6-2 and Eq.6-3. They agree relatively well.

Finally putting all together the following equation is derived.

$$W_e=(93.3+0.44*L_{wd})*\Delta/1,000+cP_c \quad \text{(Eq.6-4)}$$

W_e; Equipment weight (tons)

Table 6-3 shows weight estimates for the same designs listed in Table 6-1. Estimated values agree well with actual value.

Table 6-4

	Amoco	Estimate
Prod.&Drill	2,100 t	1,154-2,308 t
Aux. Equip	2,200 t	2,799 t

Table 6-1 Technical Data of Proposed TLPs

References	11	4,5,6	1,2,3	6,7,8,9,10	13
Steel Structure(t)	15,000	13,000	20,000	11,500	10,887
Ballast(t)	2,100		1,800	1,150	
Pretension(t)	10,000	8,000	11,500	9,526	9,072
Total Equip. Weight(t)	14,000	9,000	18,400	7,824	11,340
Displacement(t)	41,100	30,000	51,700	30,000	31,298
Prod. Capacity(b/d)	150,000	80,000	120,000	26,000	100,000
Riser & Mooring(t)			3,400	3,474	
Prod. & Aux.(t)			15,000	4,350	
Designed by	Aker	B.P.	Conoco	Amoco	Earl & Wright

Table 6-3 Weight Estimates by Hypothetical Equation

Designed by	Aker	B.P.	Conoco	Amoco	Earl & Wright
Riser & Mooring	2,713	2,640	3,344	3,485	3,278
Prod. Equip.	6,660~	3,552~	5,328~	1,154~	4,440~
	13,320	7,104	10,656	2,308	8,880
Auxiliary	3,835	2,799	4,824	2,799	2,920
Total Equip. Weight	13,208~	8,991~	13,496~	7,438~	10,638~
	19,868	12,543	18,824	8,592	15,078

Fig 6-1

Production Capacity vs Prod. & Aux. Equip. Weight

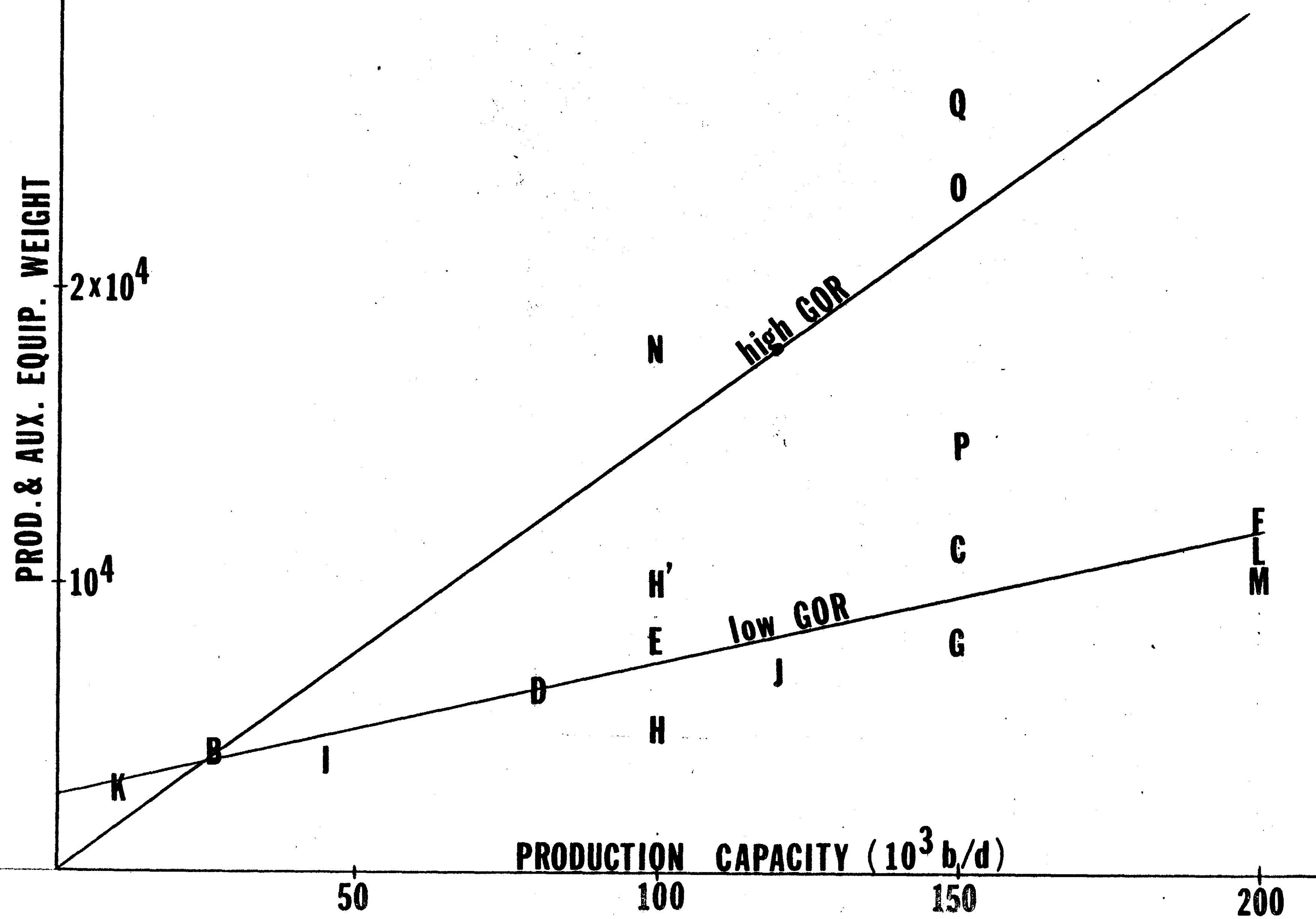


Table 6-2 Production & Auxiliary Equipment Weight

References	Prod.Cap.(b/d)	Prod. & Aux. Equipment Weight (ton)	Type	Comments
1,2,3	120,000	15,000	TLP	
6,7,8,9,10	26,000	4,350	TLP	
11	150,000	11,287	TLP	=14,000-2,713(est. R&M)
4,5,6	80,000	6,360	TLP	=9,000-2,640(est. R&M)
13	100,000	8,062	TLP	=11,340-3,278(est. R&M)
22	200,000	12,269	Jacket	
23	150,000	8,000	Gravity	
24	100,000	?	TLP	5,000~10,000
25	45,000	4,000	Jacket	
26	120,000	7,000	Semi	
27	10,000	3,000	Jacket	
27	200,000	10,932	Jacket	
27	200,000	10,433	Jacket	
28	100,000	18,000	Jacket	
28	150,000	23,480	Jacket	
28	150,000	14,800	Jacket	
28	150,000	26,400	Jacket	

6.2 Relations Among Parameters

Fig.6-2 plots various steel structure weight against displacement. Table 6-5 shows the corresponding data to Fig.6-2. From Fig.6-2 the following equation is derived.

$$W_{ss}=0.37*\Delta \quad \text{(Eq.6-5)}$$

W_{ss}; Steel structure weight (ton)

Δ ; Displacement (ton)

Fig.6-3 shows Deck Size vs. Deck Load and Table 6-6 lists the corresponding data. This information is useful to check the adequacy of deck size for a given deck load.

Fig.6-4 shows Deck Size vs. Deck Weight and Table 6-7 lists the corresponding data. From Fig.6-4 the following equation is derived.

$$W_d=(0.74\pm0.16)*L*B \quad \text{(Eq.6-6)}$$

W_d; Deck weight (tons)

L; Deck length (m)

B; Deck width (m)

Fig.6-5 plots displacement against draft. Table 6-8 shows the corresponding data. Plotted data tail-off where draft exceeds 30m probably because of construction constraints.

Fig 6-2

Steel Structure vs Displacement

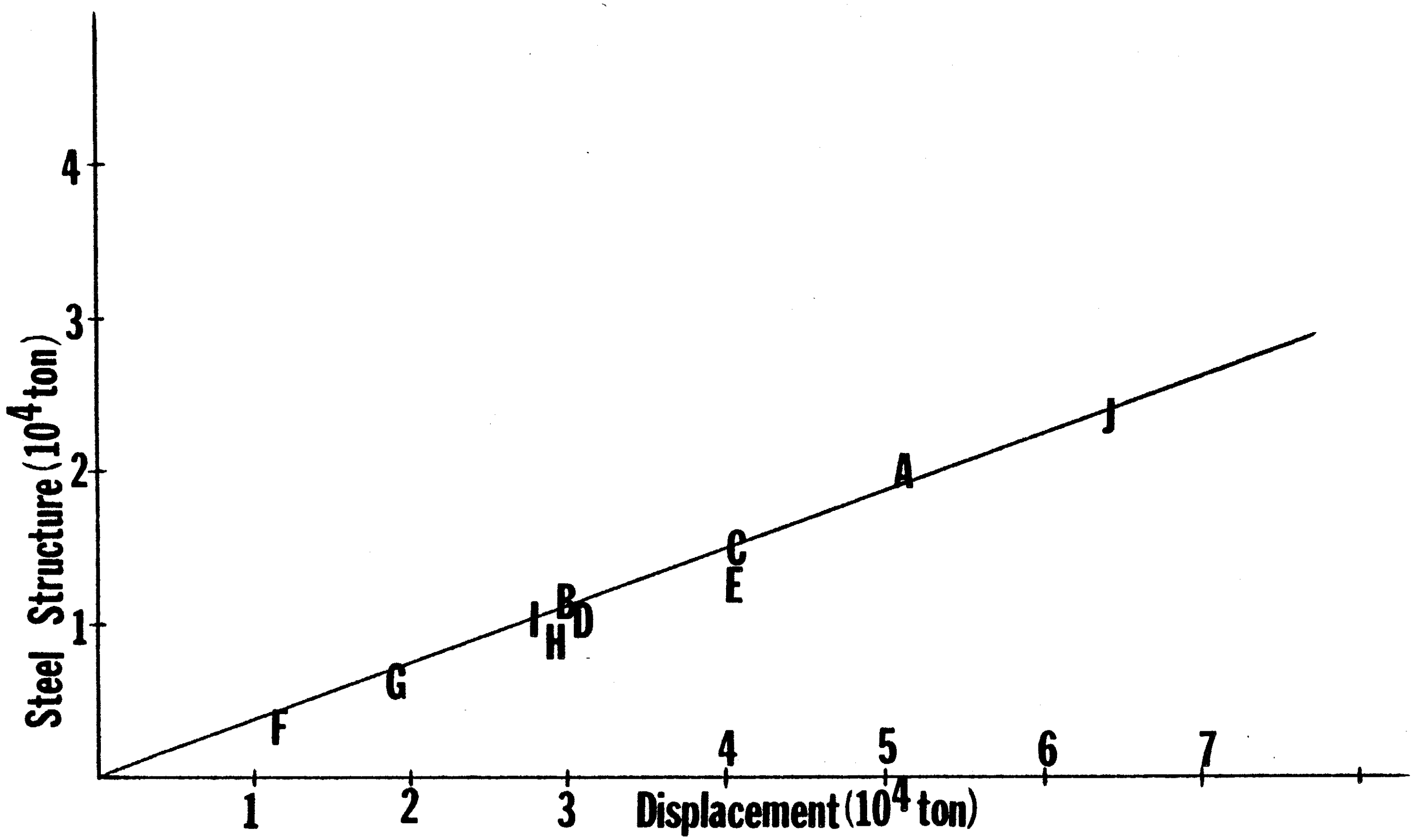


Table 6-5

Data Mark	Displacement	Steel Structure	References	Type
A	51,700 t	20,000 t	1,2,3	TLP
B	30,000 t	11,500 t	6,7,8,9,10	TLP
C	41,100 t	15,000 t	11	TLP
D	31,298 t	10,887 t	13	TLP
E	41,000 t	12,500 t	30	Semi
F	11,697 t	3,175 t	31	Semi
G	19,051 t	6,804 t	31	Semi
H	29,263 t	8,687 t	32	Semi
I	28,000 t	10,600 t	33	Semi
J	64,500 t	23,500 t	12	TLP

Fig 6-3

Deck Size vs Deck Load

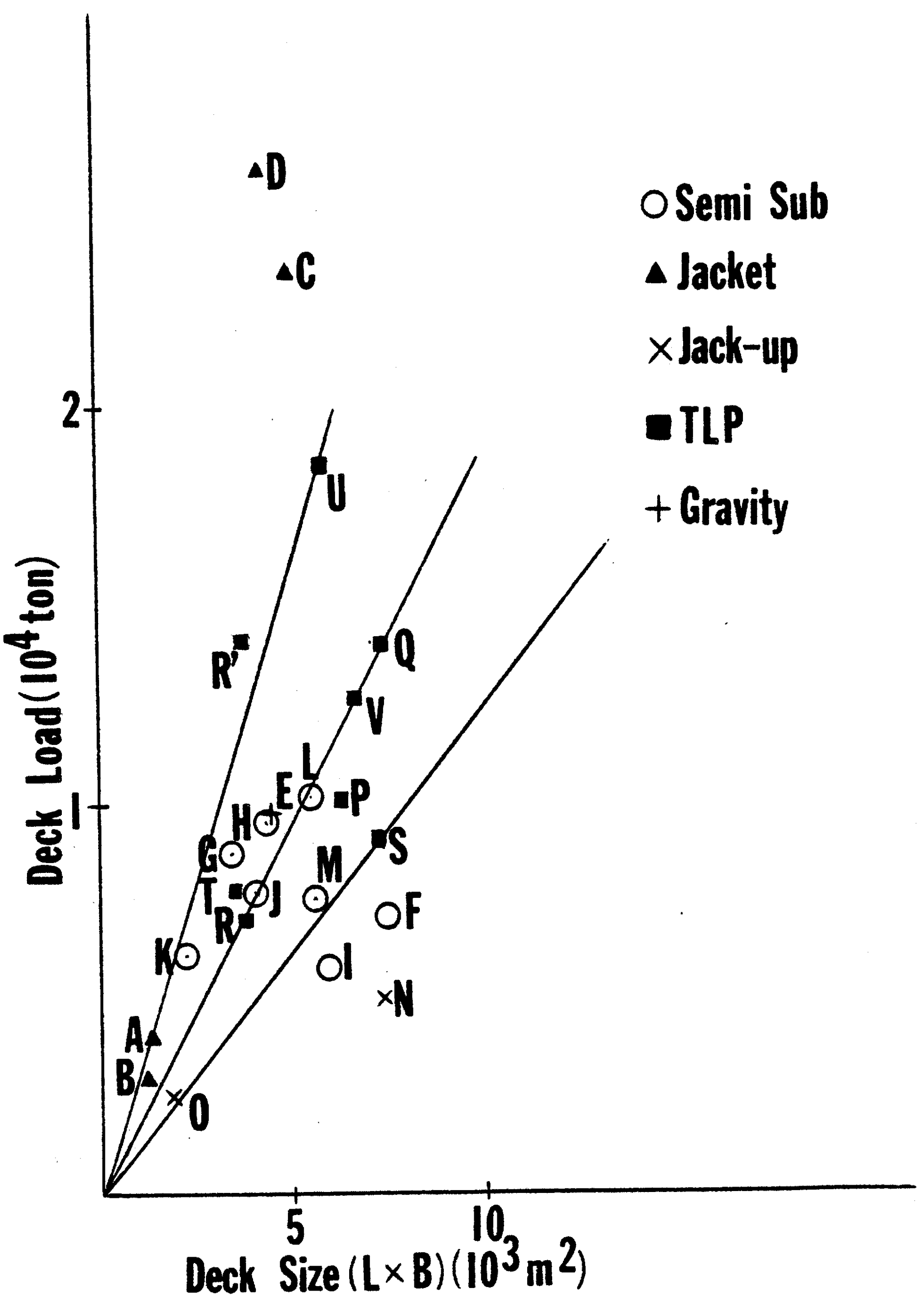


Table 6-6

Data Mark	Deck Load (t)	Deck Size (m^2)	Type	References
A	4,000	1,320	Jacket	25
B	2,948-3,402	1,034-1,100		27
C	23,522	5,002		22,34
D	26,000	4,420	Jacket	36
E	9,500	4,200	Gravity	2
F	7,100	7,641	Semi	42
G	8,628	3,418		43
H	9,396	4,200		45
I	5,785	5,869		46
J	7,366	3,861		32
K	6,175	2,253		51
L	10,092	5,576		52
M	5,593	5,593	Semi	33
N	5,000	7,544	Jack-up	54
O	2,631	1,880	Jack- up	56
P	10,000	6,335	TLP	58,59
Q	14,000	7,396		58,59
R	7,000-14,000	3,700		60
S	9,000	7,225		4
T	7,824	3,538		7,10
U	18,400	5,772		1
V	12,600	6,656	TLP	61

Fig 6-4

Deck Size vs Deck Weight

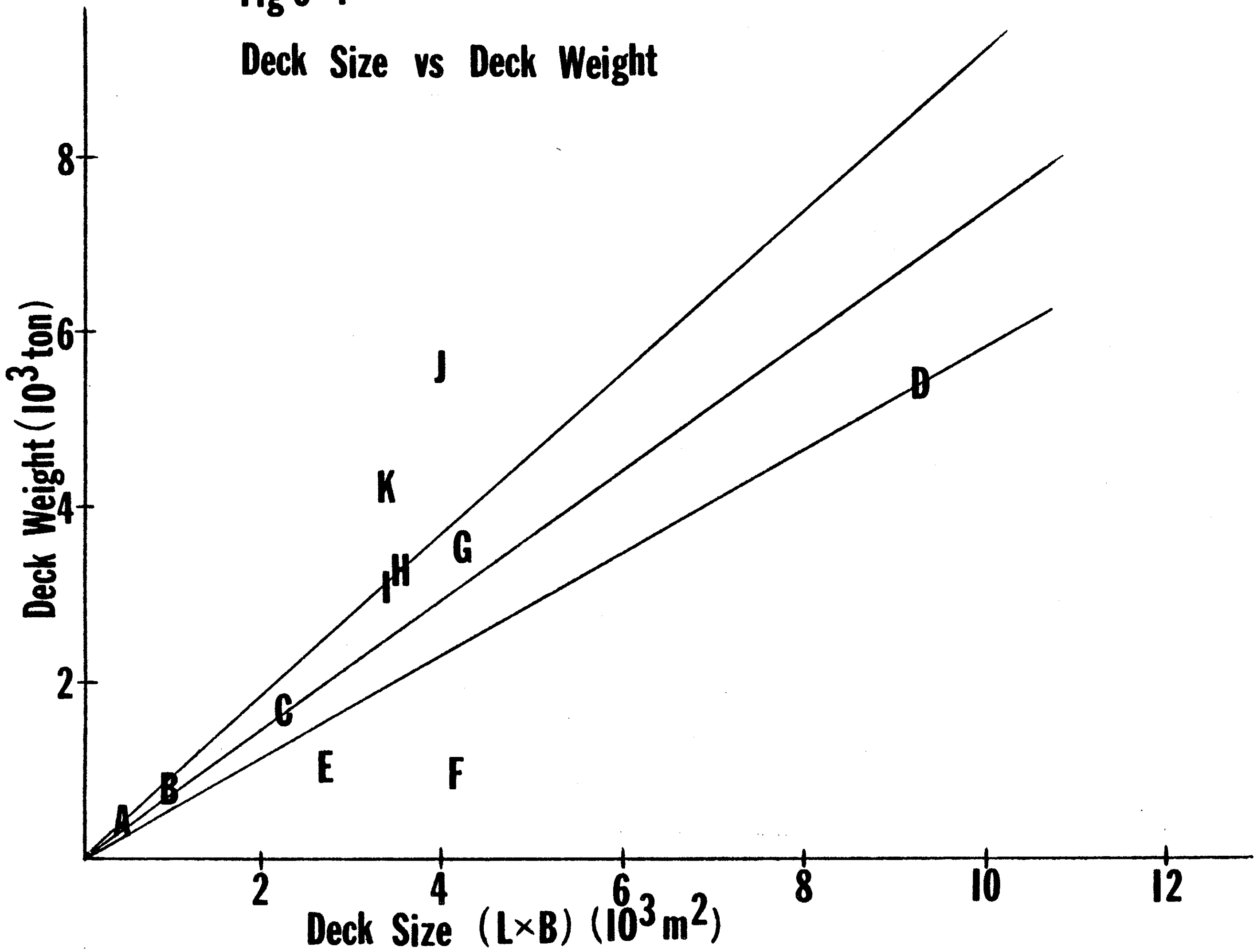


Table 6-7

Data Mark	Deck Size (m^2)	Deck Weight (t)	Type	References
A	441	408-499	Jacket	27
B	1,034	680-907		27
C	2,233	1,680		35
D	9,290	5,443		27
E	2,787	1,043		27
F	4,180	998	Jacket	27
G	4,200	3,500	Gravity	37
H	3,538	3,250	TLP	7,10
I	3,400	3,100	Gravity	28
J	4,000	5,600		28
K	3,400	4,200	Gravity	28

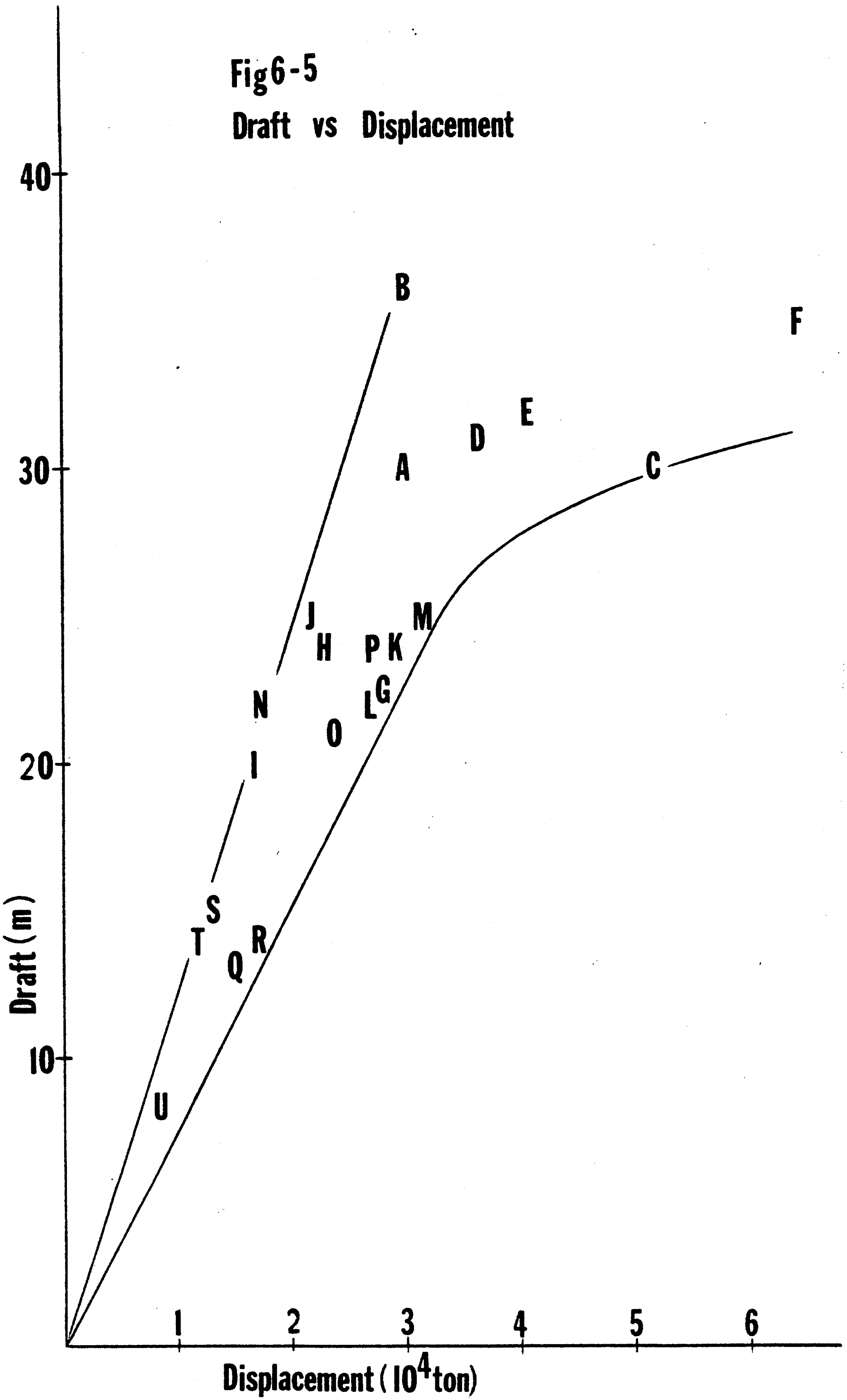


Fig 6-5
Draft vs Displacement
40
30
20
10
Draft (m)
1
2
3
4
5
6
Displacement (10^4 ton)
A
B
C
D
E
F
G
H
I
J
K
L
M
N
O
P
Q
R
S
T
U

Table 6-8

Data Mark	Draft (m)	Displacement (t)	Type	References
A	30	30,000	TLP	4
B	36	30,000		7,10
C	30	51,700		1
D	31	36,600		61
E	32	41,100		11,58,59
F	35	64,500	TLP	12
G	22.5	28,000	Semi	33
H	24	22,809		45
I	20	16,412		46
J	25	21,773		48
K	24	29,263		32
L	22	26,824		49
M	25	31,530		62
N	22	17,300		51
O	21	23,715		64
P	24	27,322		
Q	13	15,017		
R	14	17,017		
S	15	12,934		
T	14	11,685		
U	8.2	8,433	Semi	64

6.3 Estimate of Freeboard

In the case of the TLP the freeboard must satisfy the following inequality.

$$f \geq H_w/2 + (\text{tide}) + (\text{sinkage due to horizontal excursion}) + (\text{freeboard allowance})$$

f; Freeboard (m)

H_w; Max. wave height (m)

In the case of the semisubmersible, tide and sinkage due to horizontal excursion have no effect on freeboard, thus

$$f \geq H_w/2 + (\text{freeboard allowance})$$

ABS rule requires a freeboard allowance of 1.52m (5ft)(65). For simplicity, horizontal excursion is assumed to be 15% of the water depth because 15% is probably the maximum allowable horizontal excursion from the production riser's point of view.

Tide varies from site to site but usually ranges from 2 to 3 meters including astronomical and storm tides. For simplicity the sum of tide and freeboard allowance is assumed to be 4m.

Based on these assumptions the following equations are formulated for the estimate of freeboard.

$$f_e = H_w/2 + 4 + 0.0113 * L_{wd} \quad \text{(for the TLP)} \qquad \text{(Eq.6-7)}$$

$f_e = H_w/2 + 1.52$ (for semisubmersible) (Eq.6-8)

f_e; Estimated freeboard (m)

L_{wd}; Water depth (m)

Table 6-9 lists freeboard, estimated freeboard and necessary information for the estimate and Fig.6-6 plots these data.

Fig.6-6 indicates the following 2 facts:

(1) Estimated f_e gives minimum required freeboard.

(2) Actual freeboard satisfies the following relation:

$f_e \leq f \leq f_e + 5.5$ (Eq.6-9)

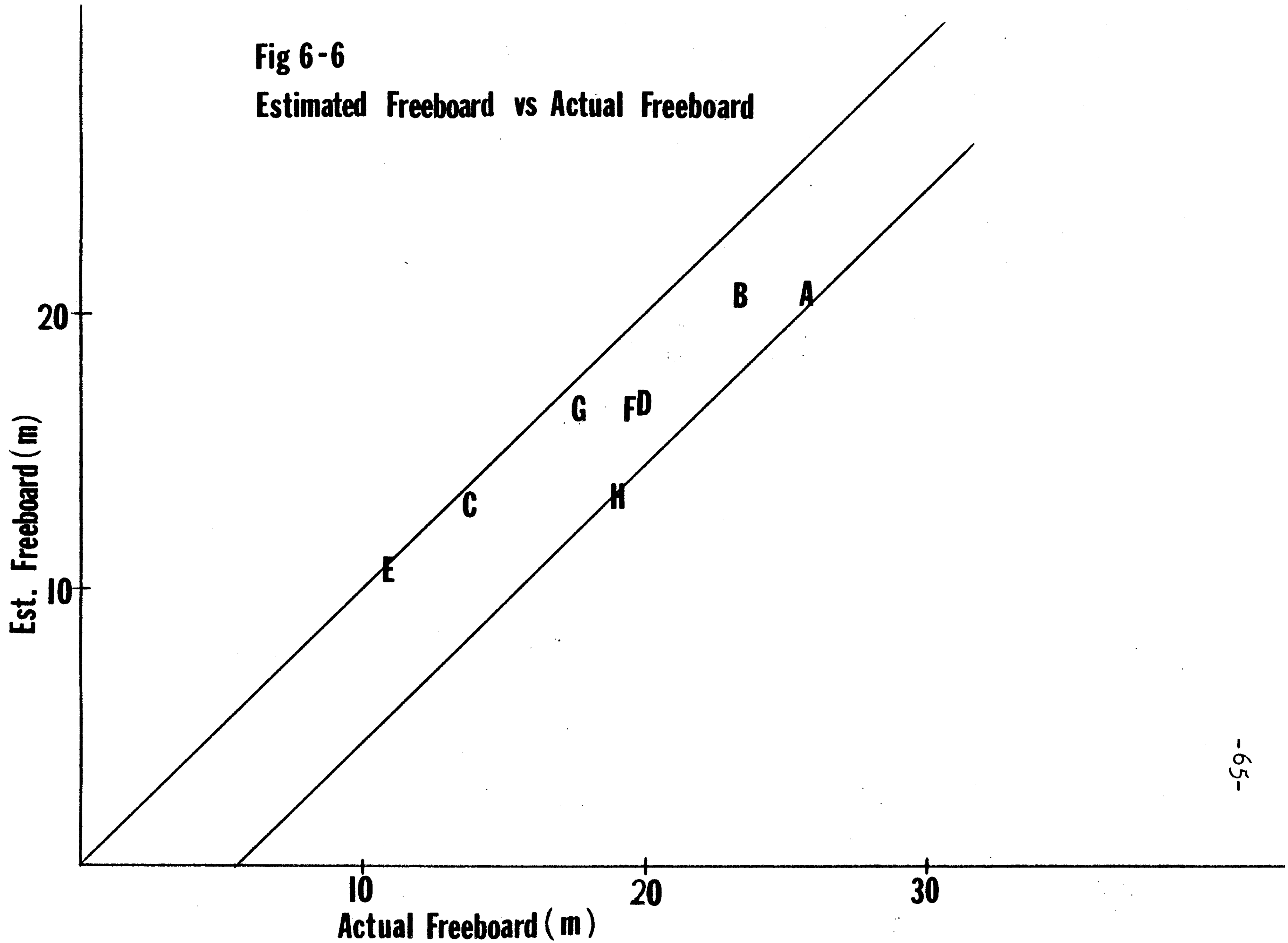


Fig 6-6
Estimated Freeboard vs Actual Freeboard
Est. Freeboard (m)
Actual Freeboard (m)
10
20
10
20
30
A
B
C
D
E
F
G
H

Table 6-9

Data Mark	H_w	L_{wd}	f_e	f	Type	References
A	30	150	20.7	25.6	TLP	11,58,59
B	30	147	20.7	23.7	TLP	1,2,3
C	23		13.02	13.7	Semi	38,39
D	30		16.52	20		33
E	18.5		10.77	11		46
F	30		16.52	19.5		48
G	30		16.52	17.7		49
H	24		13.52	19	Semi	51

Table 6-11

Data Mark	Displacement (ton)	Jacket Volume (m^3)	C'
A	51,700	75,600	0.0425
B	30,000	34,200	0.0546
C	64,500	90,200	0.0445
D	28,000	34,200	0.0509
E	22,809	25,600	0.0554
F	29,263	35,200	0.0517
G	17,300	23,300	0.0462
		Average	0.0494

6.4 Estimate of Jacket Weight

A reasonable estimate of jacket weight is necessary to facilitate the estimate of light weight and KG. Table 6-10 lists data necessary for the estimate. Deck weights are estimated using Eq.6-6 when the information is not available. From Fig.6-7, the following equation is derived.

$$W_{js}=0.18*V_j$$

W_{js}; Jacket steel weight (tons)
V_j; Jacket volume (m^3)

Assuming that 2/3 of auxiliary equipment weight is distributed evenly all over jacket and 1/3 on the deck, total jacket weight is given by the following equation.

$$W_{jt}=W_{js}+93.3/1{,}000*2/3*\Delta$$

W_{jt}; Total jacket weight (tons)
Δ; Displacement (tons)

To express total jacket weight only by jacket volume, the following coefficient is introduced.

$$c'=(93.3/1{,}000*2/3*\Delta)/V_j$$

Table 6-11 lists values of c' for several different cases. The average of c' is 0.0494, thus

$$W_{jt}=(0.18+0.0494)V_j=0.23*V_j \qquad \text{(Eq.6-10)}$$

Fig 6-7

Jacket Volume vs Jacket Weight

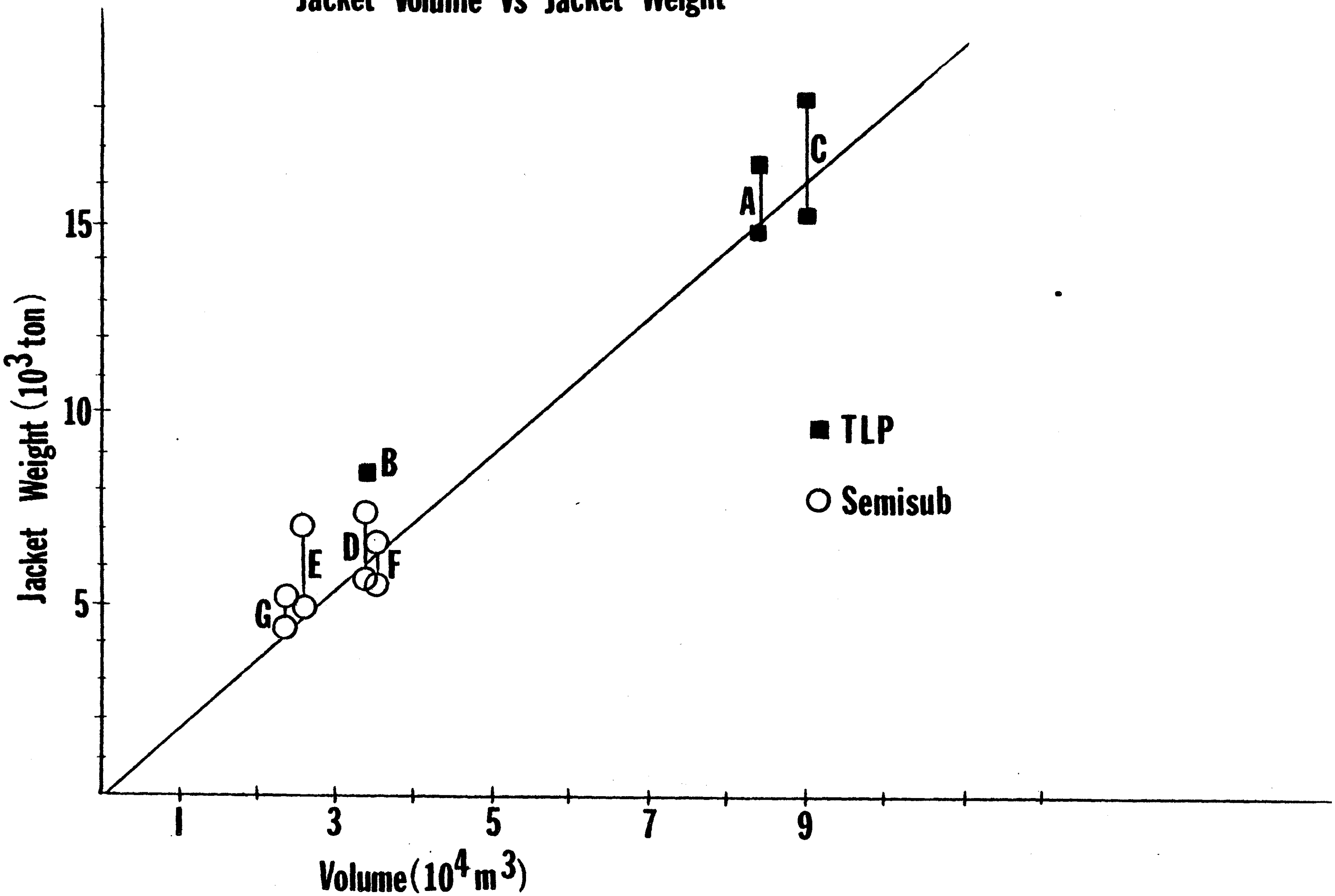


Table 6-10

Data Mark	Steel Structure (ton)	Deck Weight (ton)	Jacket Steel Weight (ton)	References
A	20,000	3,400-5,200	14,800-16,600	1,2,3
B	11,500	3,250	8,250	6,7,8,9,10
C	23,500	5,300-8,300	15,200-18,200	12
D	10,600	3,400-5,300	5,300- 7,200	33
E	8,400	2,400-3,800	4,600- 6,000	45
F	8,700	2,200-3,500	5,200- 6,500	32
G	6,400	1,300-2,100	4,300- 5,100	51

Data Mark	Displacement (ton)	Column Diameter (ton)	Freeboard (m)	Jacket Volume (m^3)	Type
A	51,700	15	23.7	83,900	TLP
B	30,000	9	19.5	34,200	)
C	64,500	17	30	90,200	TLP
D	28,000	9.4	19.825	34,200	Semi
E	22,809	9/5.4	9.75	25,600	)
F	29,263	9.4	15.9	35,200	(
G	17,300	9.4	18.6	23,300	Semi

6.5 Estimate of Light Weight

Light weight can be estimated by two different methods.

(a) One is obtained from Eq.6-4 and Eq.6-5

$$W_e=(93.3+0.44*L_{wd})*\Delta/1,000+c*P_c \qquad \text{(Eq.6-4)}$$

$$W_{ss}=0.37*\Delta \qquad \text{(Eq.6-5)}$$

Assuming that only 1/3 of the riser and mooring system is included in Δ_1, but not production & structural risers, the following equation is formulated

$$\Delta_1=c*P_c+(0.4633+0.44*L_{wd}/3,000)*\Delta \qquad \text{(Eq.6-11)}$$

(b) The other is obtained from Eq.6-10 and Eq.6-6

$$W_{jt}=0.23*V_j \qquad \text{(Eq.6-10)}$$

$$W_d=0.74*L*B \qquad \text{(Eq.6-6)}$$

$$\Delta_1=0.23*V_j+0.74*L*B+c*P_c+(93.3+0.44*L_{wd})*\Delta/3,000 \qquad \text{(Eq.6-12)}$$

Table 6-12 compares Δ_1 by two methods. They agree generally well (within 10%)

Table 6-12

Conoco Design $W_p = cP_c = 10{,}000$, $L_{wd} = 147$			
Δ	72,380	62,040	54,285
Eq 6-20	45,094	40,081	36,321
Eq 6-21	43,199	38,990	35,859

Aker Design $W_p = cP_c = 7{,}000$, $L_{wd} = 150$			
Δ	60,000	50,000	40,000
Eq 6-20	36,118	31,265	26,412
Eq 6-21	35,682	31,662	27,686

Amoco Design $W_p = cP_c = 1{,}500$, $L_{wd} = 264$			
Δ	39,000	30,000	24,000
Eq 6-20	21,079	16,561	13,548
Eq 6-21	18,387	15,010	12,776

B.P. Design $W_p = cP_c = 3{,}560$, $L_{wd} = 200$			
Δ	36,000	30,000	24,000
Eq 6-20	21,295	18,339	15,383
Eq 6-21	23,147	20,684	18,991

CHAPTER VII

ANALYSIS OF THE TLP

7.1 Design Parameters and Requirements

Two simple models of the TLP are adopted. These are shown in Fig.7-1 and Fig.7-2. In either case, we have 10 unknown parameters to be decided. They are;

Δ,D,d,R,r,f,h,H,L,B for model 1 and

Δ,D,d,P_b,r,f,h,P_h,L,B for model 2.

We need ten equations or correlations to decide these parameters.

First we have 3 simple relations among parameters.

(1) $f=D-d$

(2) $h+H=d$

(3) $\Delta=4(\pi R^2H+\pi r^2h)\rho$ for model 1

$\Delta=(2L_bL_hL+4\pi r^2h)\rho$ for model 2

Eq.6-9 gives another relation for freeboard.

(4) $f_e<f<f_e+5.5$ (Eq.6-9)

To minimize dynamic tension variation, r is determined by R,d,ω and (wave frequency).

This relation will be discussed in the next section.

(5) $r=f(R,d,\Delta,\omega)$

To avoid large wave dynamic forces the lower hull must always stay submerged.

(6) $h > H_w/2$

(7) The TLP also has to satisfy the stability requirements. This will be discussed in Section 7.3.

(8) The maximum horizontal excursion must be less than the allowable one.

$$x_{max} < x_{all}$$

(9) From the economical point of view the displacement should be minimized.

(10) From the structural point of view the minimum tension must be positive and the maximum tension must be minimized.

(11) The natural frequency of the TLP must be outside of the range of wave spectrum.

Although requirements (1) through (7) are relatively easy to handle, (8) through (11) are not. So the following procedure is adopted: First, input Δ,L,B and d and calculate the other 6 parameters by a computer program so that they satisfy requirements (1) through (7). After the computation the design is completed by satisfying requirements (8) through (11).

So, to find the optimum design, it is necessary to try several sets of Δ,L,B and d and evaluate the results.

Fig.7-3 shows the conceptual flow of the procedure.

The maximum dynamic excursion is discussed in Section 6-4. The wind & current forces are discussed in Section 6-5. The restoring force is discussed in Section 6-6.

Fig 7-1 Model 1

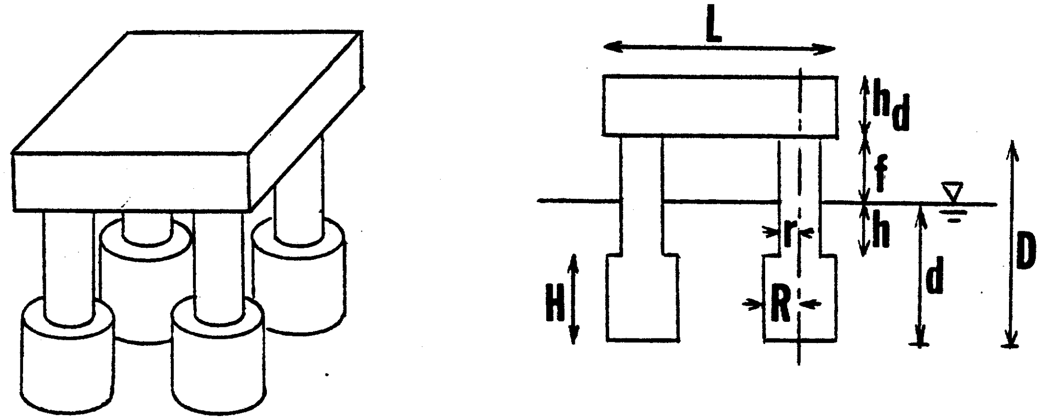


Fig 7-2 Model 2

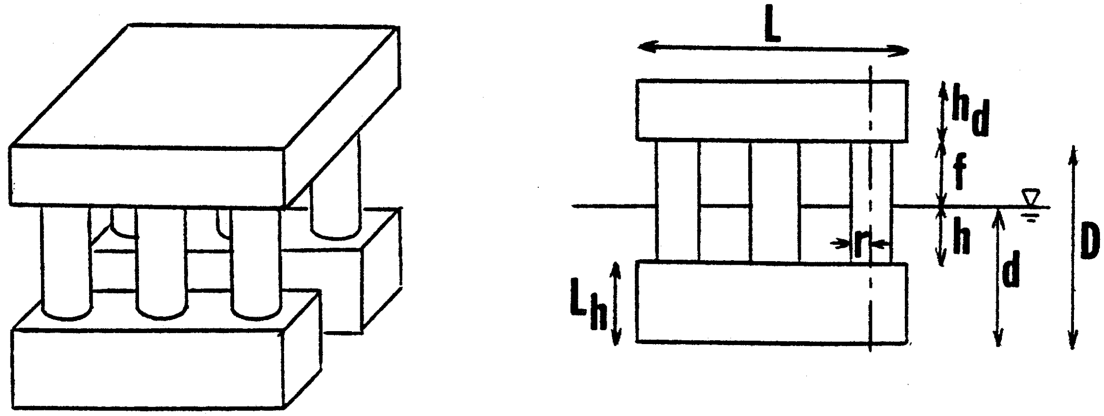
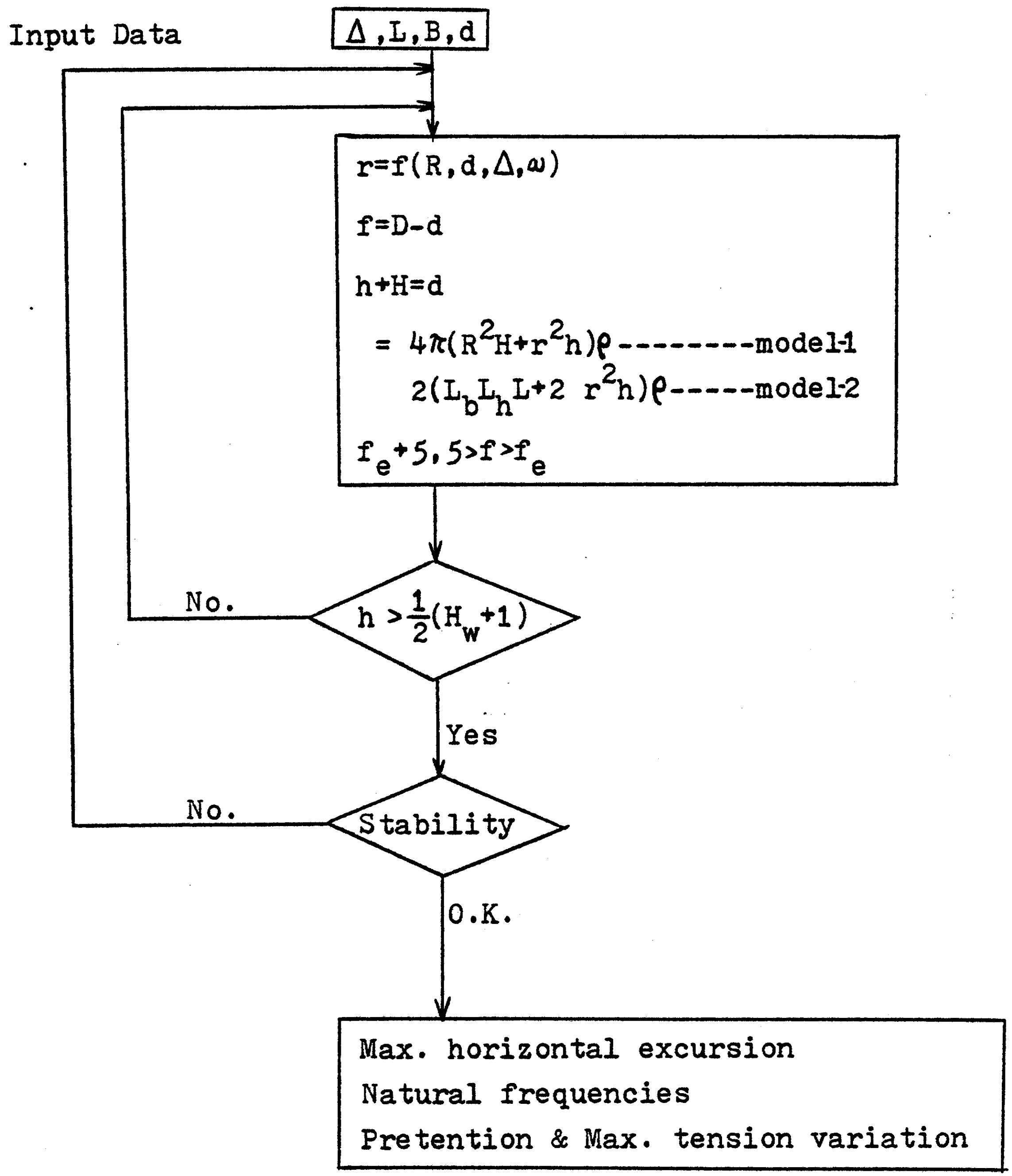


Fig 7-3


Input Data
Δ,L,B,d
r=f(R,d,Δ,ω)
f=D-d
h+H=d
= 4π(R²H+r²h)ρ---------model-1
2(L_b L_h L+2 r²h)ρ-----model-2
f_e+5,5>f>f_e
No.
h > ½(H_w+1)
Yes
No.
Stability
O.K.
Max. horizontal excursion
Natural frequencies
Pretention & Max. tension variation

7.2 Minimizing the Dynamic Tension Variation

(1) Model 1

The vertical dynamic force on the TLP is given by thefollowing equation.

$$F_v=\left(\nabla e^{-\frac{\omega^2}{g}\frac{d}{2}}+4\pi R^3/3*4e^{-\frac{\omega^2}{g}d}\right)\frac{\omega^2\rho}{g}a*\cos\omega t-4\pi r^2\rho a*\cos\omega t$$

∇; Displacement volume ($=\Delta/\rho$) (m^3) (Eq.7-1)

ω; Frequency of wave

d; Draft (m)

R; Radius of the enlarged section (m)

g; Acceleration of gravity ($9.8m/s^2$)

ρ; Density of sea water ($1.025t/m^3$)

a; Wave amplitude (m)

r; Radius of the column (m)

t; Time

The first term represents displacement and the second added mass. These two represent inertia force and the third term buoyancy force.

The problem is to minimize this vertical dynamic force for certain range of wave period. For a deterministic approach, DnV requires to consider the following range of wave period.

$$\sqrt{6.5H_w} < T < \sqrt{15H_w} \qquad \text{(Eq.7-2)}$$

$$\omega_{min}=2\pi/\sqrt{15H_w} < \omega < 2\pi/\sqrt{6.5H_w}=\omega_{max}$$

To minimize the vertical dynamic force within this range, the following equation must hold.

$$(\nabla e^{-\frac{\omega_{min}^2}{g}\frac{d}{2}} + 4\pi R^3/3*4e^{-\frac{\omega_{min}^2}{g}d})\frac{\omega_{min}^2}{g} - 4\pi r^2$$

$$= 4\pi r^2 - (\nabla e^{-\frac{\omega_{max}^2}{g}\frac{d}{2}} + 4\pi R^3/3*4e^{-\frac{\omega_{max}^2}{g}d})\frac{\omega_{max}^2}{g}$$

Denoting the wave frequency at which vertical dynamic force is set zero by ω_0, and ignoring the frequency effect on the exponential decay, we have;

$$(\nabla e^{-\frac{\omega_0^2}{g}\frac{d}{2}} + 4\pi R^3/3*4e^{-\frac{\omega_0^2}{g}d})\frac{\omega_0^2}{g} = 4\pi r^2$$

$$\omega_{min}^2 - \omega_0^2 = \omega_0^2 - \omega_{max}^2$$

$$\omega_0^2 = (\omega_{max}^2 + \omega_{min}^2)/2 = 4.3527/H_w \qquad \text{(Eq.7-3)}$$

The maximum vertical dynamic force variation is given by the following relation;

$$\Delta F_v = \left\{(\nabla e^{-\frac{\omega_{max}^2}{g}\frac{d}{2}} + 4\pi R^3/3*4e^{-\frac{\omega_{max}^2}{g}d})\frac{\omega_{max}^2}{g} - 4\pi r^2\right\}\rho a$$

$$= \left\{(\nabla e^{-\frac{\omega^2}{g}\frac{d}{2}} + 4\pi R^3/3*4e^{-\frac{\omega^2}{g}d})\frac{1}{g}(\omega_{max}^2 - \omega_0^2)\right\}\rho a$$

$$= 4\pi r^2\frac{\omega_{max}^2 - \omega_0^2}{2}\rho a = 4\pi r^2 \rho a\frac{1/6.5-1/15}{1/6.5+1/15} = 4.968 r^2 \rho a$$

$$= 5.093 r^2 a \qquad \text{(Eq.7-4)}$$

In addition to this, dynamic tension variation must include the pitching effect. (It will be discussed in Section 7-8) Finally the optimal diameter of the column is given by the following equation:

$$r^2 = \frac{\omega_0^2}{4\pi g}(\nabla e^{-\frac{\omega_0^2}{g}\frac{d}{2}} + 16\pi R^3/3*e^{-\frac{\omega_0^2}{g}d}) \qquad \text{(Eq.7-5)}$$

Fig 7-4 shows Max. wave height vs. Wave period. Table 7-1 lists design data for platforms. Fig.7-4 clearly shows that the approach of setting the vertical dynamic force zero at $\omega_0^2=4.3527/H_w$ is reasonable. Two solid lines represent limits of wave period specified by DnV rule, and doted line $\omega_0=\sqrt{4.3527/H_w}$. Most design wave periods are plotted very close to this curve.

(2) Model 2

The vertical dynamic force on the TLP is given by the following expression.

$$F_v=(\nabla e^{-\frac{\omega^2}{g}\frac{d}{2}}+2LP_b{}^2e^{-\frac{\omega^2}{g}d})\frac{\omega^2\rho}{g}a^*\cos\omega t-S_a\rho a^*\cos\omega t$$

L; Deck length (assumed equal to the lower hull lengh)

P_b; Lower hull width

(Eq.7-6)

S_a; Surface area of columns

Similarly the optimum surface area of columns is given by

$$S_a=\frac{\omega_0^2}{g}(\nabla e^{-\frac{\omega_0^2}{g}\frac{d}{2}}+2LP_b{}^2e^{-\frac{\omega_0^2}{g}d}) \qquad \text{(Eq.7-7)}$$

and the maximum dynamic tension variation is given by

$$F_v=0.4053S_a a \qquad \text{(Eq.7-8)}$$

Fig 7-4

Max. Wave Height vs Wave Period

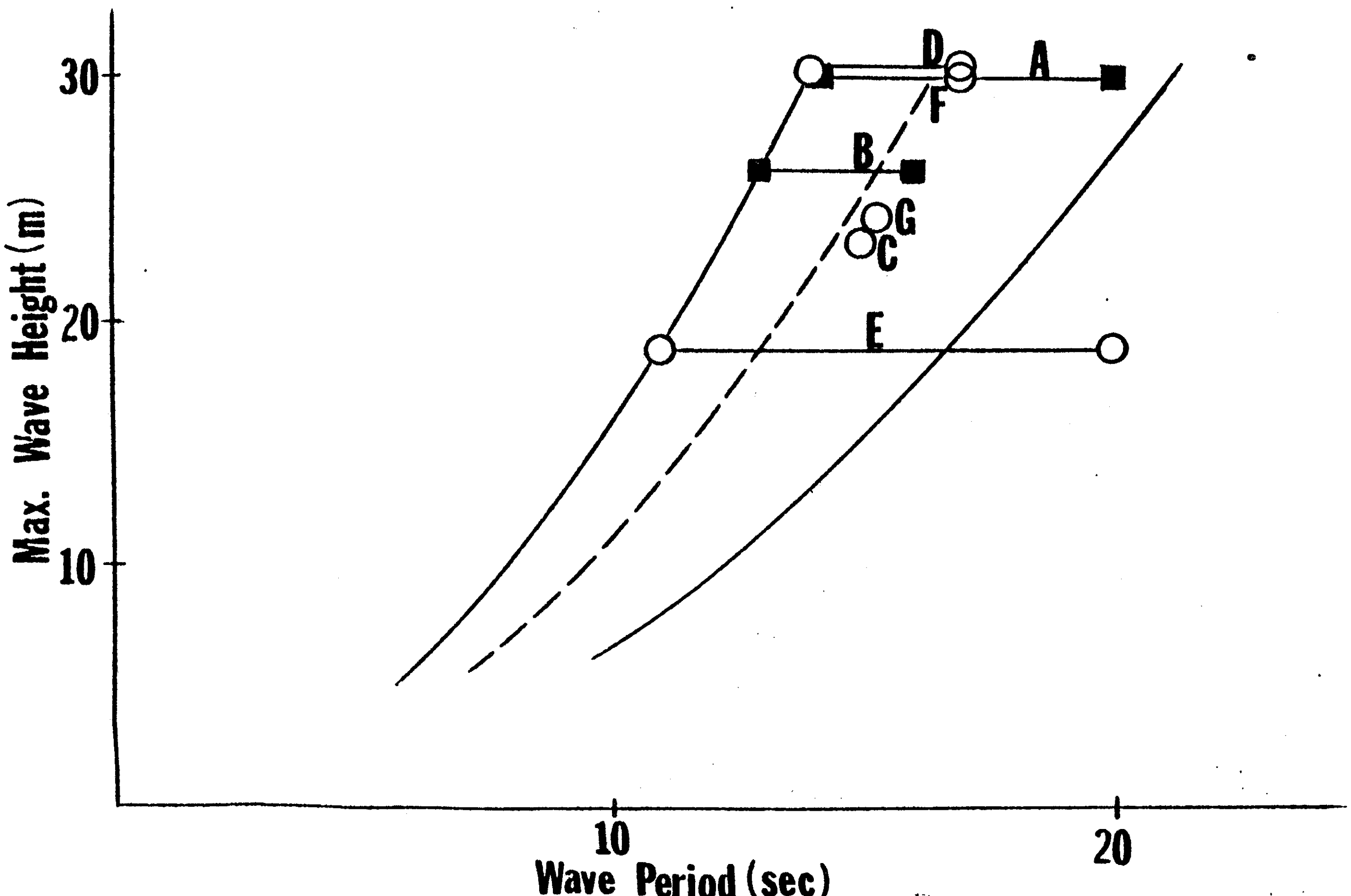


Table 7-1

Data Mark	H_{max}	Period	Type	References
A	30	14-20	TLP	11,58,59
B	26	13-16	TLP	6,7,8,9,10
C	23	15	Semi	38,39
D	30	14-17		33
E	18.5	11-20		46
F	30	17		49
G	24	15.5	Semi	51

7.3 Stability

The TLP must maintain positive stability at two different conditions. One is towing-out condition, the other is operating condition when one anchor is totally lost. Although the second case is unlikely, nonetheless the TLP must be designed for the worst possible case. Also the second condition guarantees an easy installation of the TLP without the aid of additional buoyancy tanks.

(1) Model 1

a. Towing out condition

From Eq.6-12 the light weight can be calculated.

$$\Delta_1 = \nabla_1 \rho = 4\pi\left[R^2H + r^2\{(f+d)-H\}\right]*0.23 + W_1$$

$$W_1 = 0.74LB + cP_c + (93.3+0.44L_{wd})\Delta/3{,}000$$

The towing draft is given by the following equations.

$$d_0 = H+b = H + (\nabla_1 - 4\pi R^2H)/4\pi r^2 \qquad \text{when } \nabla_1 > 4\pi R^2H$$

$$d_0 = H+b = \nabla_1/4\pi R^2 \qquad \text{when } \nabla_1 < 4\pi R^2H$$

KG,KB,BM and GM can be calculated by the following equations;

$$\Delta_1 * KG = 0.23*2\pi\left[R^2H^2 + r^2\{(f+d)^2 - H^2\}\right] + W_1(d+f+h_d/2)$$

$$\nabla_1 * KB = 2\pi\left[R^2H^2 + 2r^2b(b/2+H)\right] \qquad \text{when } b > 0$$

$$= (H+b)/2 \qquad \text{when } b < 0$$

$\nabla_1 * BM = \pi r^2 (l^2 + r^2)$ when $b \geqq -0.5$

$= \pi R^2 (l^2 + R^2)$ when $b < -0.5$

$l = B - 2r - 2$

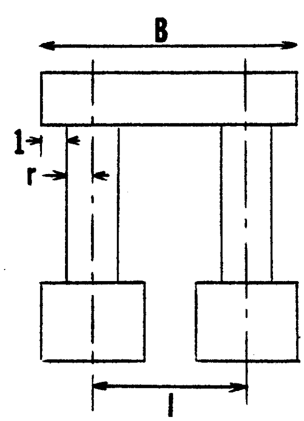
l; Column center spacing

For a very small negative value of b, BM has a very high value at upright position, yet it decreases drastically when the TLP is tilted even slightly. To avoid this problem and make sure that we are on the conservative side, the threshold value of b is set somewhat arbitrarily at -0.5m.

Finally GM is calculated and checked by the following relation:

$GM = KB + BM - KG \geqq 0.1$

The minimum value of GM, 0.1m, is arbitrarily chosen to make the design conservative.



b. Operating condition when one anchor is totally lost.

The situation can be modeled as shown in the figure below. The moment around point K is

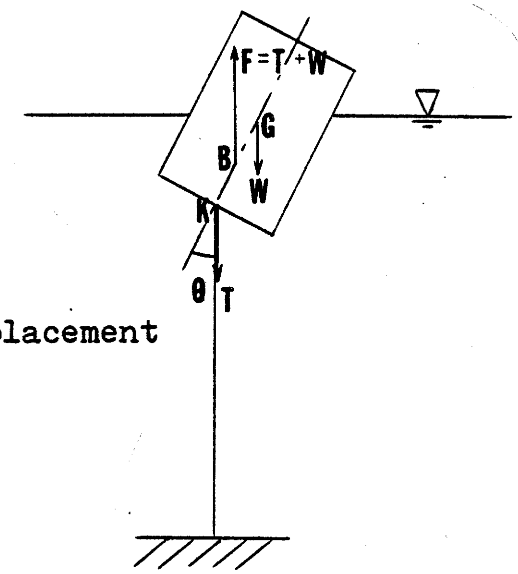
$$KB*F\sin\theta - KG*W\sin\theta + BM*F\sin\theta$$

$$=\{(BM+KB)F/W-KG\}W\sin\theta$$

$$=GM'W\sin\theta$$

$$GM'=(BM+KB)F/W-KG$$

$$=BM'+KB'-KG'$$



In this case F is the total displacement Δ, and W the light weight Δ_1,

$$BM'=\pi(l^2+r^2)r^2/\nabla_1$$

$$KB'=2\pi\{r^2(d^2-H^2)+R^2H^2\}/\nabla_1$$

$$KG'=KG=[2\pi*0.23(R^2H^2+r^2\{(f+d)^2-H^2\})+W_1(d+f+h_d/2)]/\Delta_1$$

$$GM'=KB'+BM'-KG'\geq 0.1$$

(2) Model 2

a. Towing out condition

From Eq.6-12 the light weight can be calculated:

$$\Delta_1=0.23(\nabla+S_a f)+W_1$$

$$W_1=0.74LB+cP_c+(93.3+0.44L_{wd})\Delta/3{,}000$$

The towing draft is calculated by the following equations:

$$d_0=P_h+b=(\nabla_1-2P_hP_bL)/S_a+P_h \qquad \text{when } \nabla_1>2P_hP_bL$$

$$d_0=P_h+b=\nabla_1/(2P_bL) \qquad \text{when } \nabla_1<2P_hP_bL$$

KG, KB, BM and GM can be calculated by the following equations:

$$\Delta_1 * KG = 0.23[LP_b P_h^2 + 0.5 S_a (d+f+P_h)(d+f-P_h)] + W_1 (d+f+h_d/2)$$

$$\nabla 1 * KB = S_a b(P_h + b/2) + LP_b P_h^2 \qquad \text{when } b > 0$$

$$KB = (P_h + b)/2 \qquad \text{when } b < 0$$

Next radii of corner column and intermediate column must be calculated first to compute BM. Assuming 6 columns we can calculate for simplicity the corner column spacing by the following equations.

$$r_1^2 = S_a/6$$

$$l_1 = L-2-2r_1$$

$$l_2 = B-2-2r_1$$

Also assuming that the TLP has the same stability in both directions (length and width), the radii can be computed by the following equations

$$r^2 = S_a l_2^2/4/l_1^2$$

$$r'^2 = S_a/2-2r^2$$

r; Corner column radius

r'; Intermediate column radius

Then BM can be computed by the following equations.

$$\nabla_1 * BM = S_a (l_2/2)^2 + r^4 + r'^4/2$$

$$= S_a l_2^2/4 + (r^4 + 0.5r'^4) \qquad \text{when } b > -0.5$$

$$\nabla_1 * BM = L\ B^3 - (B-2P_b)^3\ /12 \qquad \text{when } b \leq -0.5$$

Finally GM is calculated and checked by the following relation:

$$GM=KB+BM-KG \geq 0.1$$

b. Operating condition when one anchor is totally lost.

Similarly in the case of model 1

$$\nabla_1 * BM' = S_a l_2^2/4 + \pi(r^4 + 0.5r'^4)$$

$$\nabla_1 * KB' = S_a(d-P_h)(d+P_h)/2 + P_h^2 P_b L$$

$$KG' = KG = \{0.23[LP_b P_h^2 + 0.5S_a(d+f+P_h)(d+f-P_h)] + W_1(d+f+h_d/2)\}/\Delta_1$$

$$GM' = KB' + BM' - KG' \geq 0.1$$

7.4 Dynamic Excursion

The dynamic motion of the TLP is expressed by the following equation:

$$M\ddot{x}+R\dot{x}+Kx=F(t) \qquad \text{(Eq.7-9)}$$

M; Mass $(=\Delta_1+C_a\nabla\rho \approx \Delta_1+\Delta)$

R; Damping coefficient $(=2\sqrt{KM}$, $\zeta=R/2\omega_N M$, $\omega_N{}^2=K/M)$

for typical offshore structure $\zeta=0.01\sim0.03$

K; Spring constant $(=gT_0/(L_{wd}-d))$

F(t); External surging force

$$F(t)=\int(1+C_a)\rho\frac{du}{dt}d\nabla+0.5\rho AC_D u|u|$$

Assuming that the inertia force is far greater than the drag force,

$$F(t)\approx 2\rho\frac{du}{dt}\int d\nabla=2\rho\nabla\cos\frac{kl}{2}\,\frac{du}{dt}$$

x; Horizontal excursion of the TLP

$$x_{max}\;\frac{2\nabla\rho(a\omega^2)\cos\frac{kl}{2}e^{-k\frac{d}{2}}}{\left[\left\{\frac{T_0 g}{L_{wd}-d}-\omega^2(\Delta_1+\Delta)\right\}^2+4\frac{gT_0}{L_{wd}-d}(\Delta_1+\Delta)\zeta^2\omega^2\right]^{1/2}}$$

(Eq.7-10)

k; Wave number $(=\omega^2/g=2\pi/\lambda)$

T_0; Pretension

a; Wave amplitude

7.5 Static Force

(1) Wind Force

API code(66) requires to calculate the wind force by the following formula.

$$F_w = 0.0473 v_w C_s A$$

F_w; Wind force (N)

v_w; Wind velocity (km/h)

A; Area of object (m^2)

C_s; Shape coefficient

Beams--------------------------------------1.5

Sides of Building-----------------------1.5

Cylindrical sections--------------------0.5

Overall projected area of platform------1.0

API also recommends the following wind velocity profile to consider the variation of wind velocity with height to compute the wind force.

$$\left(\frac{V_y}{V_H}\right) = \left(\frac{y}{H}\right)^{1/n}$$

V_y; The wind velocity at height y

V_H; The wind velocity at a reference height H usually 10m above a reference water depth.

1/n; An exponent, usually assumed to be between 1/13 and 1/7

For both model 1 and model 2 the wind force on the TLP is given by the following equation.

$$F_w=0.06255v_w^2(1.5Lh_dC_1+4frC_2)$$

F_w; Wind force (kg)

v_w; Wind velocity (m/s)

L; Deck length (m)

h_d; Deck height (m)

f; Freeboard (m)

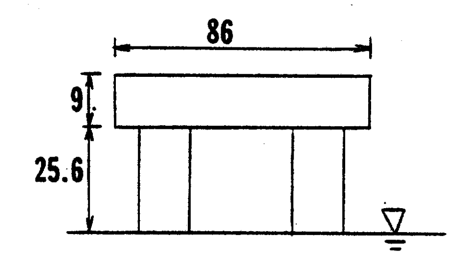
r; Radius of column (m)

C_1,C_2; Height coefficients

$C_1=(\frac{f+h_d/2}{10})^{1/n}$, $C_2=\int_0^f(\frac{y}{10})^{1/n}dy$

For example, C_1 and C_2 are computed for the Aker design.

n	C_1	C_2
13	1.08	0.99
7	1.17	1.01



Setting C_1=1.1, C_2=1.0,

the equation above becomes

$$F_w=0.06255v_w(1.65Lh_d+4fr) \qquad \text{(Eq.7-11)}$$

(2) Current Force

API(66) recommends the following method to compute the current force.

$$F_c=0.5C_D\rho v_c^2A$$

F_c; Current force per unit length (N/m)

C_D; Drag coefficient

ρ ; Mass density (kg/m^3)

v_c; Current velocity (m/s)

A; Projected area per unit length

API also recommends a current velocity profile to consider the bottom effect, but, generally, the TLP is installed in sufficiently deep water to ignore this effect.

a. Model 1

For cylinder $C_D \approx 1.0$

$$F_c=0.4188v_c^2[r(d-H)+RH] \qquad \text{(Eq.7-12)}$$

F_c; Current force (kg)

v_c; Current velocity (m/s)

r; Column radius (m)

d; Draft (m)

H; Height of enlarged section

R; Radius of enlarged section

b. Model 2

For 2-dimensional rectangular section, $C_D \approx 2.0$

$$F_c = 0.4188 v_c^2 \left[r(d-P_h) + 0.5 P_h L \right] \qquad \text{(Eq.7-13)}$$

P_h; Lower hull height

7.6 Static Excursion

Static excursion at the equilibrium point is calculated by the following equations.

$$x_{stat}=(L_{wd}-d)\sin\theta \qquad \text{(Eq.7-14A)}$$

$$T_h=T_v\tan\theta \qquad \text{(Eq.7-14B)}$$

$$T_v=T_0+4\pi\dot{r}^2(1-\cos\ )(L_{wd}-d)\rho \qquad \text{(Eq.7-14C)}$$

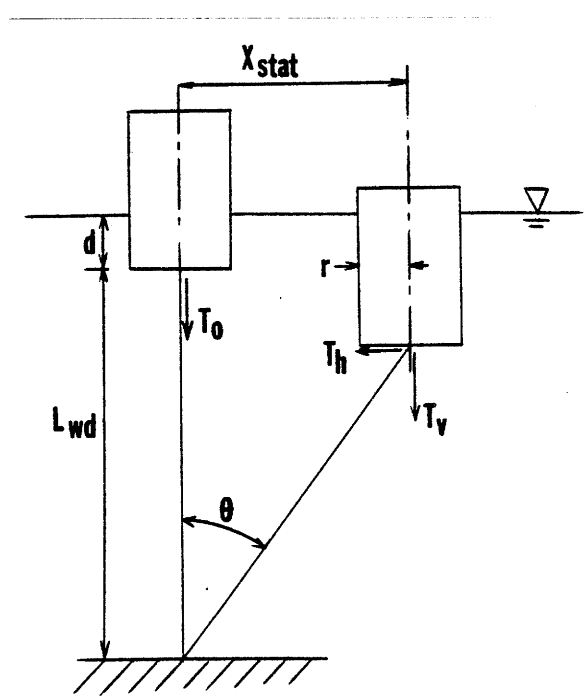
x_{stat}; Static excursion

θ; Angle between tension line and vertical line

T_0; Pretension

T_h; Horizontal component of the mooring tension

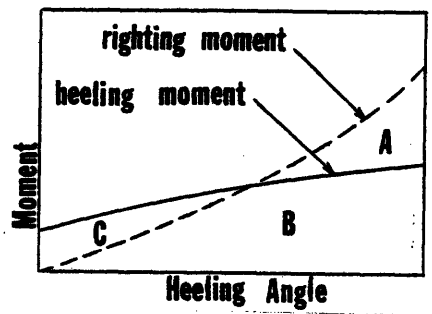
T_v; Vertical component of the mooring tension



7.7 Dynamic Stability

ABS rules(65) require that all units are to have sufficient stability to withstand the overturning effect of the force produced by a steady wind of 100 knots from any horizontal direction. In other words, the area under the righting curve at an angle corresponding to submergence of the deck edge is not to be less than 30% in excess of the area under the wind heeling moment curve to the same limiting angle.

Area(A+B)≥1.3*Area(B+C)



Righting moment can be calculated by the following equation.

$$M_r = \Delta * BM\sin\phi \frac{1+\sec^2\phi}{2} - \Delta(KG-KB)\sin\phi \qquad \text{(Eq.7-15)}$$

M_r; Righting moment

ϕ; Heeling angle

Heeling moment can be computed by the following equation.

$$M_h = [1.65(h_d\cos\phi + B\sin\phi)L(d+f+h_d/2-KG)\cos\phi$$
$$+4r\{f(d+f/2-KG)\cos^2\phi + l^2/8*\sin^2\phi\}]$$
$$*0.06255v_w^2 \qquad \text{(Eq.7-16)}$$

The derivation of these formulas is given in Appendix I. In the computer program, M_r and M_h are calculated from $\phi = 0^\circ$ to 30° by 5° increment.

7.8 Pitching Moment

(1) Moment Due to Wave

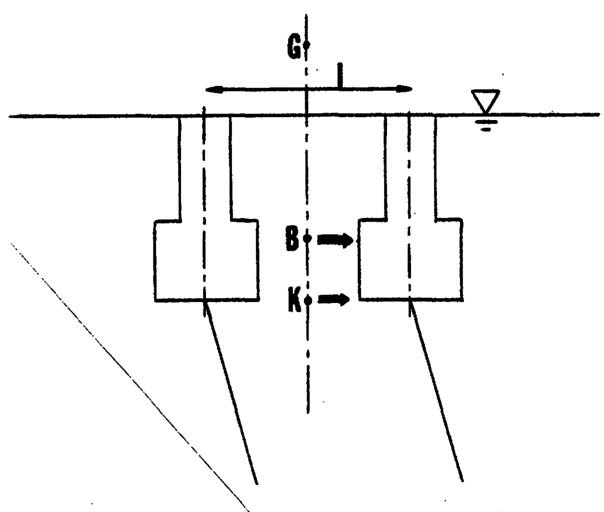
The response of the TLP to the wave is given by the following relation.

$$(\Delta_1 + \Delta)\ddot{x} + R\dot{x} + T_0 g/(L_{wd} - d)x = F(t)$$

The actual force exerted on the TLP is found as the sum of the first and the fourth terms of the left hand side. The pitching moment around G is given by the following equation.

$$M_p = \Delta_1 \ddot{x}/g * BG + T_0/(L_{wd} - d)x * KG$$

$$= \Delta_1 \, x_{max} \frac{\omega^2 * BG}{g} + T_0/(L_{wd} - d)|x_{max}| * KG \qquad \text{(Eq.7-17)}$$



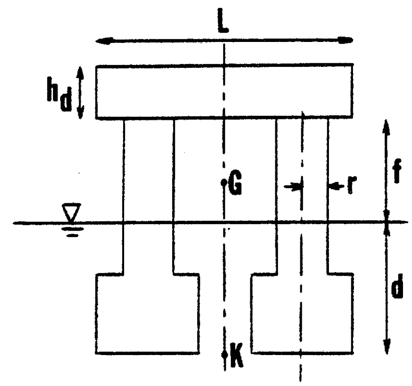
(2) Moment Due to Wind and Current

a. Moment due to wind

For both models the pitching moment around G due to wind is given by the following equation.

$$M_w=0.06255v_w^2[1.65h_dL(f+d+h_d/2)+4fr(d+f/2)]-F_w*KG$$
$$=0.06255v_w^2[1.65h_dL(f+d+h_d/2-KG)+4rf(d+f/2-KG)]$$

(Eq.7-18)



b. Moment due to current

For model 1 the pitching moment around G due to current is given by the following equation.

$$M_c=0.41878[r(d-H)(KG-H-d/2)+RH(KG-H/2)]v_c^2$$

(Eq.7-19A)

For model 2 the pitching moment around G due to current is given by the following equation.

$$M_c=0.41878[r(d-P_h)(KG-P_h+d/2)+0.5*P_hL(KG-P_h/2)]v_c^2$$

(Eq.7-19B)

7.9 Natural Frequencies

(1) Surge and Sway

Surge and sway motion of the TLP is expressed by the following equation.

$$(\Delta+\Delta_1)\ddot{x}+R\dot{x}+T_0 gx/(L_{wd}-d)=F(t)$$

As $T_0=\Delta-\Delta_1$

$$\omega_{ss}^2=T_0 g/(l_{wd}-d)/(\Delta+\Delta_1)=\frac{\Delta-\Delta_l}{\Delta+\Delta_l}\frac{g}{L_{wd}-d}$$

$$\omega_{ss}=\sqrt{\frac{\Delta-\Delta_l}{\Delta+\Delta_l}\frac{g}{L_{wd}-d}} \qquad \text{(Eq.7-20)}$$

(2) Heave

a. Model 1

The heave motion of the TLP is given by the following equation.

$$(\Delta_1+16\pi R^3\rho/3)\ddot{z}+R_z\dot{z}+(EA_T g/(L_{wd}-d)+4\pi r^2\rho g)z=F_z(t)$$

$$\omega_h^2=\frac{EA_T g/(L_{wd}-d)+4\pi r^2\rho g}{\Delta_1+16\pi R^3\rho/3}$$

$$\omega_h=\sqrt{\frac{EA_T g/(L_{wd}-d)+4\pi r^2\rho g}{\Delta_1+16\pi R^3\rho/3}} \qquad \text{(Eq.7-21)}$$

E; Young's modulus of elasticity

A_T; Cross sectional area of the mooring line

b. Model 2

The heave motion of this model is given by the following equation.

$$(\Delta_1+2P_b^2L\rho)\ddot{z}+R_z\dot{z}+(\frac{EA_T}{L_{wd}-d}g+S_a\rho g)z=F_z(t)$$

$$\omega_h^2=\frac{EA_Tg/(L_{wd}-d)+S_a\rho g}{\Delta_1+2P_b^2L\rho}$$

$$\omega_h=\sqrt{\frac{EA_Tg/(L_{wd}-d)+S_a\rho g}{\Delta_1+2P_b^2L\rho}} \qquad \text{(Eq.7-22)}$$

(3) Pitch and Roll

Pitch and roll motion of the TLP is given by the following equation.

$$(I_\theta+i_\theta)\ddot{\theta}+R_\theta\dot{\theta}+g\left\{\frac{EA_Tl^2}{4(L_{wd}-d)}+A_s\rho(l/2)^2\right\}\theta=M_\theta(t)$$

$$\omega_{pr}^2=\frac{EA_Tl^2/(L_{wd}-d)/4+A_s\rho(l/2)^2}{I_\theta+i_\theta}g$$

To be conservative, assuming that most of the mass is concentrated at the columns, we require that:

$$i_\theta \leq \Delta(l/2)^2$$

$$I_\theta \leq \Delta_1(l/2)^2$$

Thus, for model 1

$$\omega_{pr}\geq\sqrt{\frac{g\{EA_T/(L_{wd}-d)+4\pi r^2\rho\}}{(\Delta+\Delta_1)}} \qquad \text{(Eq.7-23)}$$

Usually ω_{rp} is very large, thus if the minimum possible ω_{pr} is above the range of wave frequencies, it is assured that the pitch and roll motion causes no significant dynamic problem.

For model 2

$$\omega_{pr} \geq \sqrt{\frac{g\{EA_T/(L_{wd}-d)+S_a\rho\}}{(\Delta+\Delta_1)}} \qquad \text{(Eq.7-24)}$$

(4) Yaw

The yaw motion of the TLP is given by the following equation.

$$(I_\psi+i_\psi)\ddot{\psi}+R_\psi\dot{\psi}+4T_0/(L_{wd}-d)(1/\sqrt{2})^2\psi=M\ (t)$$

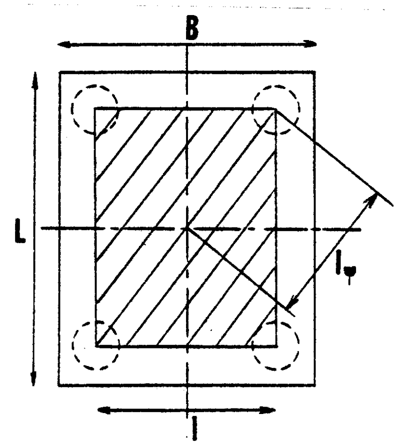
$$\omega_y^2=\frac{2T_0 l^2/(L_{wd}-d)}{I_\psi+i_\psi}$$

To be conservative, assuming that the mass is distributed evenly, all over the shaded area shown below, we require that:

$$i_\psi \geq \Delta/3*l_\psi^2 \geq \Delta/6*l^2$$

$$I_\psi \geq \Delta_1/3*l_\psi^2 \geq \Delta_1/6*l^2$$

$$\omega_y \leq \sqrt{\frac{12T_0}{(L_{wd}-d)(\Delta+\Delta_1)}}$$



(Eq.7-25)

CHAPTER VIII

SENSITIVITY ANALYSIS

The computer calculation was carried out on both model 1 and model 2. 4 proposed designs, Conoco, B.P., Amoco, and Aker are chosen for the sensitivity analysis of displacement, deck size and draft. Appendix III gives some typical results of this analysis. Sensitivity analysis of other parameters is carried out only on the Aker design of model 1.

8.1 Displacement and Deck size

Fig.8-1 through Fig.8-4 shows the summary of these sensitivity analysis. These figures clearly show the following:

(1) When the dynamic effect is relatively large the horizontal excursion decreases as displacement decreases or deck size increases. The Conoco design of model 1 is the typical example.

(2) When the static effect is relatively large the horizontal excursion decreases as displacement increases or deck size decreases. B. P. and Amoco designs are typical example of this.

For model 1 the following quantity can be used to check the relative importance of static effects against dynamic effects.

$$M_1 = \frac{(v_c^2 + v_w^2/1{,}000)L_{wd}}{H_w} \qquad \text{(Eq.8-1)}$$

For $M_1 < 20$, the dynamic effects are dominant.

For $M_2 > 20$, the static effects are dominant.

For model 2, since current is more important than for model 1, the following quality can be used to check the relative importance of static effects against dynamic effects.

$$M_2 = \frac{(2v_c^2 + v_w^2/1{,}000)L_{wd}}{H_w} \qquad \text{(Eq.8-2)}$$

For $M_2 < 20$ the dynamic effects are dominant.

For $M_2 > 20$ the static effects are dominant.

The actual numbers calculated for 4 proposed designs are listed below.

	Conoco	Aker	Amoco	B.P.
M_1	15.6	24.8	118.5	31.7
M_2	21.7	33.9	191.4	46.7

Comparing these values with Fig.8-1 through 8-4, it is concluded that although these qualities are very approximate, they are generally reliable and useful, especially at relatively shallow water, where, the horizontal excursion is most critical.

The explanation of these results is as follows: As displacement decreases, pretention also decreases and this leads to increased static excursions and decreased dynamic excursions.

As the deck size increases, wind force increases and leads to increased static excursions. An increase of the deck size also causes a decreased dynamic excursion, because, the response lag of the columns in the wave propagation direction increases. Therefore it depends heavily on the relative importance of static versus dynamic effects, whether the change of displacement or deck size will really increase or decrease the total horizontal excursion.

8.2 Positive Tension Restriction

A table below shows some values of Δ_1/Δ. The left column shows values of Δ_1/Δ with positive tension and the right column values of Δ_1/Δ with negative tension. From this table it is concluded that approximately 0.75 is the threshold value. Positive tension restriction becomes more critical when water depth increases.

Case	Positive Tension	Negative Tension
Conoco	0.713	0.730
Aker	0.741	0.766
B.P.	0.743	0.791
≀	0.728	0.768
B.P.	0.713	0.730

8.3 Draft

The influence of the draft on the horizontal excursion is very complicated. The major influences are summarized below:

As the draft increases,

(1) The wind force decreases as column diameter decreases, and thus, the static excursion decreases.

(2) The current force increases, because projected area increases and thus the static excursion increases.

(3) The exponential decay of the fluid particle motion increases and the dynamic excursion decreases.

(4) Pretention increases because the column diameter decreases and so does the jacket weight. Thus the static excursion decreases.

(5) Increased pretension makes the mooring system more stiff and leads to an increased dynamic excursion due to increased dynamic amplification.

Fig.8-5 and 8-6 show the result of this sensitivity analysis. For model 1 (3) or (4) dominate the behavior, but for model 2 the current effect becomes more important, and (2) dominates the behavior of the Amoco and B.P. designs.

8.4 Natural Period

(1) Surge and Yaw

In all 4 cases $T \approx 40 \sim 50$sec for relatively shallow water(150~300m) and they increase with increased water depths, so no problems are expected from wave excitation.

(2) Heave and Pitch

In all 4 cases $T \leq 2.5$sec for relatively shallow water and it is very sensitive to the change of ultimate strength of the mooring lines and the factor of safty used. So the estimate of ultimate strength and factor of safty must be done very carefully. An increase in water depth also increases natural periods. So in deep water, natural frequencies of heave and pitch will be very critical.

8.5 Dynamic Stability

Some cases with small GM or GM' are shown in Fig.8-7 and 8-8. In all cases, it is demonstrated that the TLP with GM or GM' of at least 0.1m has sufficient restoring moment.

8.6 Water Depth

Fig.8-9 summarizes the sensitivity analysis for the water depth. Fig.8-9A shows the minimum displacement necessary to keep mooring tension positive versus water depth. Fig.8-9B shows natural heave periods versus water depth. This analysis was carried out on the Aker design of model 1. One important observation is that beyond 600m the displacement increases exponentially. This is probably due to the assumption that the weight of the riser and mooring system is proportional to not only water depth but also displacement. This is absolutely true for mooring risers, but not quite for production risers. Another reasonable assumption is that the weight of production risers is proportional only to water depth and is independent of displacement. Thus the equipment weight is expressed as follows.

$$W_r = L_{wd}\Delta/1{,}000 + 9L_{wd}(1 + cP_c/1{,}000) \qquad \text{(Eq.8-3)}$$

The results of a similar sensitivity analysis based on this assumption are shown in Fig.8-10. The results indicate that displacement increases exponentially as before, but less rapidly. In spite of the different assumptions, natural heave periods show almost the same results. This analysis indicates that the practical limit of the TLP is at most 1,000m, where the natural heave period approaches 5 seconds and displacement becomes too large(about 80,000tons).

The actual limit must be between 1,000m and 600m but due to the limited data available, more practical prediction was not possible. For more accurate computation, more data on the weight of the riser and mooring system is necessary.

8.7 Other Results

(1) Wave Height

As wave height increases, the dynamic excursion and dynamic variation increase. It has no significant effect on stability.

(2) Freeboard

As freeboard increases, static excursion increases, while dynamic excursion decreases, thus the total excursion may increase or decrease. Pretention decreases while dynamic tension variation increases, thus the positive tension restriction becomes more important.

(3) Deck Height

As deck height increases, static excursion increses and so does dynamic tension variation. An increase of deck height also has an adverse effect on stability.

(4) Wind Velocity

As wind velocity increases, static excursion increases, while dynamic tension variation also increses but slightly.

(5) Current Velocity

As current velocity increases, static excursion increases.

(6) Production Equipment Weight

As production equipment weight increases, pretention decreases and dynamic tension variation increases thus the positive tension restriction becomes more important. An increase of production equipment weight has also adverse effect on stability, however it may decrease the total horizontal excursion.

8.8 Horizontal Excursion

The horizontal excursion can be calculated approximately by the following equations.

$$x_{dyn} \cong \frac{0.725\Delta}{(\Delta_1 + \Delta)/H_w - (\Delta - \Delta_1)*2.25/(L_{wd} - d)} \qquad \text{(Eq.8-4)}$$

$$x_{stat} \cong \frac{(77+1.15\,\Delta/1,000)(v_c{}^2 + v_w{}^2/1,000)}{(\Delta - \Delta_1)}(L_{wd} - 30)$$

(Eq.8-5)

Table 8-1 compares these approximate values with predicted values by the computer program. They agree generally well.

Fig 8-1 Displacement vs Horizontal Excursion(Model- I)

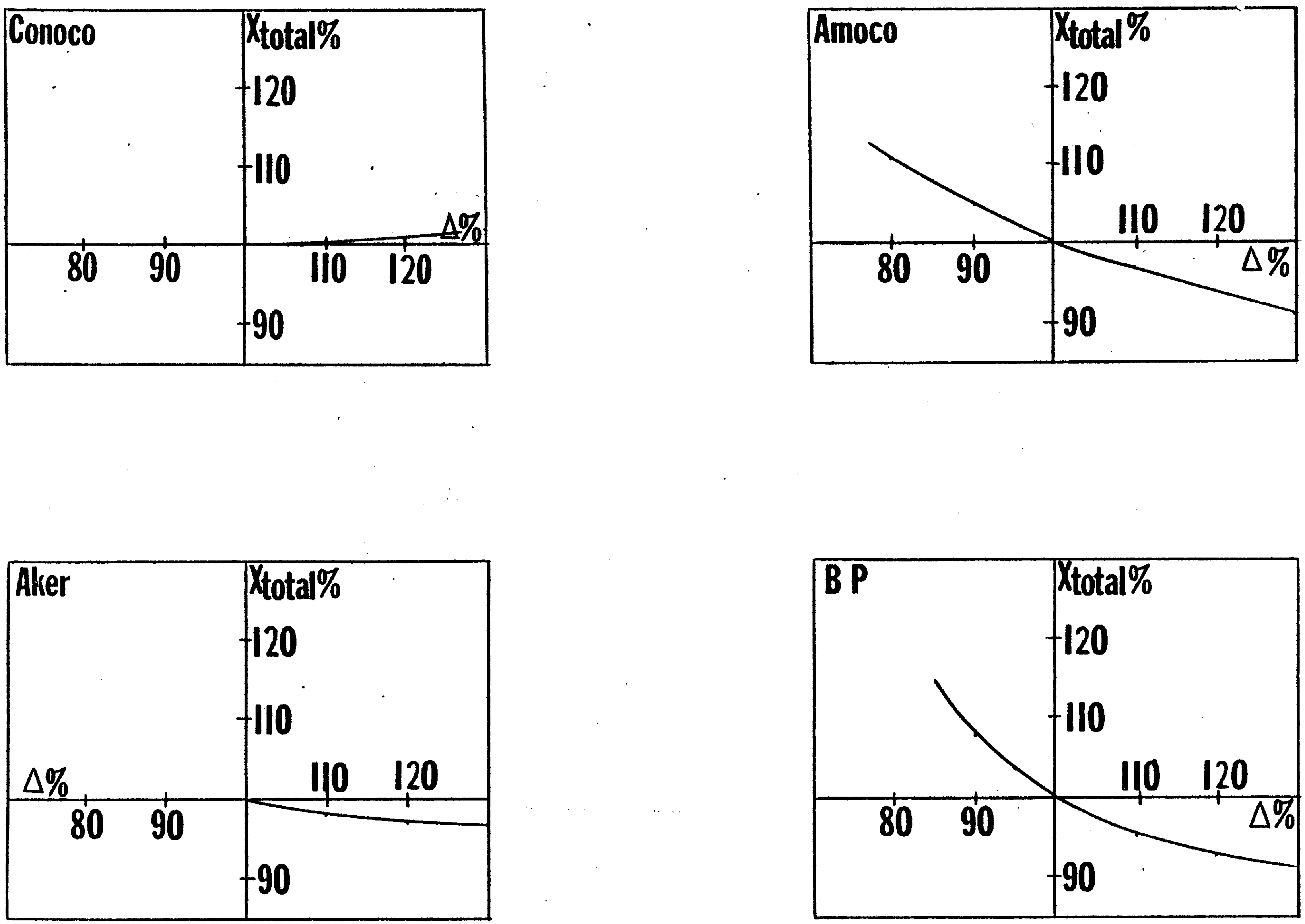


Fig 8-2 Deck Size vs Horizontal Excursion(Model-I)

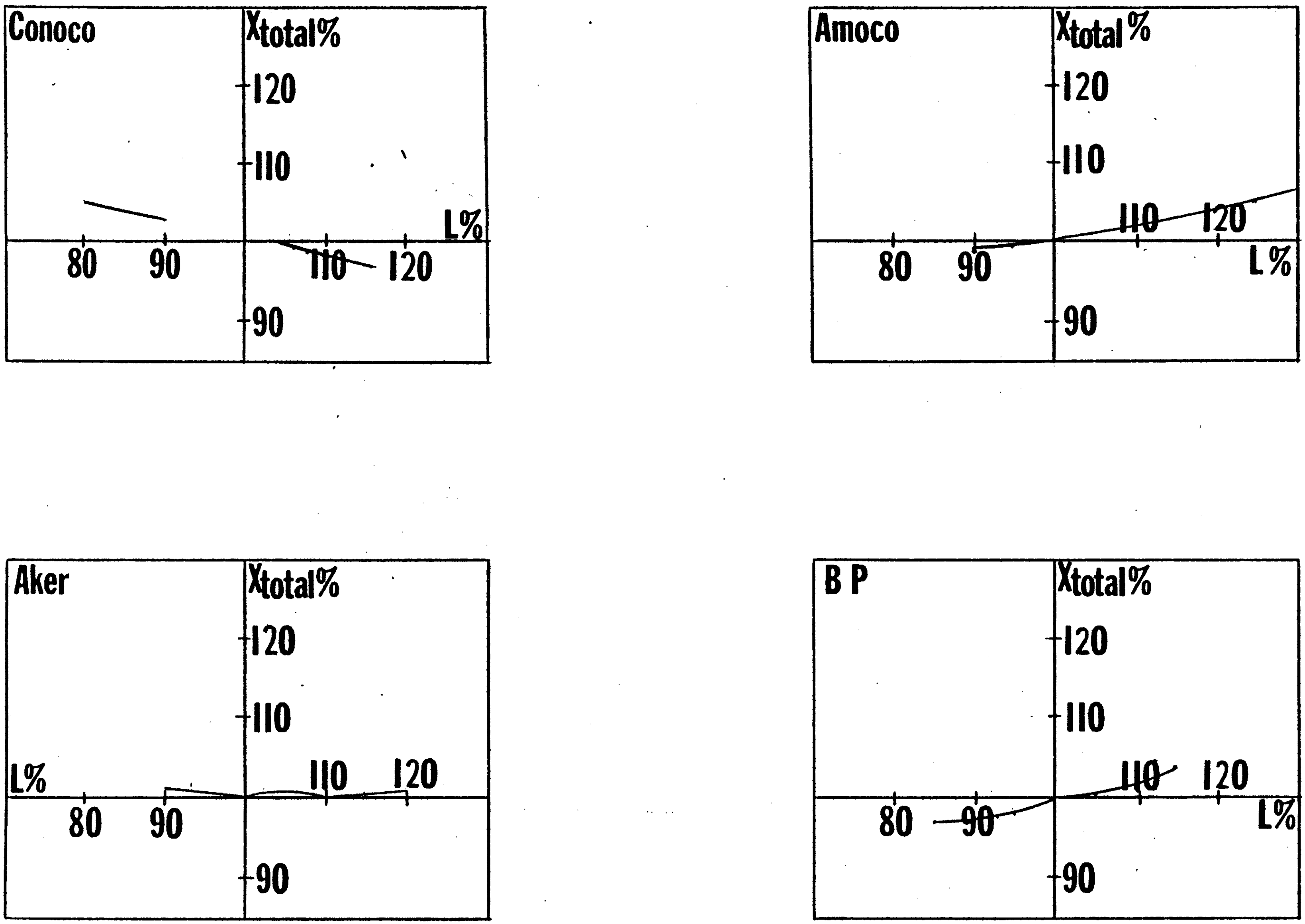


Fig 8-3 Displacement vs Horizontal Excursion(Model- 2)

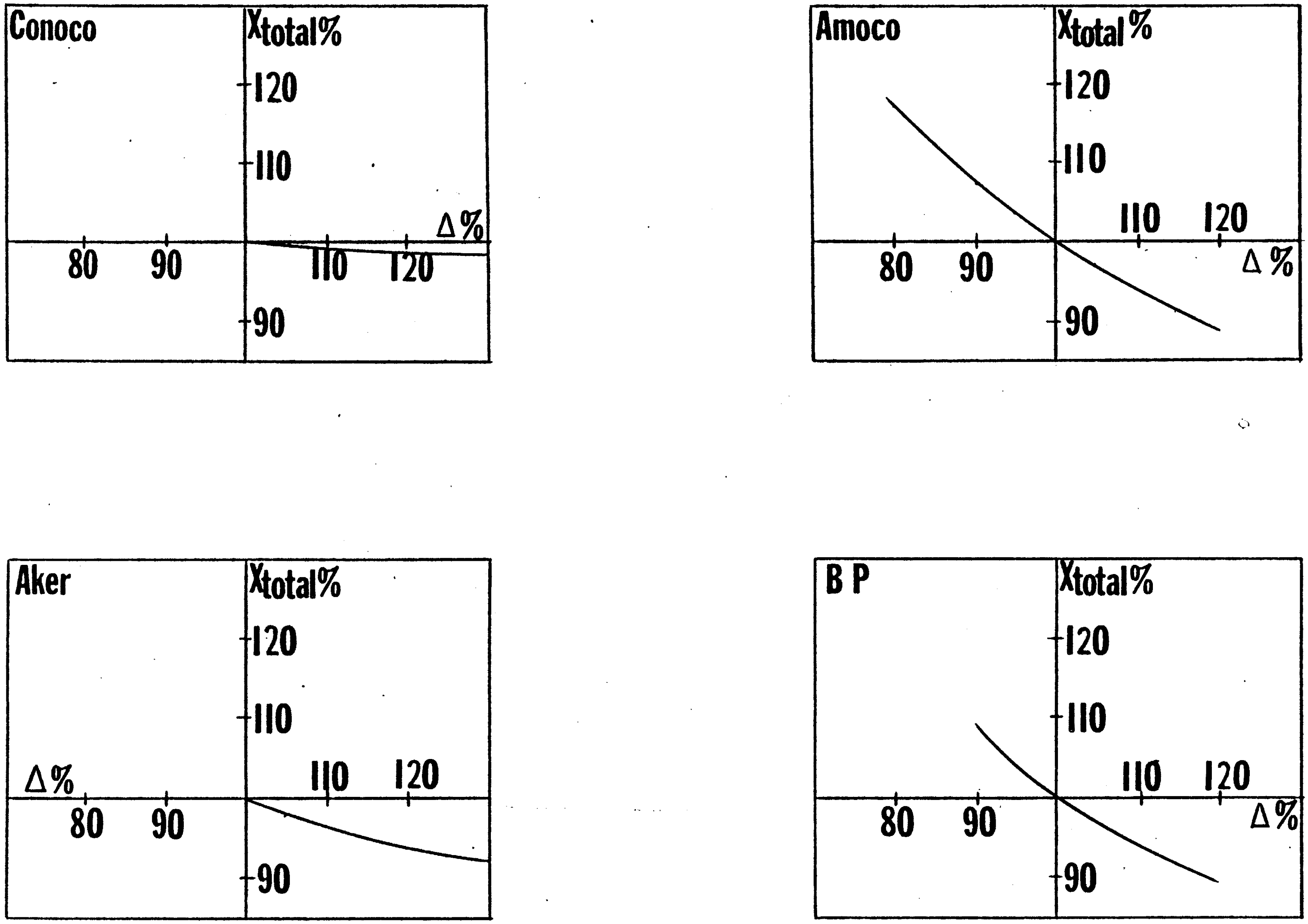


Fig 8-4 Deck Size vs Horizontal Excursion(Model- 2)

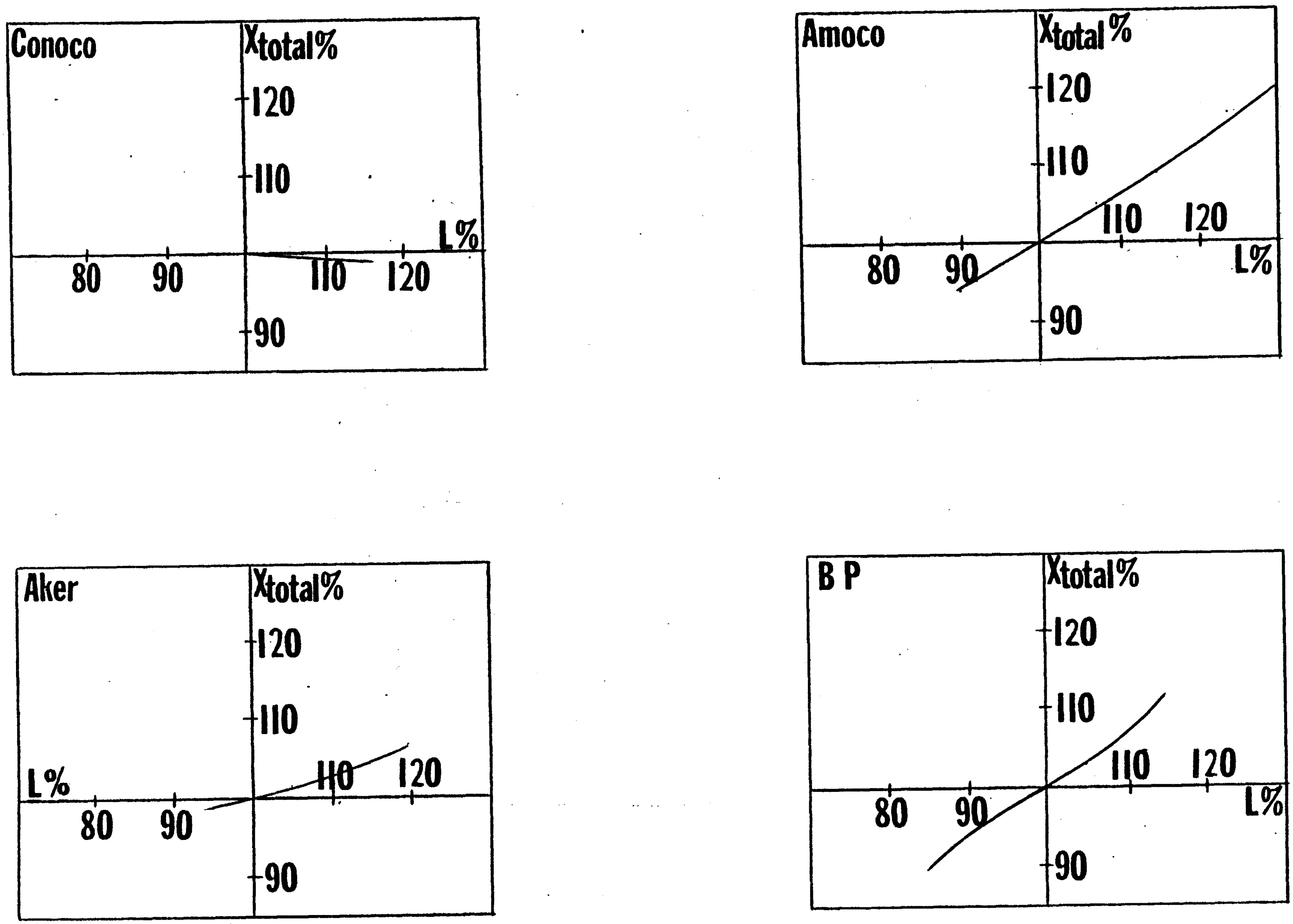


Fig 8-5 Draft vs Horizontal Excursion(Model- I)

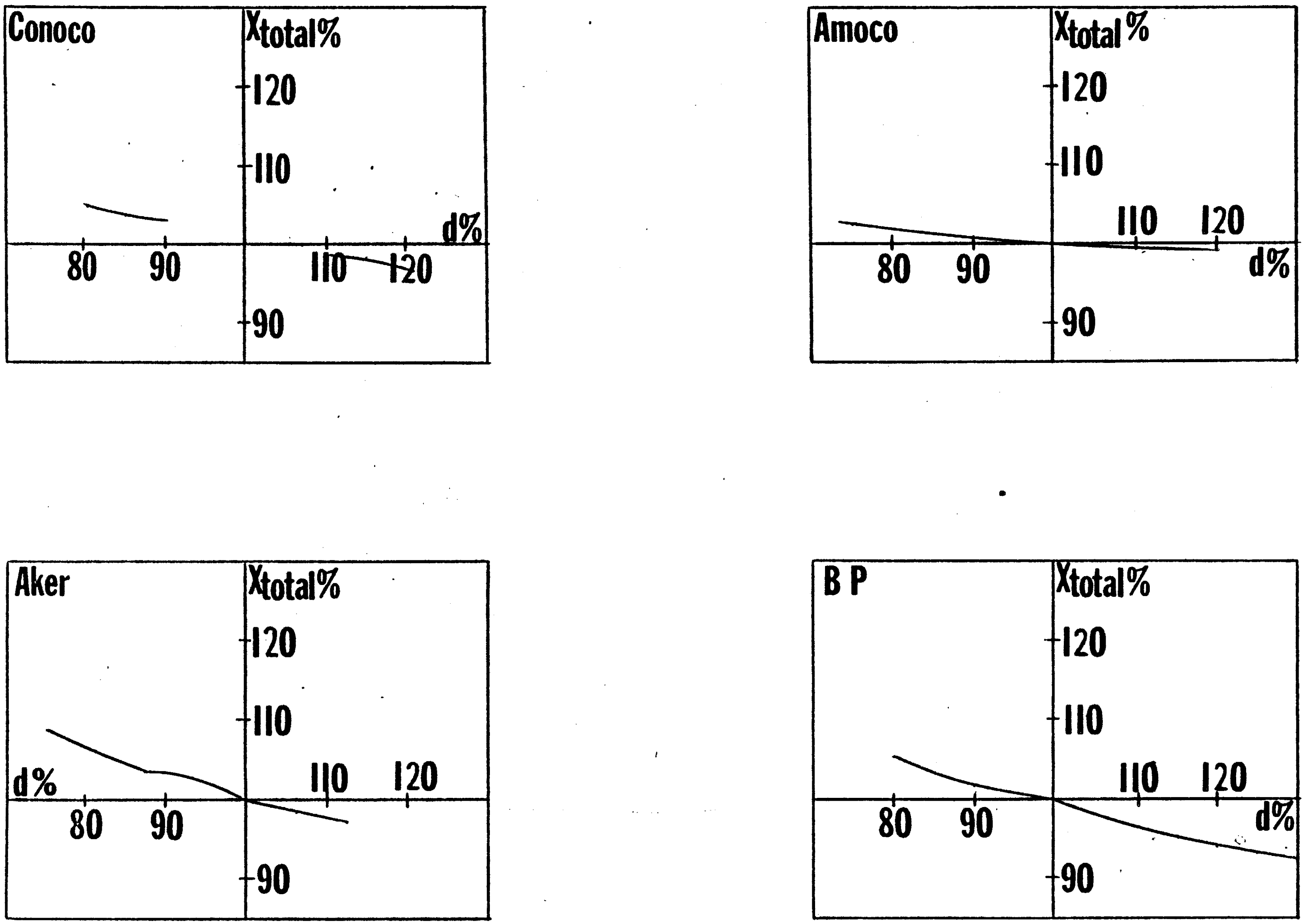


Fig 8-6 Draft vs Horizontal Excursion(Model-2)

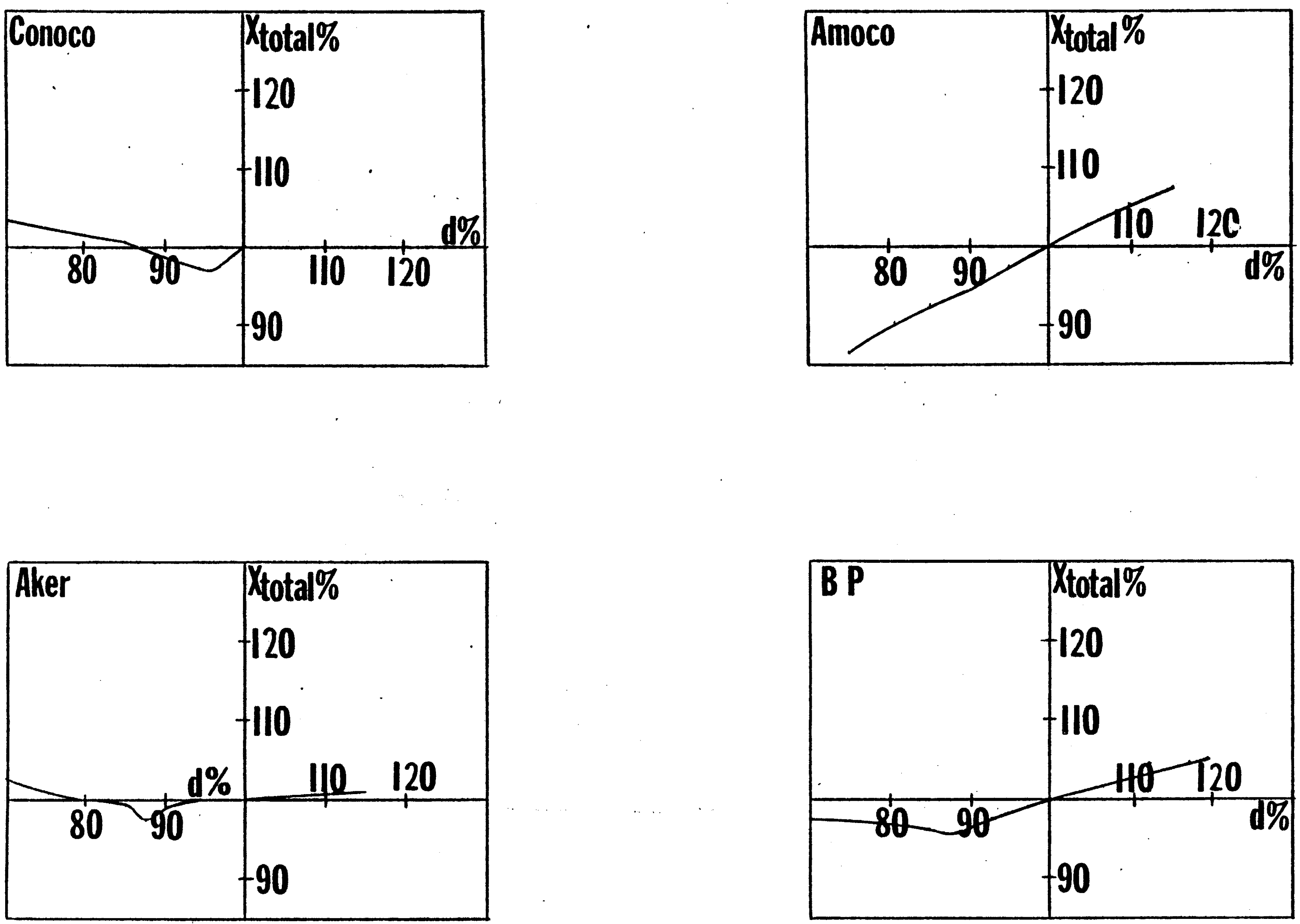


Fig 8-7 Dynamic Stability

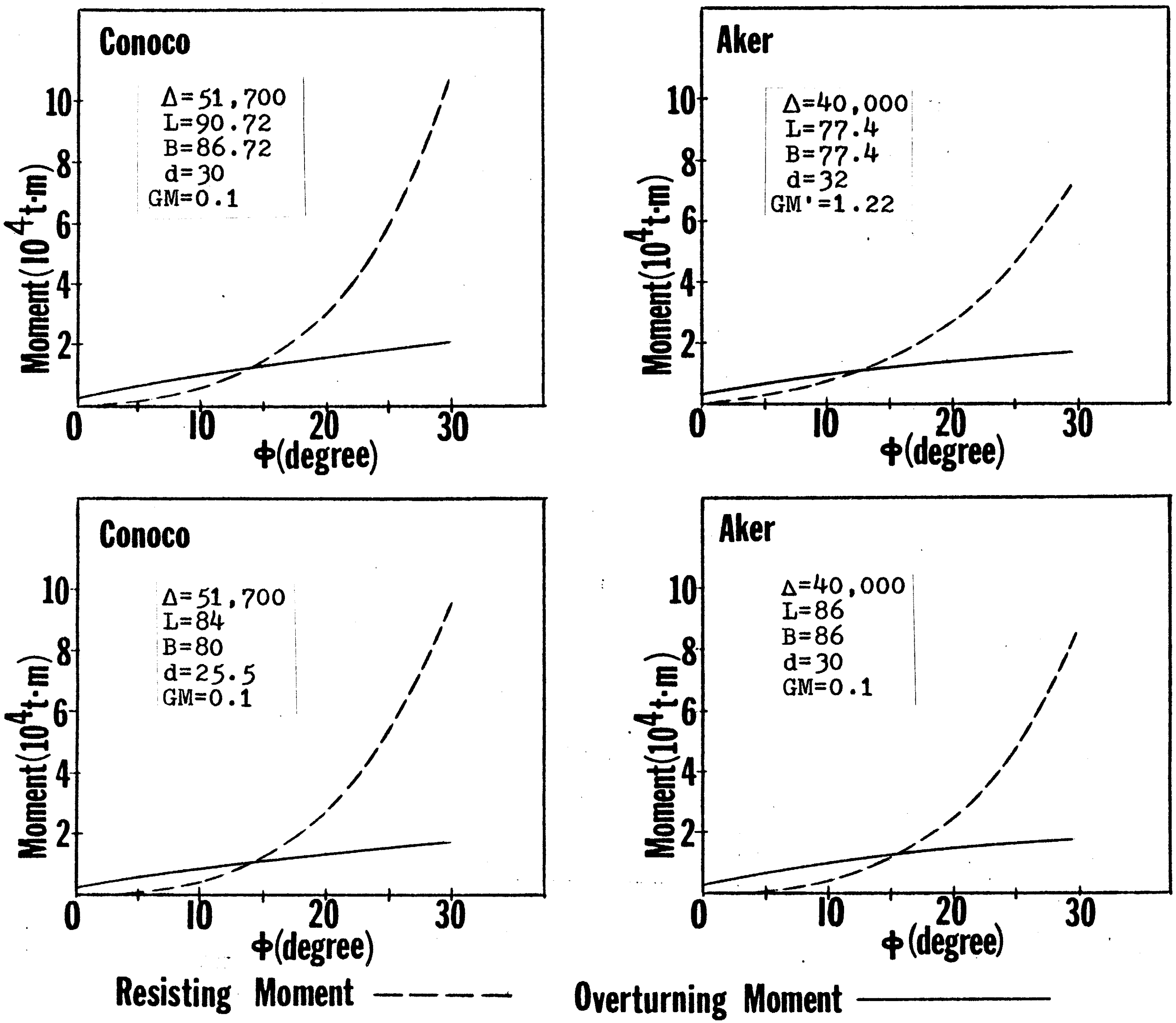


Fig8-8 Dynamic Stability

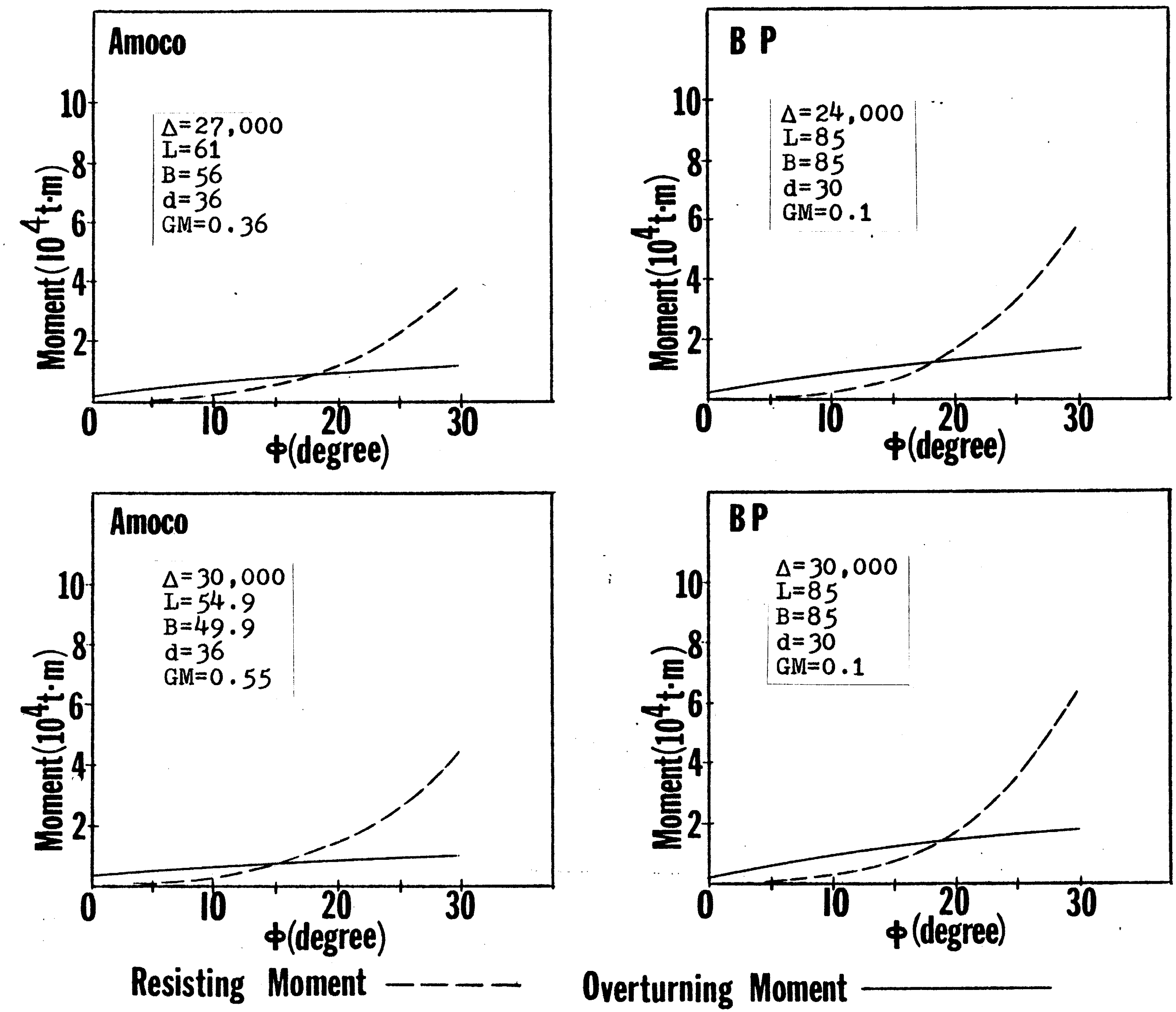


Fig 8-9B　　Fig 8-9A

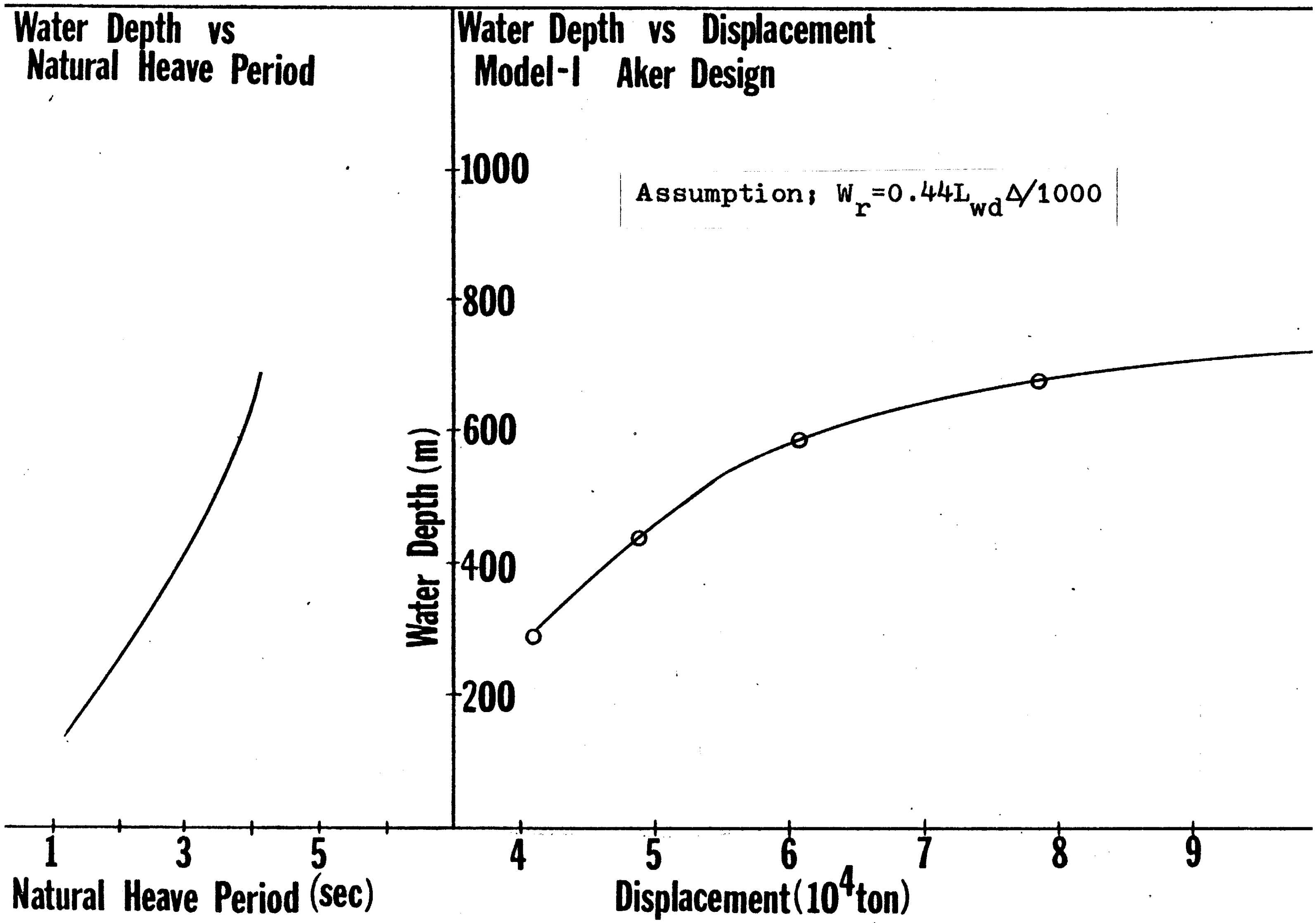


Fig 8-10B

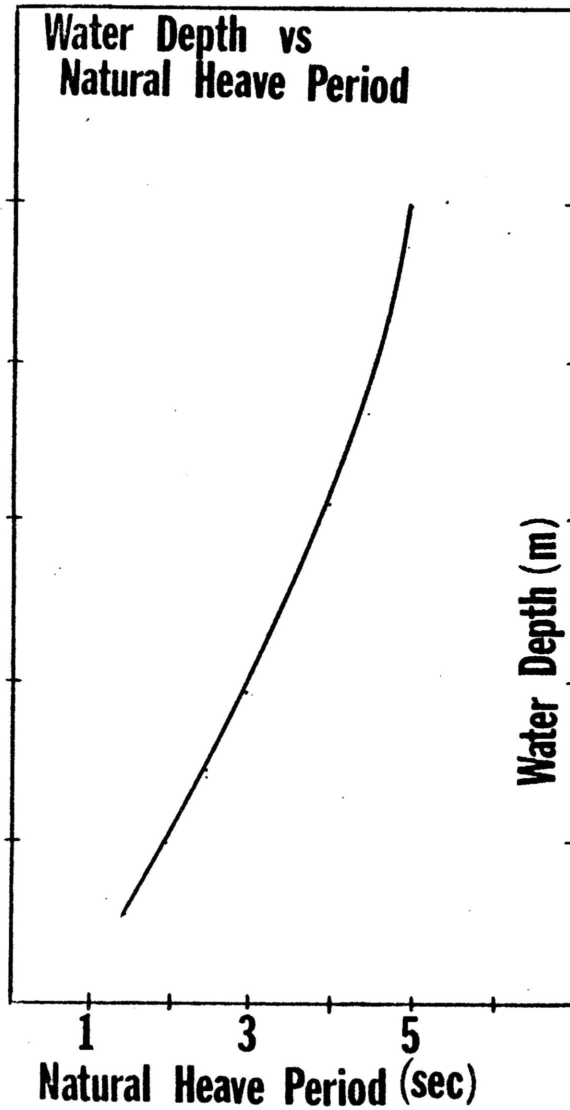


Fig 8-10 A

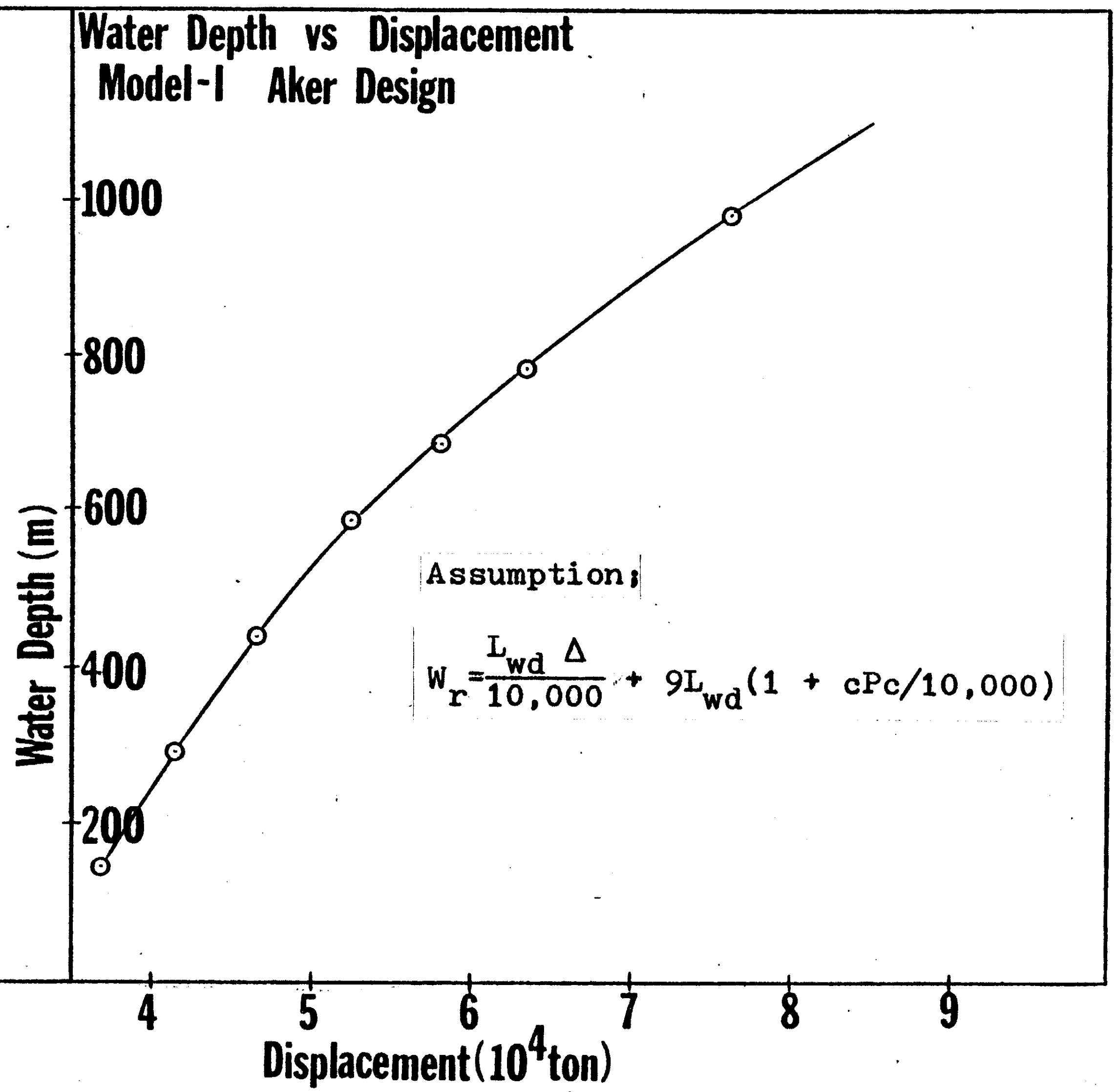


Table 8-1

Conoco Design		Approximate		By Computer	
Δ	Δ_ℓ	X_{stat}	X_{dyn}	X_{stat}	X_{dyn}
72,380	43,199	2.04	15.94	2.07	16.09
64,625	40,038	2.29	15.54	2.36	15.61
54,285	35,859	2.81	14.85	2.96	14.83
Aker Design		**Approximate**		**By Computer**	
Δ	Δ_ℓ	X_{stat}	X_{dyn}	X_{stat}	X_{dyn}
60,000	35,682	3.57	15.91	3.53	15.38
50,000	31,662	4.36	15.24	4.38	14.62
40,000	27,686	5.94	14.32	6.03	13.62
Amoco Design		**Approximate**		**By Computer**	
Δ	Δ_ℓ	X_{stat}	X_{dyn}	X_{stat}	X_{dyn}
39,000	18,387	16.14	14.07	15.11	13.44
30,000	15,010	20.30	13.70	18.47	13.01
24,000	12,988	25.93	13.21	22.49	12.51
B.P. Design		**Approximate**		**By Computer**	
Δ	Δ_ℓ	X_{stat}	X_{dyn}	X_{stat}	X_{dyn}
36,000	23,147	7.44	14.49	7.62	14.01
30,000	21,086	10.10	13.72	10.27	13.24
24,000	18,991	16.86	12.73	16.20	12.20

CHAPTER IX

DESIGN PROCEDURE OF THE TLP

As a result of this study the following design procedure is proposed.

Step 1 Using Eq.6-11, compute Δ for $\Delta_1 = 0.75\,\Delta$

$$\Delta_1 = cP_c + (0.4633 + 0.44L_{wd}/3{,}000)\Delta \qquad \text{(Eq.6-11)}$$

Step 2 Estimate the horizontal excursion by the following equations.

$$x_{dyn} \cong \frac{0.725\,\Delta}{(\Delta_1 + \Delta)/H_w - (\Delta - \Delta_1)*2.25/(L_{wd} - 30)} \qquad \text{(Eq.8-4)}$$

$$x_{stat} \cong \frac{(77 + 1.15\,\Delta/1{,}000)(v_c{}^2 + v_w{}^2/1{,}000)}{\Delta - \Delta_1}(L_{wd} - 30) \qquad \text{(Eq.8-5)}$$

Step 3 If $x_{dyn} + x_{stat} < x_{all}$ go to Step 4

If not, try smaller Δ_1/Δ and find out Δ so that

$$x_{dyn} + x_{stat} < x_{all}$$

Step 4 Compute M_1 or M_2 depending on which model is closer to your TLP configuration.

Step 5 If M_1 or $M_2 > 20$, set L as small as possible within the restriction of construction and operational requirements. If M_1 or $M_2 < 20$, set L as large as possible within the restriction of construction and operational requirements. Fig.6-3 is useful to obtain a reasonalbe estimate.

Step 6 Generally the draft is determined by construction and operational requirements. However for model 1 larger draft is prefered, whereas a smaller draft is prefered for model 2, if the other requirements allow it.

Fig.6-5 may be used to select a reasonable draft.

Step 7 The freeboard must satisfy Eq.6-15. Construction and operational requirements also influence its choice.

Step 8 Deck height is usually from 9 to 11m. This is determined mostly by operational and structural requirements.

Step 9 Start iteration to find real optimal design.

CHAPTER X

CONCLUSIONS AND RECOMMENDATIONS

Previous work, including 5 proposed designs, has been reviewed. Published data, including existing semi-submersibles, existing fixed platforms and proposed TLPs have been collected and analyzed. Based on this analysis, an equation to estimate the equipment weight has been derived and relations among various parameters are presented.

Fig.6-2 shows a very good approximation of the relation between the steel structure weight and the displacement. On the other hand, Fig.6-3 shows considerable scatter of data and thus further investigation is required to establish a more reliable relation between deck size and deck load. Other relations show generally good agreement with available data, but further analysis of additional design data is recommended.

The design requirements of the TLP have been studied and a method to determine the design period when only the maximum wave height is given, is presented. The towing-out condition and the operating condition with one leg totally lost are considered for stability requirements. The maximum horizontal excursion, dynamic stability and natural frequencies are also studied.

The natural frequencies of pitch and roll are approximated by their lower limit and the natural frequency of yaw is approximated by its upper limit, since mass distribution is not known. As the natural frequencies of pitch and roll become critical in deep water, a detailed investigation of the mass distribution is required. Information for the mass distribution is also important for an accurate evaluation of the static stability. The ultimate strength of the mooring line and its factor of safty are other unknown factors conserning the evaluation of the natural frequencies. Considerable research is required for the adequate selection of the material of the mooring line and its factor of safty.

As a result, computer programs are developed for two alternative models. Computer analysis indicates that for shallow water(less than 300m), the stability and the horizontal excursion are more important. On the other hand, for deep water(more than 300m), the positive tension requirement and the natural heave, pitch and roll frequencies become critical.

A sensitivity analysis has been carried out. The influence of the displacement, deck size and draft has been studied in detail. Two quantities are introduced for the two models, which indicate the relative importance of the dynamic effects versus static effects. It has been demonstrated that these quantities are useful to predict the change in the horizontal excursion due to change in the displacement, or the deck size.

It has been shown that the light weight can not exceed 75% of the displacement without violating the positive tension requirement. The results of the computer calculation also indicate that the natural frequencies of pitch, roll and heave will be very critical in deep water. It has been demonstrated that the TLP with a reasonable static stability satisfies the dynamic stability requirement.

The influence of the water depth on the TLP has also been studied using two different estimation procedures for the riser and mooring system weight. It has been shown that the maximum possible water depth for the TLP is about 1,000m although two results based on different estimation procedures give considerably different answers in deep water. For future research further analysis of additional design data is required.

As a result of the sensitivity analysis, a design procedure is proposed. Finally, further information is also required to improve the accuracy of estimating the light weight.

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APPENDIX I

DYNAMIC STABILITY

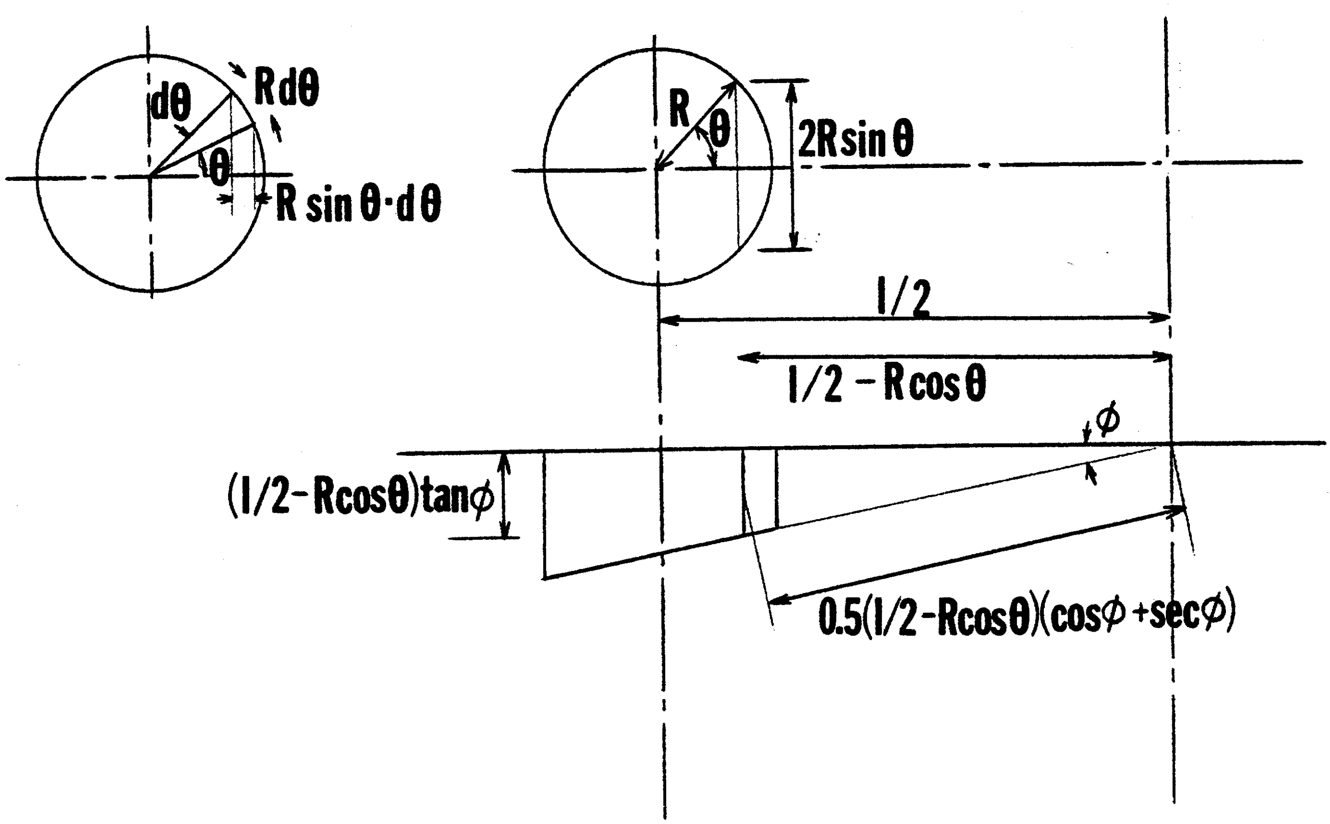
I.1 Righting Moment

$$dm_r=(2R\sin\theta)\rho(l/2-R\cos\theta)\tan\phi R\sin\theta d\theta(l/2-R\cos\theta)$$
$$*(\cos\phi+\frac{1}{\cos\phi})/2$$

$$dm_r=\tan\phi(\cos\phi+\frac{1}{\cos\phi})R^2\rho\sin^2\theta(l/2-R\cos^2\theta)^2d\theta$$

$$m_r=\tan\phi(\cos\phi+\frac{1}{\cos\phi})R^2\rho\int_0^{\pi}\sin^2\ (l/2-R\cos^2\theta)^2d\theta$$

$$=\rho\pi R^2\ \frac{1}{8}(l^2+R^2)(\cos\phi+\frac{1}{\cos\phi})\tan\phi$$



For 4 columns,

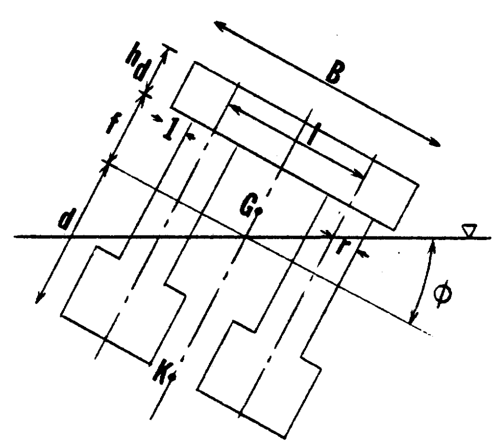
$$M_r=\frac{\rho\pi}{2}R^2(l^2+R^2)\sin\phi(1+\sec^2\phi)-\Delta^*BG\sin\phi$$
$$=\Delta BM\sin\phi\frac{1+\sec^2\phi}{2}-\Delta(KG-KB)\sin\phi$$

I.2 Heeling Moment

$$M_h=0.06255v^2\Big[1.65(h_d\cos\phi+B\sin\phi)L(f+d+h_d-KG)\cos\phi$$
$$+2r(f+\tfrac{l}{2}\tan\phi)\cos\phi(d+\tfrac{f}{2}-KG+\tfrac{l}{4}\tan\phi)\cos\phi$$
$$+2r(f-\tfrac{l}{2}\tan\phi)\cos\phi(d+\tfrac{f}{2}-KG-\tfrac{l}{2}\tan\phi)\cos\phi\Big]$$

$$M_h=0.06255v^2\Big[1.65(h_d\cos\phi+B\sin\phi)L(f+d+h_d/2-KG)\cos\phi$$
$$+4r\Big\{f(d+\tfrac{f}{2}-KG)\cos^2\phi+\tfrac{l^2}{8}\sin^2\phi\Big\}\Big]$$

l=B-2r-2



APPENDIX II

CPU PROGRAM LIST

(Model 1)

```
10   COM Velw,Velc,Fb,Hd,Bl,Sd,Dw,Wdl,Pi,W,G,Fu,Ib,C1,Wb,Wpe
20   INPUT Wpe,Dw0,W,Sd0,Fb,Bl0,Bb0,Velc,Velw,Wdl,Hw,Wb,Hd
30   PRINT Wpe,Dw0,W,Sd0,Fb,Bl0,Bb0,Velc,Velw,Wdl,Hw,Wb,Hd
40   INPUT Pind,Rmin,Rmax,Rinc
50   PRINT Pind,Rmin,Rmax,Rinc
60   Pi=3.1416
70   G=9.8
80   C1=.23
90   IF Pind<1.5 THEN GOTO Disp
100  IF Pind<2.5 THEN GOTO Leng
110  Bl=Bl0
120  Dw=Dw0
130  Bb=Bb0
140  PRINT "D"
150  FOR I=1 TO 30
160  I1=I-1
170  Coef=Rmin+I1*Rinc
180  IF Coef>Rmax THEN GOTO Term
190  Sd=Sd0*Coef
200  CALL Cont2(W1,Wl,Sr,Br,Bh,Dgm,Egm,Sb,Ftol,Xtol)
210  NEXT I
220  GOTO Term
230 Disp: Bl=Bl0
240  Bb=Bb0
250  Sd=Sd0
260  PRINT "DISP"
270  FOR I=1 TO 30
280  I1=I-1
290  Coef=Rmax-I1*Rinc
300  IF Coef<Rmin THEN GOTO Term
310  Dw=Dw0*Coef
320  CALL Cont2(W1,Wl,Sr,Br,Bh,Dgm,Egm,Sb,Ftol,Xtol)
330  NEXT I
340  GOTO Term
350 Leng: Sd=Sd0
360  Dw=Dw0
370  PRINT "B&L"
380  FOR I=1 TO 30
390  I1=I-1
400  Coef=Rmax-I1*Rinc
410  IF Coef<Rmin THEN GOTO Term
420  Bl=Bl0*Coef
430  Bb=Bl-Bl0+Bb0
440  CALL Cont2(W1,Wl,Sr,Br,Bh,Dgm,Egm,Sb,Ftol,Xtol)
450  NEXT I
460 Term: PRINT "END"
470  END
480  SUB Cont2(W1,Wl,Sr,Br,Bh,Dgm,Egm,Sb,Ftol,Xtol)
490  COM Velw,Velc,Fb,Hd,Bl,Sd,Dw,Wdl,Pi,W,G,Fu,Ib,C1,Wb,Wpe
```

```
500  Wra=(93.3+.44*WdT)*Dw/3000.0
510  Wds=.74*Bl*Bb
520  W1=Wra+Wds+Wpe
530  Bh0=Sd-.5*(Hw+1)
540  Bri=12
550  CALL Column(Bri,Bhi,Sbi,W1,W1,Sr)
560  Ebhi=Bh0-Bhi
570  Aebhi=ABS(Ebhi)
580  Brf=7
590  FOR I=1 TO 30
600  CALL Column(Brf,Bhf,Sbf,W1,W1,Sr)
610  Slope=(Bri-Brf)/(Bhi-Bhf)
620  Ebhf=Bh0-Bhf
630  Aebhf=ABS(Ebhf)
640  Espl=Aebhf/Bh0
650  IF Espl<1E-6 THEN GOTO Okey
651  IF Bhf<0 THEN GOTO Gost
660  IF Aebhi<Aebhf THEN GOTO Gost
670  Bri=Brf
680  Bhi=Bhf
690  Aebhi=Aebhf
700 Gost:     Dbr=Slope*(Bh0-Bhi)
710  Adbr=ABS(Dbr)
720  Dbr=Dbr/(1+Adbr)
730  Brf=Bri+Dbr
740  NEXT I
750  PRINT "STOP02"
751  PRINT Bhf,Brf,Sr,Sbf
760 Okey: CALL Stab(Brf,Bhf,Sbf,Dgm,Egm,W1,W1,Sr,Dkg,Dkb,Dbm,Ekb,Ebm,1)
770  IF Dgm<0 THEN GOTO Sbchek
780  IF Egm<0 THEN GOTO Sbchek
830  GOTO Xmax
840 Sbchek: IF Sbf>-.5 THEN GOTO Hmin
841  Bri=10
850  IF Sbf>0 THEN GOTO Hmin
860  CALL Column(Bri,Bhi,Sbi,W1,W1,Sr)
870  Sb0=-.5
880  Esbi=Sb0-Sbi
890  Aesbi=ABS(Esbi)
900  Brf=15
910  FOR I=1 TO 30
920  CALL Column(Brf,Bhf,Sbf,W1,W1,Sr)
930  Slope=(Bri-Brf)/(Sbi-Sbf)
940  Esbf=Sb0-Sbf
950  Aesbf=ABS(Esbf)
960  Asb0=ABS(Sb0)
970  Eps2=Aesbf/Asb0
980  IF Eps2<1E-6 THEN GOTO Good
990  IF Aesbi<Aesbf THEN GOTO Stay
1000 Sbi=Sbf
1010 Bri=Brf
1020 Aesbi=Aesbf
1030 Stay:Dsb=Slope*(Sb0-Sbi)
1040 Adsb=ABS(Dsb)
1050 Dsb=Dsb/(1+Adbr)
```

```
1060 Brf=Bri+Dsb
1070 NEXT I
1080 PRINT "STOP03"
1090 Good:CALL Stab(Brf,Bhf,Sbf,Dgm,Egm,W1,Wl,Sr,Dkg,Dkb,Dbm,Ekb,Ebm,1)
1100 IF Dgm<0 THEN GOTO Hmin
1110 IF Egm<0 THEN GOTO Hmin
1120 GOTO Xmax
1130 Hmin:Bri=10
1140 CALL Column(Bri,Bhi,Sbi,W1,Wl,Sr)
1150 Ebhi=5-Bhi
1160 Aebhi=ABS(Ebhi)
1170 Brf=15
1180 FOR I=1 TO 30
1190 CALL Column(Brf,Bhf,Sbf,W1,Wl,Sr)
1200 Slope=(Bri-Brf)/(Bhi-Bhf)
1210 Ebhf=5-Bhf
1220 Aebhf=ABS(Ebhf)
1230 Eps3=Aebhf/5
1240 IF Eps3<1E-6 THEN GOTO Nice
1250 IF Aebhi<Aebhf THEN GOTO Same
1260 Bri=Brf
1270 Bhi=Bhf
1280 Aebhi=Aebhf
1290 Same: Dbr=Slope*(5-Bhi)
1300 Adbr=ABS(Dbr)
1310 Dbr=Dbr/(1+Adbr)
1320 Brf=Bri+Dbr
1330 NEXT I
1340 PRINT "STOP04"
1350 Nice:CALL Stab(Brf,Bhf,Sbf,Dgm,Egm,W1,Wl,Sr,Dkg,Dkb,Dbm,Ekb,Ebm,-1)
1360 IF Dgm<0 THEN GOTO Over
1370 IF Egm<0 THEN GOTO Over
1380 Dgm1=Dgm
1381 Egm1=Egm
1382 Bri=Brf
1383 Brf=Bri-1
1384 FOR I=1 TO 30
1385 CALL Column(Brf,Bhf,Sbf,W1,Wl,Sr)
1390 CALL Stab(Brf,Bhf,Sbf,Dgm,Egm,W1,Wl,Sr,Dkg,Ikb,Dbm,Ekb,Ebm,-1)
1391 IF Dgm<Egm THEN GOTO Tow
1392 Slope=(Bri-Brf)/(Egm1-Egm)
1393 Ergm=.1-Egm
1394 Ergm1=.1-Egm1
1395 Dbr=Slope*(.1-Egm)
1400 GOTO Ten
1401 Tow:Slope=(Bri-Brf)/(Dgm1-Dgm)
1402 Ergm=.1-Dgm
1403 Ergm1=.1-Dgm1
1404 Dbr=Slope*(.1-Dgm)
1410 Ten:Aergm=ABS(Ergm)
1411 Eps9=Aergm/.1
1412 IF Eps9<1E-6 THEN GOTO Agri
1420 Aergm1=ABS(Ergm)
1421 IF Aergm1<Aergm THEN GOTO Yuke
1422 Bri=Brf
```

```
1423 Dgm1=Dgm
1430 Egm1=Egm
1431 Yuke:Adbr=ABS(Dbr)
1432 Dbr=Dbr/(1+Adbr)
1440 Brf=Bri+Dbr
1450 NEXT I
1460 PRINT "STOP 08"
1461 Agni:CALL Stab(Brf,Bhf,Sbf,Dgm,Egm,W1,Wl,Sr,Dkg,Dkb,Dbm,Ekb,Ebm,1)
1470 GOTO Xmax
1480 Over:CALL Stab(Brf,Bhf,Sbf,Dgm,Egm,W1,Wl,Sr,Dkg,Dkb,Dbm,Ekb,Ebm,1)
1481 PRINT "NO GOOD"
1490 GOTO Fin
1500 Xmax:CALL Xmax(Sr,Bhf,Brf,Wl,Ekb,Dkg)
1510 Fin:SUBEND
1520 SUB Column(Br,Bh,Sb,W1,Wl,Sr)
1530 COM Velw,Velc,Fb,Hd,Bl,Sd,Dw,Wdl,Pi,W,G,Hu,Eb,C1,Wb,Wpe
1540 Br2=Br^2
1550 Sk=W^2/G
1560 Ek2=EXP(-Sk*Sd)
1570 Ek1=SQR(Ek2)
1580 Dv4=Dw/(4.0*Pi)/1.025
1590 Br3=Br^3
1600 Sr2=(Dv4*Ek1+4.0*Br3*Ek2/3)*Sk
1610 Sr=SQR(Sr2)
1620 Bh=(Dv4-Sr2*Sd)/(Br2-Sr2)
1630 R2h=Br2*Bh
1640 R2d=Sr2*(Fb+Sd-Bh)
1650 Wl=4.0*Pi*(R2h+R2d)*C1+W1+Wb
1660 Vl=Wl/1.025
1670 Sb=(Vl-4.0*Pi*Br2*Bh)/(4.0*Pi*Br2)
1680 IF Sb<0 THEN GOTO Neg
1690 Sb=Sb*Br2/Sr2
1700 Neg:SUBEND
1710 SUB Stab(Br,Bh,Sb,Dgm,Egm,W1,Wl,Sr,Dkg,Dkb,Dbm,Ekb,Ebm,Ii)
1720 COM Velw,Velc,Fb,Hd,Bl,Sd,Dw,Wdl,Pi,W,G,Hu,Eb,C1,Wb,Wpe
1730 Vl=Wl/1.025
1740 Br2=Br^2
1750 R2h2=Br^2*Bh^2
1760 Sr2=Sr^2
1770 R2d2=Sr2*(Fb+Sd-Bh)*(Fb+Sd+Bh)
1780 Sl=Bb-2*(Sr+1)
1790 Sl2=Sl^2
1800 Shb=1.1*Wb/(4*Pi*Br2)
1810 IF Shb<Bh THEN GOTO Usul
1820 Bwsr=1.1*Wb-4*Pi*Br2*Bh
1830 Shb=Bh+Bwsr/(4*Pi*Sr2)
1840 Slb=(1.1*Wb*Bh+Shb*Bwsr)*.5/(1.1*Wb)
1850 GOTO Moto
1860 Usul: Slb=.5*Shb
1870 Moto:Dkg=(2*Pi*(R2h2+R2d2)*C1+W1*(Sd+Fb+.5*Hd)+Wb*Slb)/Wl
1880 IF Sb<0 THEN GOTO Bigr
1890 Br2=Br^2
1900 Dkb=2*Pi*(R2h2+Sr2*Sb*(Sb+2*Bh))/Vl
1910 Dbm=Pi*Sr2*(Sl2+S^2)/Vl
1920 Dgm=Dkb+Dbm-Dkg
```

```
1930 GOTO Emerg
1940 Bign:    Br2=Br^2
1950 Dkb=2*Pi*Br2+(Bh+Sb)^2/Vl
1951 IF Sb>-.5 THEN GOTO Baka
1960 Dbm=Pi*Br2*(Sl2+B^2)/Vl
1961 GOTO Aho
1962 Baka:Dbm=Pi*Sr2*(Sl2+Sr2)/Vl
1963 Aho:Ii=1
1970 Dgm=Dkb+Dbm-Dkg
1980 Emerg:    Sd2=Sd^2
1990 Bh2=Bh^2
2000 D2h2=Sd2-Bh2
2010 Ekg=Dkg
2020 Ekb=2*Pi*(R2h2+Sr2*D2h2)/Vl
2030 Ebm=Pi*Sr2*(Sl2+S^2)/Vl
2040 Egm=Ekb+Ebm-Ekg
2041 IF Ii<0 THEN GOTO Rtun
2050 PRINT "DATA";Dw,Bl,Bb,Sd
2060 PRINT Br,Sr,Bh,Wl,Sb,Dgm,Dbm,Dkb,Dkg,Egm,Ebm,Ekb
2070 IF Dgm<0 THEN GOTO Rtun
2080 IF Egm<0 THEN GOTO Rtun
2084 DEG
2100 Velw0=51.5^2
2110 FOR I=1 TO 6
2120 Fai=5*I
2130 Cosf=COS(Fai)
2140 Sinf=SIN(Fai)
2150 Cosf2=Cosf^2
2160 Sinf2=Sinf^2
2170 Rmom1=(1+1/Cosf2)*.5
2180 Rmomd=Wl*Sinf*(Dbm*Rmom1+Dkb-Dkg)
2190 Rmome=Wl*Sinf*(Ebm*Rmom1+Ekb-Ekg)
2200 Omom1=1.65*(Hd*Cosf+Bb*Sinf)*Bl*(Fb+Sd+.5*Hc-Dkg)*Cosf
2210 Omom2=4*Sr*Cosf2*Fb*(Sd+.5*Fb-Dkg)
2220 Omom3=.5*Sr*Sl2*Sinf2
2230 Omome=Omom1+Omom2+Omom3
2240 Omome=Omome*Velw0*.06255/1000
2250 IF Sb<0 THEN GOTO Bneg
2260 Omom4=2*Sr*Cosf2*(Sd-Sb-Bh)*(Sd+Sb+Bh-2*Dkg)
2270 GOTO Bpos
2280 Bneg:Omom5=2*Sr*Cosf2*(Sd-Bh)*(Sd+Bh-2*Dkg)
2290 Omom6=4*Br*Cosf2*Sb*(Bh+.5*Sb-Dkg)
2300 Omom4=Omom5-Omom6
2310 Omom3=Omom3*Br/Sr
2320 Bpos:Omomd=Omom1+Omom2+Omom3+Omom4
2330 Omomd=Omomd*Velw0*.06255/1000
2340 PRINT Omomd,Rmomd,Omome,Rmome
2350 NEXT I
2360 RAD
2370 Rtun:SUBEND
2380 SUB Xmax(Sr,Bh,Br,Wl,Ekb,Dkg)
2390 COM Velw,Velc,Fb,Hd,Bl,Sd,Dw,Wdl,Pi,W,G,Hu,Ib,Cl,Wb,Wpe
2400 Sr2=Sr^2
2410 W2=W^2
2420 Sk=W2/G
```

```
2430 Sl=Bb-2*(Sr+1)
2440 Ek3=EXP(Sk*Sd/2)
2450 Velw2=Velw^2
2460 Arw=1.65*Bl*Hd+4.0*Fb*Sr
2470 Fwin=.06255*Velw2*Arw/1000
2480 Velc2=Velc^2
2490 Arc=(Sd-Bh)*Sr+Br*Bh
2500 Fcur=.4188*Velc2*Arc
2510 Ftol=Fwin+Fcur
2520 Tens=Dw-Wl
2530 Cabl=Wdl-Sd
2540 Xstat=Cabl*Ftol/Tens
2550 FOR I=1 TO 30
2560 Sinf=Xstat/Cabl
2570 Tet=ASN(Sinf)
2580 Cosf=COS(Tet)
2590 Rest1=4.0*Pi*(1-Cosf)*Cabl*Sr2
2600 Restf=(Rest1+Tens)*Sinf/Cosf
2610 Esp4=(Restf-Ftol)/Ftol
2620 Esp4=ABS(Esp4)
2630 IF Esp4<1E-6 THEN GOTO Xdyn
2640 Xstat=Xstat+(Ftol-Restf)*Cabl/Tens
2650 NEXT I
2660 PRINT "STOP05"
2670 Xdyn:  Tet2=Sk*Sl/2
2680 Cost=COS(Tet2)
2690 Disr=Wl/Dw
2700 A1=(1-Disr)*G/Cabl
2710 B1=Disr+1
2720 Ab1=W2*B1-A1
2730 Ab2=Ab1^2
2740 Ab3=Ab2+.01*4.0*A1*B1*W2
2750 Ab4=SQR(Ab3)
2760 Xdyn=Hw*W2*Cost/Ab4/Ek3
2770 Xtol=Xdyn+Xstat
2780 Trm=2*.44*Wdl*Dw/3000.0
2790 Prest=Tens-Trm
2800 Dtdyn=2.546*Hw*Sr2
2810 Slf=Ekb*Wl/Dw
2820 Fwav=Xdyn*W2*Wl/G
2821 Ften=Tens*Xdyn/Cabl
2830 Amwa=Fwav*(Dkg-Slf)+Ften*Slf
2840 Alwin=1.65*Bl*Hd*(Fb+Sd+.5*Hd)+4.0*Fb*Sr*(Sc+.5*Fb)
2850 Slw=Alwin/Arw
2860 Amwi=Fwin*(Slw-Dkg)
2870 Alcur=Sr*(Sd-Bh)*(Bh+.5*Sd)+.5*Br*Bh^2
2880 Slc=Alcur/Arc
2890 Amcu=Fcur*(Dkg-Slc)
2900 Amtol=Amwa+Amcu+Amwi
2910 Dtpit=Amtol*2/Sl
2920 Dtt2=Dtdyn^2+Dtpit^2
2930 Dtt=SQR(Dtt2)
2940 PRINT "XMAX";Fwin,Fcur,Ftol,Xstat,Xdyn,Xtol,Prest
2950 PRINT Dtdyn,Disr,Dtpit,Dtt
2960 Fsaf=6
```

```
2970 Sall=100000
2980 Ey=2.1E7
2990 Wss2=(1-Disr)*G/(1+Disr)/Cabl
3000 Wss=SQR(Wss2)
3010 Tss=2*Pi/Wss
3020 Acbl=Prest*Fsaf/Sall
3030 Wh1=(Acbl*Ey/Cabl+4.0*Pi*Sr2*1.025)*G
3040 Wh2=Wl+Br^3*Pi*1.025*16/3
3050 Who2=Wh1/Wh2
3060 Who=SQR(Who2)
3070 Tho=2*Pi/Who
3080 Wpr1=Wl+Dw
3090 Wpr2=Wh1/Wpr1
3100 Wpr=SQR(Wpr2)
3110 Tpr=2*Pi/Wpr
3120 Wy1=12*Tens/Cabl
3130 Wy2=Wy1/Wpr1
3140 Wy=SQR(Wy2)
3150 Ty=2*Pi/Wy
3160 PRINT Wss,Who,Wpr,Wy
3170 PRINT Tss,Tho,Tpr,Ty
3180 SUBEND
```

APPENDIX III

TYPICAL RESULTS

Table A-1

Aker Design(Model 1)		Parameter-Draft		
Input Data				
W(t)	Δ(t)	ω (1/s)	d(m)	f(m)
7,000	40,000	0.38	?	25.6
L(m)	B(m)	v_c(m/s)	v_w(m/s)	L_{wd}(m)
86	86	1.35	56	150
H_w(m)	h_d(m)			
30	9			

Output Results							
d	22	24	26	28	30	32	34
x_{total}	22.23	21.40	20.80	20.36	20.21	19.62	19.35
T_0	7,618	8,596	9,303	9,829	9,673	10,554	10,813
ΔT_{dyn}	9,256	8,471	7,945	7,595	7,825	7,188	7,076
GM	12.78	8.16	4.28	0.98	0.1	42.04	35.79
GM'	18.35	14.83	11.95	9.59	9.48	6.04	4.69
Δ_1/Δ	0.766	0.741	0.723	0.71	0.714	0.692	0.686
T_{ss}	62.31	58.42	55.79	53.86	53.84	51.12	50.06
T_H	2.28	1.97	1.76	1.61	1.66	1.40	1.32
T_{pr}	1.93	1.80	1.71	1.64	1.64	1.55	1.52
T_y	56.31	52.80	50.42	48.67	48.66	46.19	45.24

Table A-2

Aker Design(Model 1)		Parameter-Displacement		
Input Data				
W(t)	Δ(t)	ω (1/s)	d(m)	f(m)
7,000	?	0.38	32	25.6
L(m)	B(m)	v_c(m/s)	v_w(m/s)	L_{wd}(m)
86	86	1.35	56	150
H_w(m)	h_d(m)			
30	9			

Output Results							
Δ	60,000	55,000	50,000	45,000	40,000	35,000	30,000
x_{total}	18.90	18.92	19.00	19.21	19.65		
T_0	21,678	18,913	16,138	13,352	10,554		
ΔT_{dyn}	10,834	9,899	8,980	8,076	7,188		
GM	46.44	45.98	45.16	43.89	42.04		
GM'	16.11	14.18	11.90	9.22	6.04		
Δ_1/Δ	0.595	0.612	0.633	0.659	0.692		
T_{ss}	43.25	44.45	46.01	48.12	51.12		
T_H	1.205	1.23	1.27	1.32	1.40		
T_{pr}	1.287	1.33	1.38	1.45	1.55		
T_y	39.08	40.17	41.58	43.48	46.19		

Table A-3

Aker Design(Model 1)		Prameter-Deck Size		
Input Data				
W(t)	Δ(t)	ω (1/s)	d(m)	f(m)
7,000	40,000	0.38	32	25.6
L(m)	B(m)	v_c(m/s)	v_w(m/s)	L_{wd}(m)
?	?	1.35	56	150
H_w(m)	h_d(m)			
30	9			

Output Results							
L	103.2	98.9	94.6	90.3	86.0	81.7	77.4
B	103.2	98.9	94.6	90.3	86.0	81.7	77.4
x_{total}	19.82	19.68	19.62	19.79	19.65	19.72	19.82
T_0	8,149	8,789	9,405	9,600	10,554	11,088	11,594
T_{dyn}	6,177	6,388	6,623	7,223	7,188	7,530	7,923
GM	8.23	4.98	1.79	0.1	42.04	34.49	27.18
GM'	16.49	13.81	11.16	9.92	6.04	3.59	1.22
Δ_1/Δ	0.752	0.736	0.721	0.716	0.692	0.679	0.666
T_{ss}	58.00	55.94	54.14	53.59	51.12	49.85	48.71
T_H	1.62	1.55	1.49	1.57	1.40	1.36	1.32
T_{pr}	1.79	1.72	1.66	1.64	1.55	1.51	1.47
T_y	52.41	50.55	48.92	48.43	46.19	45.05	44.02