FOURTEENTH WEGEMT SCHOOL: SLENDER MARINE STRUCTURES

Trondheim, Norway, 21-25 January 1991

SCHOOL PROGRAMME

Day 1 - Monday 21 January

11.00 - 11.45	Introduction Motivation Carl M Larsen, NTH
12.00 - 12.45	Tensioned beam, the basic differential equation Marinus Van Holst, Delft
14.00 - 14.45 15.00 - 15.45	Effect from pressure and internal flow Hydrodynamic force models Van Holst/Erling Huse
16.00 - 16.45	Finite element models C M Larsen

Day 2 - Tuesday 22 January

09.00 - 09.45	Introduction to marine riser systems D G Owen, Heriot-Watt
10.00 - 10.45 11.00 - 11.45	Methods for stochastic dynamic analysis Linearization techniques C M Larsen
14.00 - 14.45	Modelling strategy, global riser analysis D G Owen
15.00 - 18.00	Exercise: Use of RISANA, riser analysis

Day 3 - Wednesday 23 January

09.00 - 09.45	Stress analysis, bonded flexible pipes D G Owen
10.00 - 10.45	Stress analysis, unbonded pipes Arild Bech, SINTEF

11.00 - 11.45	Components in riser systems D G Owen
12.00 - 12.45	Fatigue analysis, steel risers C M Larsen
14.00 - 14.45	Extreme response estimation, steel risers C M Larsen
15.00 - 15.45	Model tests of tensioned and flexible risers D G Owen
16.00 - 18.00	Exercise: Use of RISANA, riser analysis

Day 4 - Thursday 24 January

09.00 - 09.45	Installation of pipelines and risers Ivar Fylling, MARINTEK
10.00 - 10.45 11.00 - 11.45	Upheaval buckling of pipelines P T Pedersen, DTH
12.00 - 12.45	Towing of long pipeline sections H Boonstra, Delft
14.00 - 14.45	Buckling of pipelines H Boonstra
15.00 - 17.00	Exercise: Pipeline analysis, upheaval buckling

Day 5 - Friday 25 January

09.00 - 09.45 10.00 - 10.45	On-bottom stability of pipelines Torbjørn Sotberg, SINTEF
11.00 - 11.45 12.00 - 12.45	Free-span problems Mads Bryndum, Danish Hydraulic Inst
14.00 - 14.45 15.00 - 15.45	Pipeline stresses and deformation during laying P T Pedersen

CONTENT

WEGEMT School, Trondheim, January 1991.

TOWING OF LONG PIPELINE SECTIONS.

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SUMMARY

The transport of long pipeline sections from fabrication site to an offshore field by means of towing is introduced. The dynamic analysis of one of the possible towing methods, the Controlled Depth (or Mid-Depth) Tow is discussed in more detail. The method of analysis in a lot of aspects resembles the analysis of risers, but due to the specific circumstances there are differences as well.

1. INTRODUCTION

Most offshore pipelines are installed by means of a lay-barge at which prefabricated pipeline sections of about 12 m length are welded together to one continuous pipe just before installation onto the seabed. While for long bulk lines from offshore field to shore there hardly exist alternative installation methods, the lines of smaller length, (generally in-field lines) can be fabricated on-shore and subsequently be transported to their final location.

For small diameter lines the so-called reel-method can be used: the steel pipe is wound on a large diameter reel placed on a barge and is at the final location un-reeled, straightened and lowered to the seabed in a similar way as a conventional pipeline.

For large diameter pipe and bundles of pipes, flowlines and control lines the reel-method is not a viable option, in this case the transport can take place by means of towing of the pipe string from the fabrication site at shore to the offshore field.

The advantages of transporting pipelines by means of towing can be summarized as follows:

- A number of small diameter flowlines and control lines can be enclosed by one carrier pipe, which prevents congestion in the field.
- If required the individual lines can be provided by insulation.
- All lines can be tested extensively on shore.
- The method does not require expensive offshore installation equipment.

Limitations of the method are:

- The length of the pipeline bundles is generally limited to say four or seven kilometers (although in one case a single line of 21 kilometers length has been transported by means of an off-bottom tow).
- A spacious on-shore construction site and shallow beach and landfall are required at a location not to far from the field of installation.

This paper introduces the various possible methods of towing pipe strings and discusses methods to analyze the complex dynamic behavior of the pipe strings during transport.

2. TRANSPORT AND INSTALLATION OF PIPELINE STRINGS

Methods to tow a prefabricated pipeline or pipeline bundle to an offshore field can be distinguished as follows:

- Bottom tow
- Off-bottom tow
- Surface tow
- Near-surface tow
- Controlled depth tow
  (also called: Mid-depth tow)

These various tow methods are schematically shown in Figure 1 and are discussed briefly below.

It will be clear that the bottom tow method can only be applied when the length of the pipeline is limited, otherwise the required pull force becomes excessive. Furthermore the seabed along the route must be flat and clear from even the smallest obstacle and manoeuvering with the pipe section is virtually impossible. The bottom tow is in practice only used for the installation of land-fall pipelines of small length.

In case of the off-bottom tow the pipeline (or pipeline bundle) is provided with buoyancy just in excess of its own weight. This buoyancy can either be achieved by tying separate buoyancy cans at regular intervals to the pipeline or by putting the pipeline into a larger diameter carrier pipe, which then provides continuous buoyancy along the total length. Small sections of ballast chain prevent the pipeline from floating to the surface, the structure hovers at a few.meters above the seabed (see Figure 2). It goes without saying that also in this case the seabed along the route must be reasonably flat and should be surveyed extensively for (larger) obstacles like ship wrecks, rocks, etc. Pipelines have been towed successfully in the past for hundreds of kilometers in this way. Advantages with respect to the bottom tow are the smaller towing resistance, the increased maneuverability and the clearance from the seabed.

When the pipeline is provided with excess buoyancy which is not compensated by ballast chain sections, the system will float to the surface. It can easily be shown that relatively small waves will cause unacceptable stresses in the pipe. Therefore surface tow can only take place in areas with a very mild climate, and even then the towing distance must be limited in order to be sure of very calm sea conditions.

A transport mode which is less vulnerable to sea waves is the near-surface tow. In this case the buoyancy provided along the pipeline is just insufficient to let the pipe float. Buoys are connected at regular intervals to the pipe by means of a steel or nylon wire of say 25 m length, from which the pipeline is supported. As a result the pipeline itself will remain free of the most severe wave loading, although the buoys itself introduce dynamic loadings to the pipeline when sailing in waves. Therefore the buoys should be kept as small as possible, which calls for a very accurate weight and buoyancy control of the pipeline system.

The most interesting pipeline transport method, certainly from the engineering point of view is the controlled depth (or mid-depth) tow. The configuration of the system is similar to the off-bottom tow, however care is taken to keep the submerged weight of the pipeline (or pipeline bundle), including the chains very small and within narrow limits. At zero forward speed the pipeline floats at a few meters above the seabed, with a few links of each chain section resting on the seabed, just as in case of the off-bottom tow method. When the leading tug starts moving the pipeline the drag force on the chain will cause the chain to hang slightly out of the vertical plane (see Fig.3). As a result the same drag force no longer will act in the horizontal plane, but it will get a (small) component in the vertical (upward) direction. This lift force will decrease the virtual weight of the pipeline and, when the submerged weight is sufficiently small, at a certain sailing speed the pipeline will lift off the seabed. The system now looks like shown in Fig.4. By varying the sailing speed, the paid-out length of the towing lines and the tension in the towing wires fore and aft, the vertical position of the towheads of the pipeline depth of the midsection of the pipeline can be controlled. With the controlled depth tow described here the disadvantages of the (near)-bottom tow and the (near)-surface tow are avoided: the pipeline remains well clear of the seabed and the transport is not hindered by obstacles such as pipelines or shipwrecks, on the other hand the pipeline can be kept sufficiently below the water surface to avoid excessive wave loading.

3. STATIC BEHAVIOR OF PIPELINES DURING TOW

The most important static engineering calculations and checks which have to be made for a controlled depth tow are indicated below ("static" in the present context means the exclusion of wave effects):

- Towing forces and axial tension.

The axial tension in the pipeline at an arbitrary section is calculated from the tow wire tension (fore and aft), the frictional resistance of the pipe and of the chain sections, and the end-cap forces due to internal pressure and external water head. In the static towing situation the tow forces of the leading and trailing tug are in equilibrium with the frictional resistance. For the dynamic analysis only the tension caused by the tow wires (the effective tension) is of importance. Tension due to end cap loads caused by external and internal hydrostatic pressure does not effect the dynamic behavior.

- Hydrostatic pressure and hoop stresses.

It must be checked that the hydrostatic pressure does not exceed the tube buckling (also called: collapse) pressure or the buckle propagating pressure of the carrier pipe. Because of weight considerations the wall thickness of the carrier pipe is limited and buckling may become a governing factor, especially for deep water applications. As a precaution the carrier pipe is generally pressurized with nitrogen at a pressure equal to the external waterhead at the final location. However if insulated flowlines are present inside the carrier pipe, the thin wall sleeve pipe around insulation may collapse due to external pressure. An other point of concern is that the internal pressure may cause excessive tensile hoop stresses in the carrier pipe when the pipe is still at or near the water surface.

- <u>Weight control.</u>

Pipelines which are to be towed to the final location by means of off-bottom tow, controlled depth tow or near surface tow are very weight sensitive. Therefore during design, but also during fabrication the weight must be monitored carefully. A small amount of under-weight can be compensated by adding some additional shackles of chain but over-weight is detrimental because it will hamper lift-off from the seabed.

- <u>Vertical positioning of the pipeline during tow.</u>

By a careful calculation of tow wire forces, effective weight of the pipeline, and lift forces acting on the chain sections due to forward speed, the static position of the pipeline as shown in Fig 6b can be predicted.

4. DYNAMIC ANALYSIS OF PIPELINES DURING TOW

For a description of the dynamic analysis (i.e. the effects caused by waves) of towed pipelines, we will concentrate on the controlled depth tow, as this is the most complicated situation. Rather than giving one possible method of analysis, the basic aspects will be discussed, paying attention on how a rather complicated problem can be simplified by engineering judgement to a mathematical description which can be tackled with computational tools known from similar engineering problems. The pipeline (and also the tow wires) can be modelled by means of a Finite Element Method (FEM) or a Lumped Mass Method (LMM), the solution can be found analytically (which requires a lot of simplifications), numerically in the frequency domain (which requires linearization of the fluid loading), or numerically in the time domain (which calls for a lot of computer time). For various methods of analysis and descriptions of pipeline towing methods reference is made to papers by Verner [1], Langley [2], Rooduyn and Boonstra [3,4], Van den Boom [5] and De Boef [6].

- **The problem.**

Figure 4 shows schematically the components of the system.
There is a very long (several kilometers) pipe with a relatively small diameter (say 0.4 m to 1.0 m) at some 30 to 60 m below the water surface. The pipe is provided with small sections of chain each 20 m or so. A towhead at the fore and aft end of the pipe facilitates the connection of tow wires. These towheads have cross-sectional dimensions and a mass per unit length which is a magnitude higher than the properties of the pipeline. The tow wires consist of a steel bridle, a nylon hawser (to increase the elasticity) and a steel wire rope which is run from the winch at the tug.

The dynamic loading of the pipe is caused by:

- direct fluid loading on the pipe itself (and on the chain sections)
- direct fluid loading on the towheads
- varying forces in the tow wires caused by wave induced motions of the tugs

It is not easy to find out which of these effects are the most important, so basically all loadings should be included in an engineering model, if possible.

The behavior of the pipeline under dynamic loading and the resulting stresses depend on its flexural rigidity (EI) and the tension in the pipe caused by the pull forces applied by the two tugs, the so-called "effective tension". This tension varies along the pipeline due to the viscous resistance of the pipeline and the chain sections. Please note that the axial stress in the pipe due to internal or external pressure does not contribute to the bending stiffness!

From the description above it follows that the system of a towed pipeline in a lot of aspects resembles a rigid or a flexpipe riser system, where a tensioned pipe, with flexural rigidity is suspended from a floating platform. There are differences as well off course, the most important probably is caused by the enormous length of the pipeline, which makes any phase relation between fluid loadings on the fore and aft part of the pipe and between direct fluid loadings and dynamic tow wire forces speculative. Still the experience gained in the analysis of riser systems is worthwhile when analyzing pipelines during tow.

- Methods of analysis.

The differential equation the motion in the vertical plane of a segment of the pipeline may be written as:

$$m\frac{\partial^2 z}{\partial t^2}+EI\frac{\partial^4 z}{\partial x^4}-T_x\frac{\partial^2 z}{\partial x^2}=f_x \qquad (1)$$

In which:

m = mass per unit length
z = vertical displacement
x = coordinate along the pipeline
EI = flexural rigidity
T_x = axial tension
f_x = fluid loading per unit length
t = time

The mass, the flexural rigidity, the tension and the fluid loading will generally vary along the pipeline.

The fluid loading on the pipeline can be approximated by the generalized Morison equation, which, again for forces and motions in the vertical plane only, reads:

$$f_x=\frac{1}{2}\rho C_d D\left(v-\frac{\partial z}{\partial t}\right)\cdot\left|v-\frac{\partial z}{\partial t}\right|+\rho\frac{1}{4}\pi D^2\left[C_m\frac{\partial v}{\partial t}-(C_m-1)\frac{\partial^2 z}{\partial t^2}\right] \qquad (2)$$

In which:

ρ = specific density of seawater
Cd = drag coefficient
Cm = inertia coefficient
v = vertical component water particle velocity
D = diameter of the pipeline

For fluid loadings in the horizontal plane, perpendicular to the pipeline, and the resulting displacements similar equations can be formulated. The highest stresses in the pipeline are however caused by waves traveling (approximately) along the longitudinal axis of the pipeline and in that case loadings and motions in the vertical plane are dominant. For most practical cases the effects in the horizontal plane can be neglected therefore.

The solution of equation (1) yields the deflection (in the vertical plane) of each element of the pipe out of the neutral position. From the resulting curvature of the pipe the shear forces and bending moments can easily be deducted, which are, with given sectional properties, translated into dynamic stresses in the pipeline. Figure 6 shows the general configuration of the system, the static deflection of the pipeline during tow and the characteristic dynamic deflection of the pipeline when loaded by regular waves. The envelope of the deflections shows "humps" and "hollows" at both ends of the pipeline. Particularly at the trailing end, where the tension is low, the "tail-wagging" is very distinct. In this graph also a snapshot of the pipeline deflection at an arbitrary moment is shown. A very simple calculation model of the pipeline is shown in figure 7. Methods to achieve such a solution are discussed below.

- Analytical solution.

One can try to solve the equation (1) in an analytical way, as has been done by Langley [2]. This requires a drastic schematization of the system however. Sectional properties and the axial tension must be kept constant over the length of the pipeline, the complete pipeline must be at the same depth (whereas in reality the central part is often 20 to 40 m lower than the tow heads), and the fluid loading must be linearized, which appears to be not a serious restriction. The great advantage of this method of analysis is the limited computational time required. Langley [2] has shown that, despite the schematization, an analytical solution yields reasonable results and it can be of value for a feasibility study or a parameter study in the preliminary phase of a project.

- Numerical solution.

In order to solve the equation (1) for a more complicated model of the pipeline a numerical solution has to be sought. Just as in case of riser systems the pipeline is split up into discrete elements, each with their own mass, flexural rigidity, axial tension, hydrodynamic properties etc.

One possibility is to describe the pipeline with Finite Elements (FEM), see e.g. Verner [2] and Rooduyn and Boonstra [4]. Two dimensional tensioned rod-elements with four degrees of freedom are the most suitable selection as structural elongation of the pipe and torsion do not play a significant role and are neglected in the analysis. This method of solution resembles conventional structural analysis methods.

Another description of the pipeline is achieved by reducing each element to a concentrated mass, which is connected to the adjacent masses by means of springs. The springs represent axial stiffness (which is of no importance for the pipeline, but also can be used to simulate elasticity in the tow wire), bending stiffness and shear. This approach is followed by Van den Boom et.al. [5], and De Boef [6]. This method of modelling, generally referred to as the Lumped Mass Method (LMM), originally is developed for the analysis of mooring lines to which bending stiffness is later added. The method results in very efficient use of computer time as only the diagonal of the matrices for mass and stiffness are filled, whereas in case of the FEM a band around the diagonal of the matrices is created. Tests have shown that the accuracy of the LMM is not inferior to the FEM (see

De Boef et.al. [6]).

- Time Domain Analysis versus Frequency Domain Analysis

Just as in practically all engineering problems in the marine field in which wave loading plays a role, the solution can be sought either in the frequency domain or in the time domain. For the analysis in the frequency domain it is required that the relationship between wave amplitude and resulting effects such as motions and stresses is linear (or can be linearized). As can be seen from equation (2) the wave loading contains the well known quadratic drag part, therefore at least theoretically there exists no linear relationship between the wave amplitude and the fluid loading.

In most cases reported in the literature (e.g. Verner [1], De Boef [6]) the solution is derived in the time domain. Apart from the fact that no linearizations are required, it is also possible to analyze in the time domain conditions which are by definition non-harmonic, such as the behavior of the pipeline after rupture of a tow wire or the transient effects due to changing towing speed, tow wire length etc.. The major disadvantage of the time domain analysis is the computer time required, especially when irregular wave conditions are to be investigated and a simulation time of at least ten or twenty minutes (real time) is required in order to make a realistic estimate of extreme responses.

However it can easily be shown that for a pipeline with a diameter of 0.5 m to 1 m at a depth of say 30 m below the surface in relatively severe waves of 8 m height and 8 s period the fluid force is dominated by inertia (the drag term in the formula of Morison contributes in this condition for less than twenty percent to the total loading, for smaller waves this percentage decreases). This means that the drag term can be linearized without introducing unacceptable approximations. In fact in most cases the drag term in the fluid-structure interaction formula acts as a damping factor, which limits amplification of motions of the pipeline, rather than acting as an excitation force. This can be shown by selecting an unrealistically small value of the drag coefficient Cd in the calculation: the motions of the pipeline and the resulting dynamic stresses will increase instead of decrease.

In a paper by Rooduyn and Boonstra [4] it is shown that for a typical case the results of a time domain analysis and a frequency analysis the resulting stresses are almost the same.

The frequency domain analysis has one disadvantage: it is difficult to take into account in a realistic way the effect of dynamic forces in the tow wires caused by motions of the tugs.

Because the frequency analysis is very efficient in usage of computer time it is well suited to investigate a large number of different conditions such as variations in towing speed, tension in the pipeline, depth of the pipeline below the sea surface, wave direction, seastate etc. The results are used to indicate optimal conditions for those aspects which can be chosen by man. The most severe weather conditions can than be simulated by a time domain analysis.

5. SOME EXAMPLES AND RESULTS

Figure 6 shows a typical dynamic deflection of a towed pipeline in regular waves. The dotted line in the graph represents the envelope of the deflections, the solid line shows one arbitrary position of the pipeline. The result is achieved with a relatively simple model of the pipeline as shown in Fig 7.

Figure 8 shows a more detailed picture of the deflections when calculated in the time domain. It appears that the result is not exactly harmonical: there is a static offset from the original position of the pipeline.

Generally one is not particularly interested in the deflection of the pipeline; shear force, bending moment and resulting stresses are of greater importance. In Figure 9 the envelopes of deflection, shear force and bending moment are plotted, again for regular waves. In this case the calculations have been performed in the frequency domain.

By a systematic variation of one parameter in the calculations it is possible to investigate the effect of that parameter on the behavior of the pipeline. As an example the effect of the vertical position of the towheads below the water surface on maximum bending moment in the pipeline is shown in Fig 10. For this analysis the frequency domain is to be preferred over the time domain because of the smaller computation time required.

In Fig 11 and 12 the behavior of the pipeline is plotted after rupture of a tow wire. This type of analysis can of course only be made in the frequency domain.

REFERENCES

1. E A Verner et.al.:"Predicting Motions of Long Towed Pipe Strings", (OTC 4666), Offshore Technology Conference, Houston, 1984.
2. R S Langley: "Random Dynamic Analysis of Towed Pipelines", Ocean Engineering, Vol 16. No 1, 1989.
3. E J Rooduyn: "Submarine Flowlines: Transportation of Prefabricated Pipelines with the Controlled Depth Tow Method", The Design and Installation of Subsea Systems, (vol 2), London 1985 (ISBN 0-86010-667-5).
4. E J Rooduyn and H Boonstra: "Design Aspects of the Controlled Depth Tow Method for Pipeline Bundles", Deep Offshore Technology Conference, 1985, Sorrento, Italy.
5. H J J van den Boom et. al.:"Dynamic Aspects of Offshore Riser and Mooring Concepts", (OTC 5531), Offshore Technology Conference, Houston, 1987.
6. W J C de Boef et. al.: "Analysis of Flexible Riser Systems", Fifth International Conference on Floating Production Systems, London, December 1989

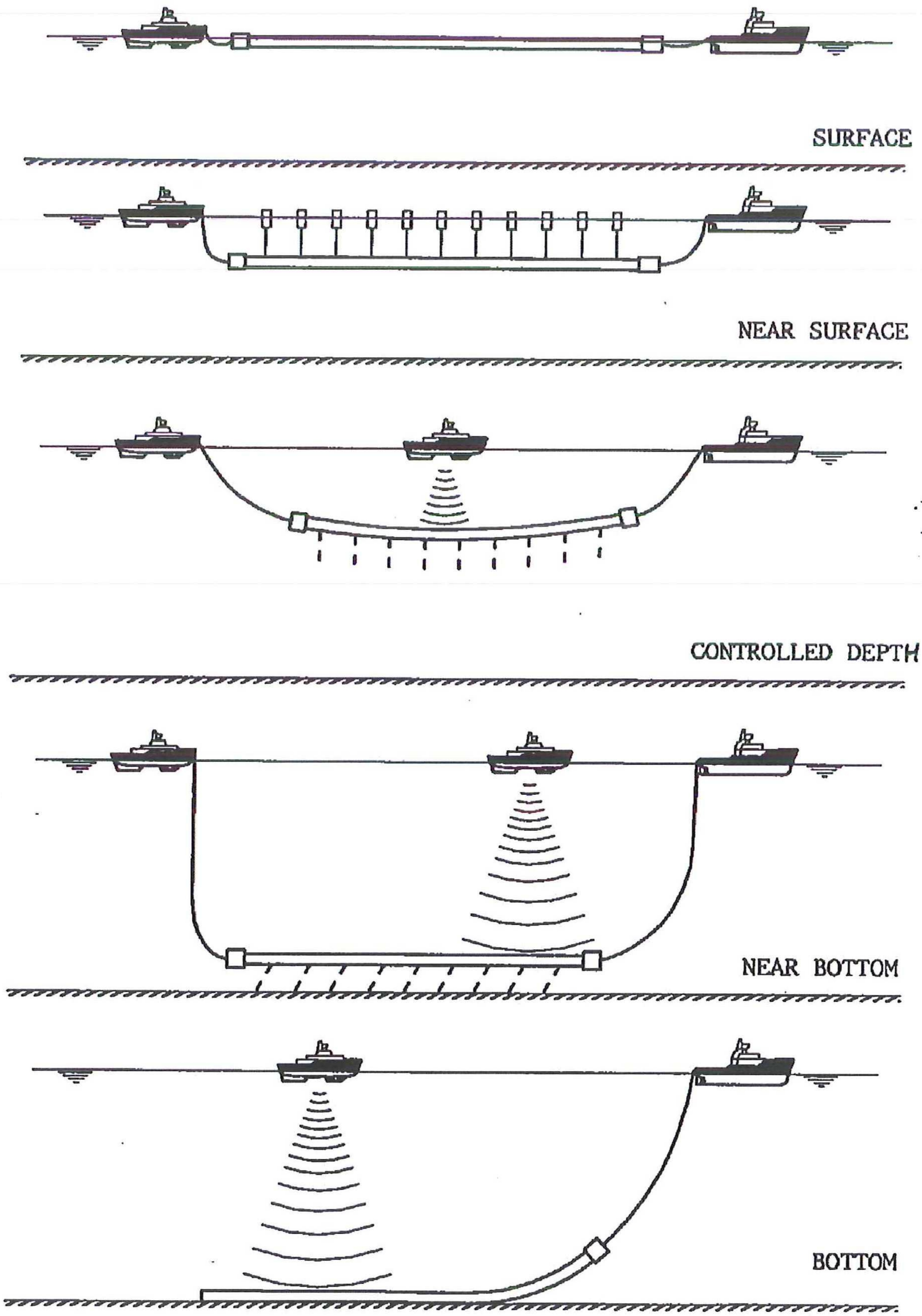

FIG 1. VARIOUS PIPELINE TOWING METHODS

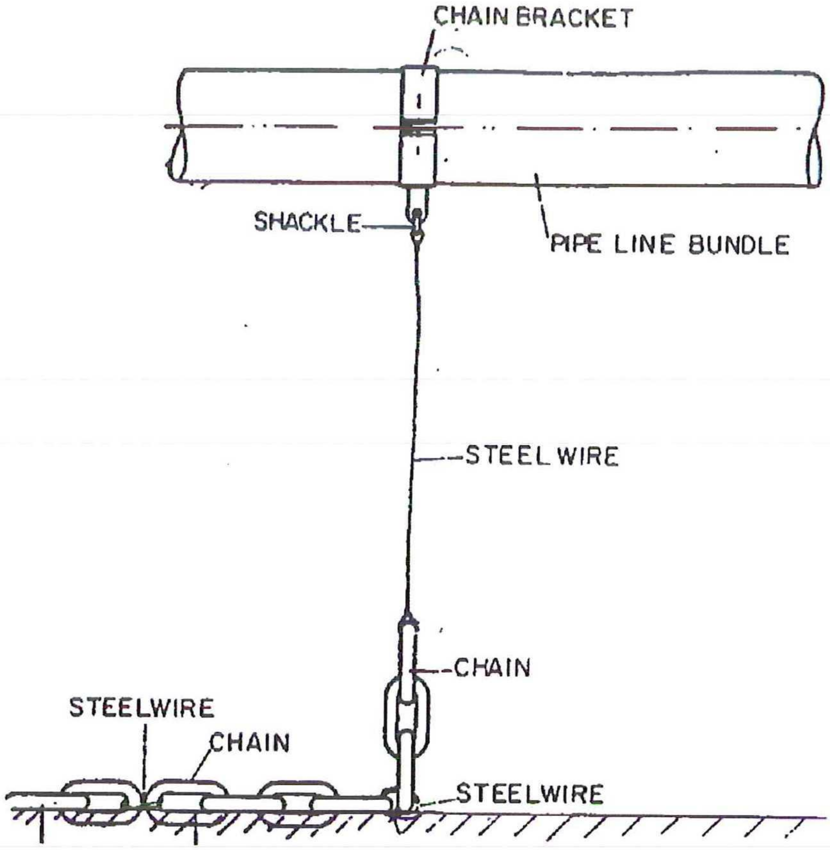

FIG 2. CHAIN SECTIONS TO INCREASE THE SUBMERGED WEIGHT OF THE PIPELINE

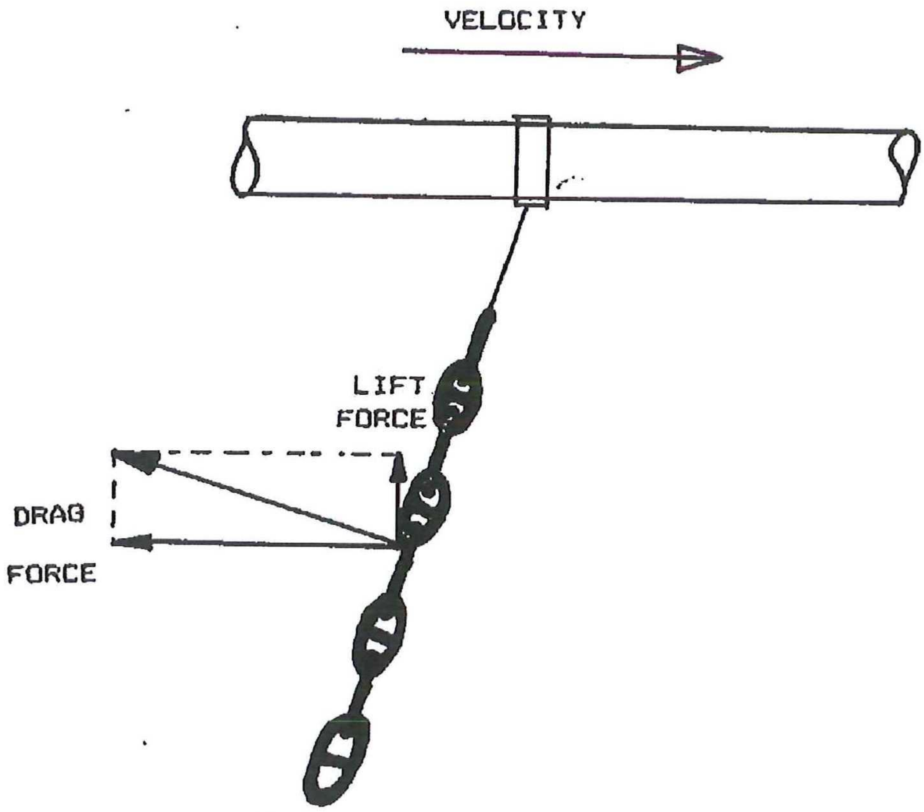

FIG 3. EFFECT OF DRAG FORCE ON CHAIN SECTIONS

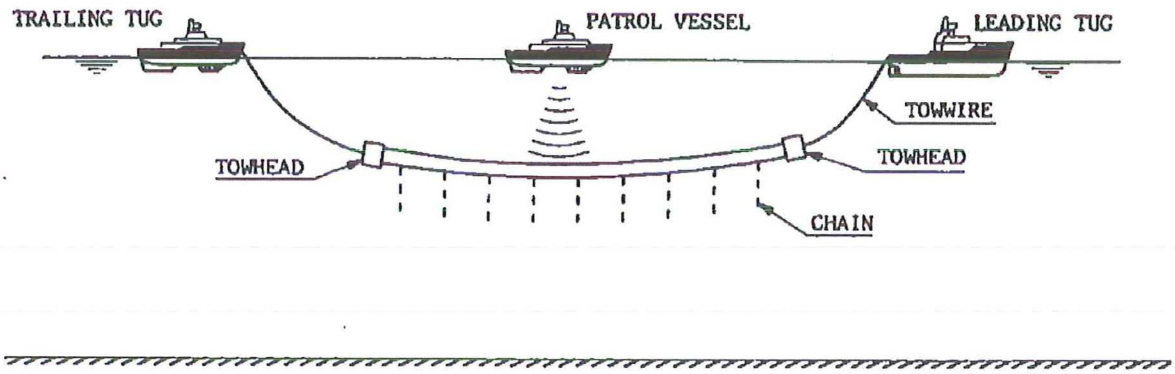

FIG 4. GENERAL ARRANGEMENT OF THE CONTROLLED DEPTH TOW CONFIGURATION

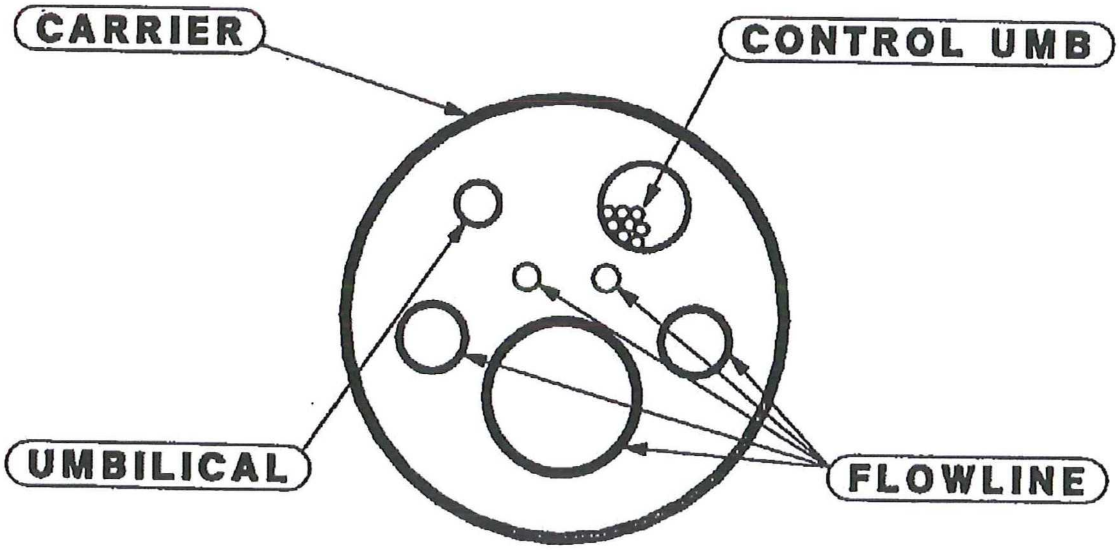

FIG 5. TYPICAL CROSS SECTION OF A PIPELINE BUNDLE

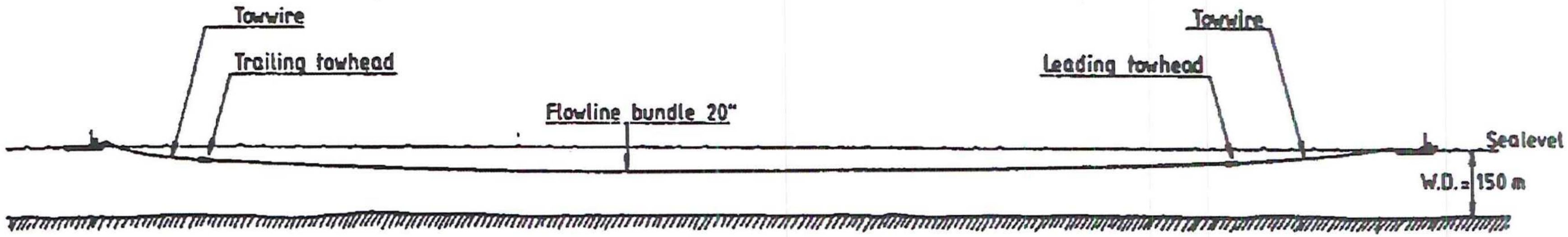

Fig. 6A Configuration of the flow line bundle during tow.

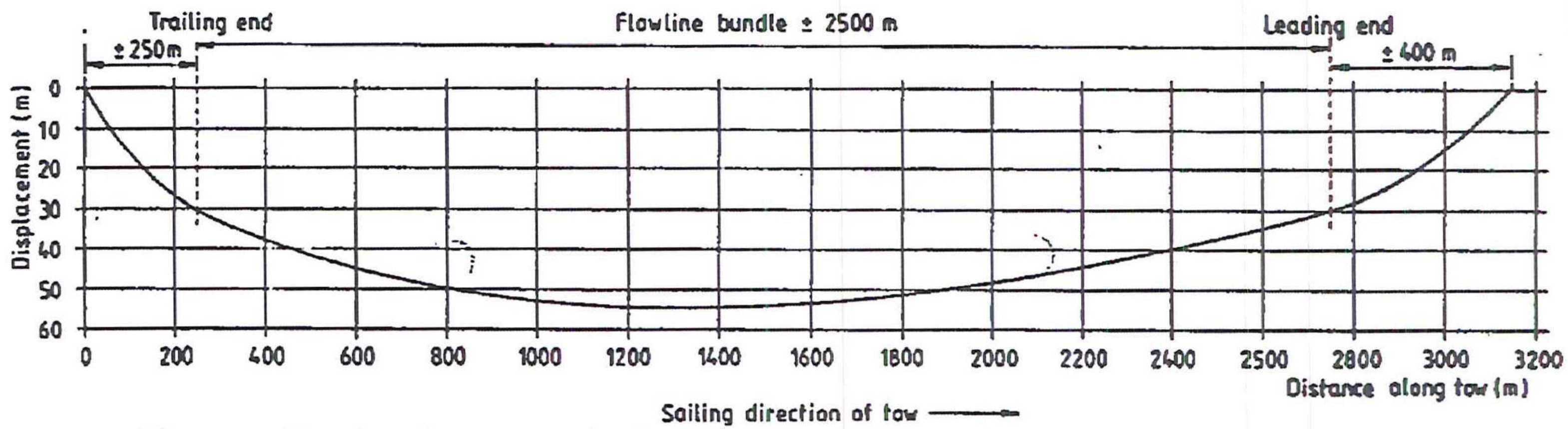

Fig. 6B Static shape of the bundle.

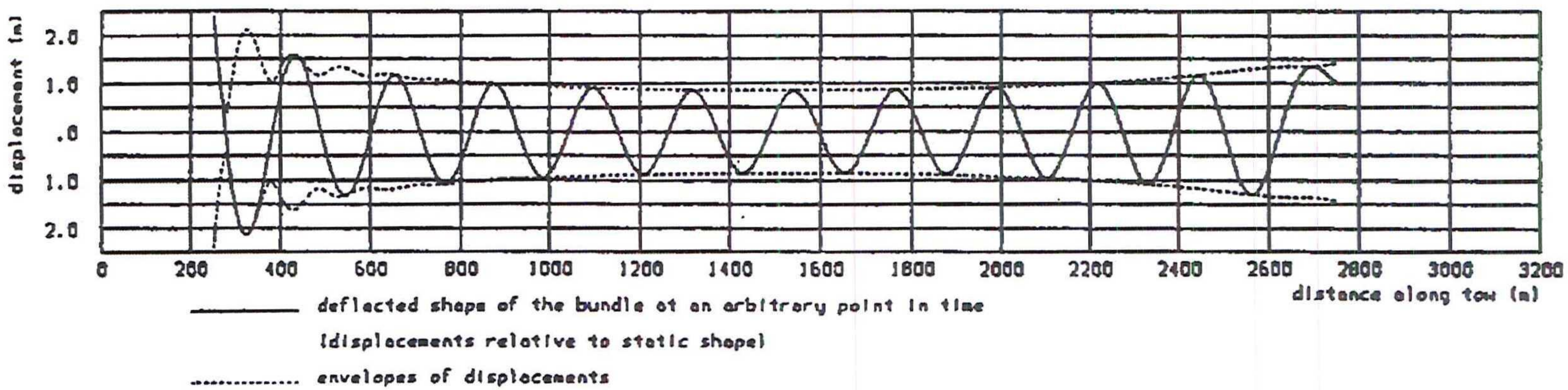

Fig. 6C Deflections of the bundle due to a regular wave.

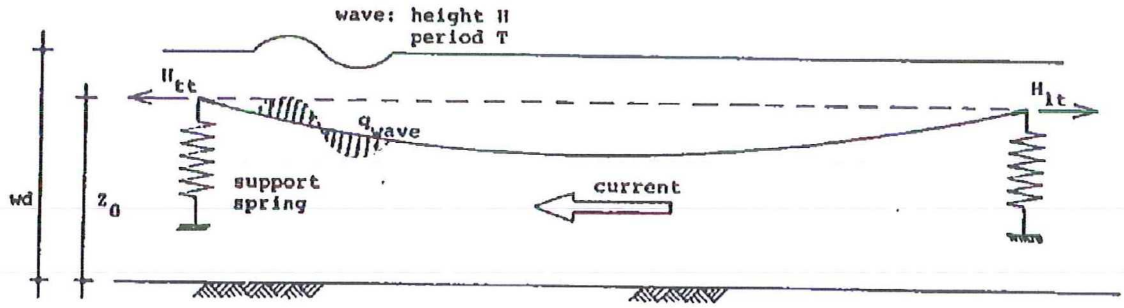

FIG 7. SIMPLE MODEL FOR THE ANALYSIS IN THE FREQUENCY DOMAIN.

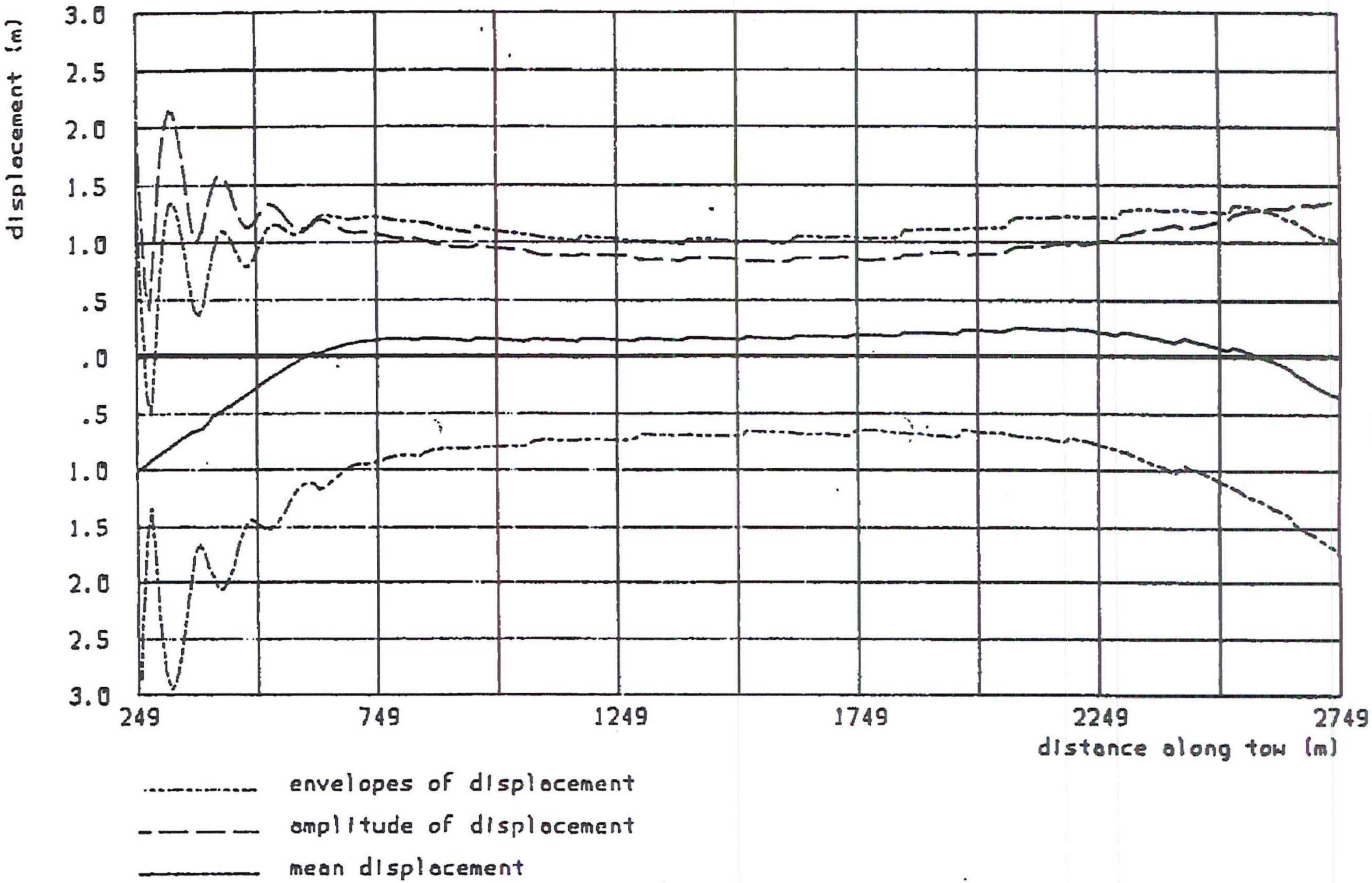

FIG 8. DEFLECTIONS IN THE TIME DOMAIN ANALYSIS.

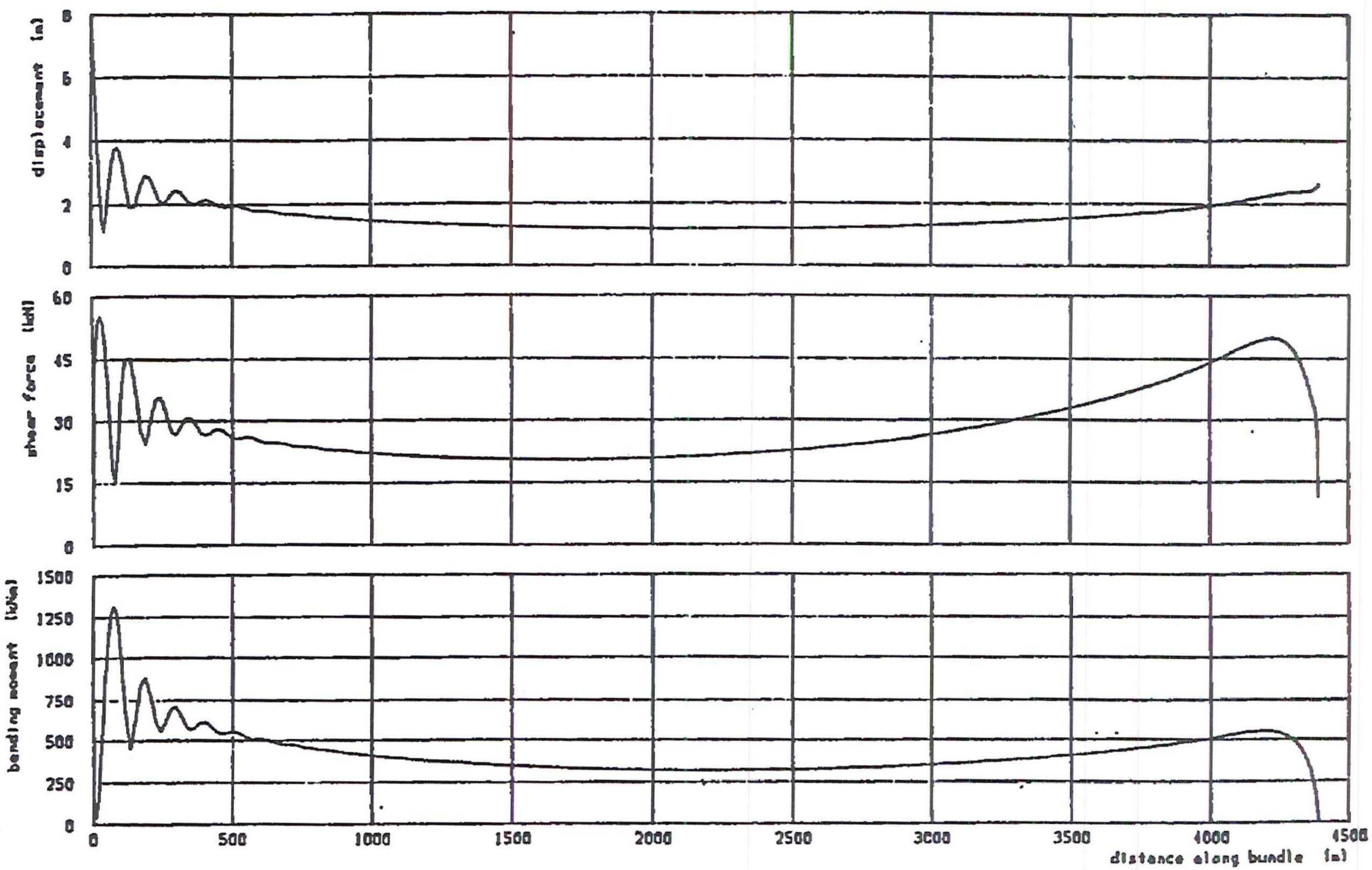

FIG 9. VERTICAL DEFLECTION, SHEAR FORCE AND BENDING MOMENT IN THE PIPELINE DUE TO REGULAR WAVES.

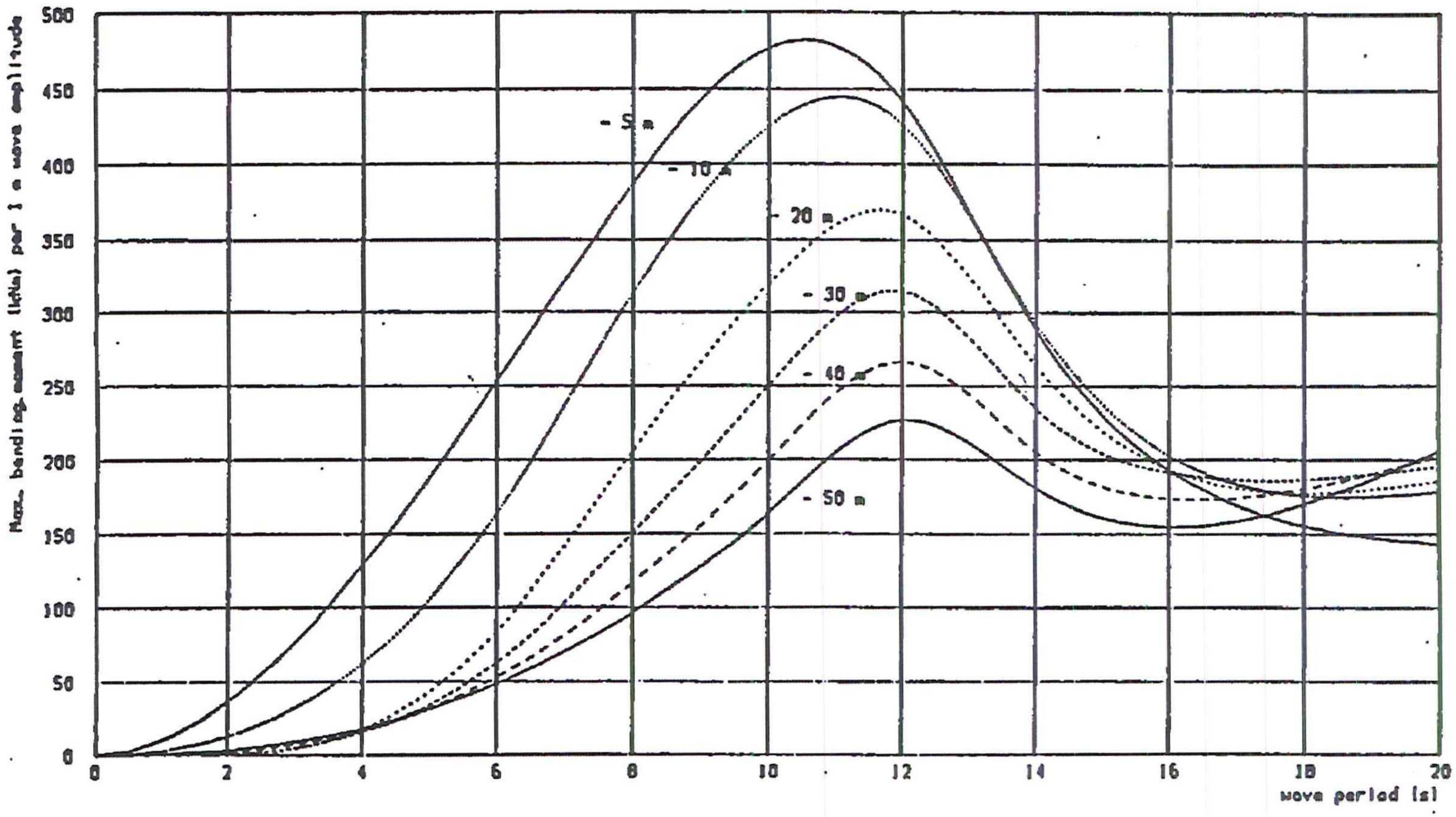

FIG 10. EFFECT OF DEPTH OF TOWHEAD ON BENDING MOMENT IN THE PIPELINE.

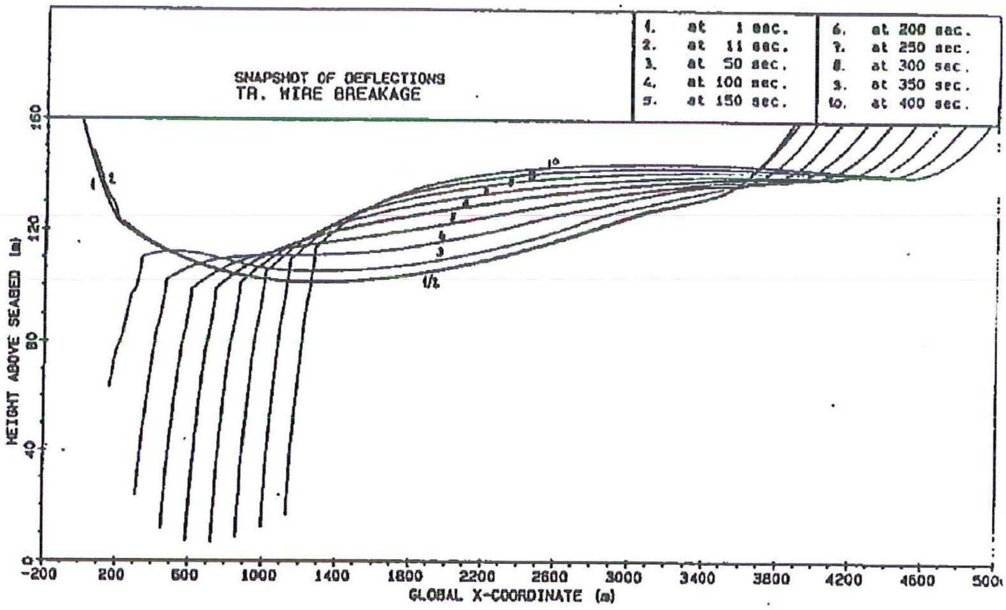

FIG 11. DEFLECTIONS OF THE PIPELINE AFTER FAILURE OF THE TRAILING TOW WIRE.

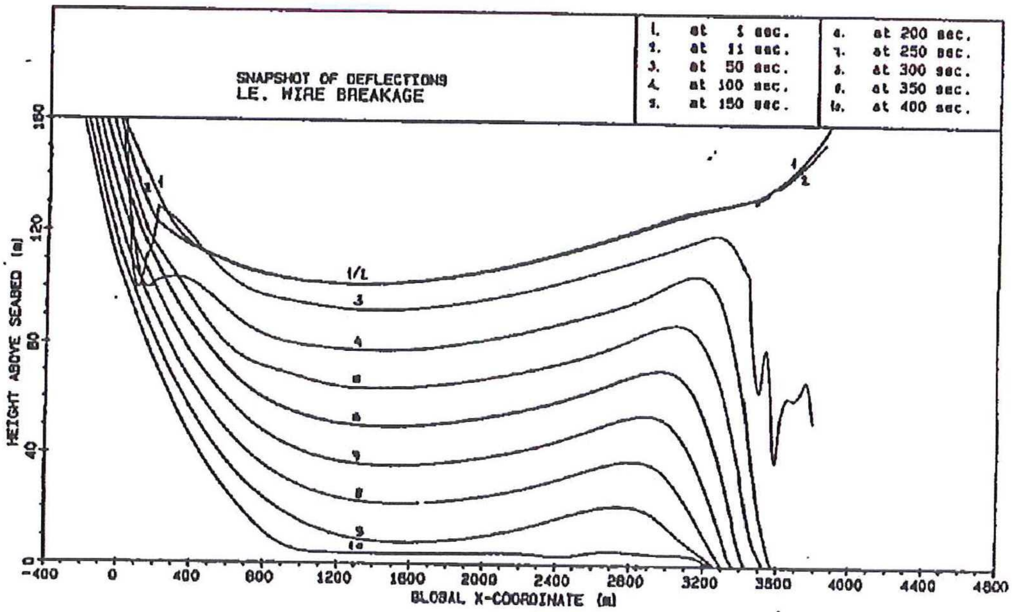

FIG 12. DEFLECTIONS OF THE PIPELINE AFTER FAILURE OF THE LEADING TOW WIRE.