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**SUPPRESSION OF THE VORTEX INDUCED VIBRATION
OF SLENDER CYLINDRICAL STRUCTURES**

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ABSTRACT

This thesis presents the results of a detailed investigation into the performance of devices designed to suppress the vortex induced vibration of slender cylindrical structures.

The phenomenon of vortex induced vibration is explained, followed by a literature review of published work on methods of suppression. The review revealed a lack of knowledge in several key areas relating to the performance of suppression devices. Furthermore, there were no analytical or empirical methods of predicting device performance.

Slender cylindrical structures, such as marine risers, play a crucial role in the recovery of offshore oil reserves. An economic analysis was carried out which showed that vortex induced vibration can cause operational problems which significantly increase the risks and costs involved in offshore oil field exploitation. There was, therefore, an industrial need for more information on suppression devices and a means of predicting their performance.

A large scale experimental investigation was performed to increase the understanding of suppression device performance under a variety of simulated offshore conditions, which included the effects of sheared flow, critical flow and partial suppression device coverage.

Of the three traditional suppression devices tested, the fairing performed best, followed by the helical strake and then the perforated shroud. For the strake and shroud, the relationship between coverage and amplitude suppression was linear. For the fairing, however, the relationship became non-linear. With partial device coverage, the position of the device relative to the cylinder's modal nodes and antinodes was very important.

Sheared flow was found to increase the performance of all three suppression devices. In critical flow, vortex induced vibration of a bare cylinder was found to occur, but at reduced levels. Critical flow increased the performance of the suppression device.

Computer based mathematical models have been developed which are capable of predicting the amplitude of vibration and drag force on bare cylinders, and cylinders fitted with suppression devices, under a variety of ambient conditions. The models were validated against the data obtained from the experimental programme.

Two novel suppression devices have been invented, developed, tested and patented. They offer significant advantages over traditional devices for use in offshore operations.

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NOTATION

Symbols which appear only once in the text are defined where they appear, and are not included in this notation.

Symbol	Description	Unit
A	Area.	m ²
ADC	Analogue to digital converter.	-
ARM	Amplitude response model.	-
ASR	Amplitude suppression ratio.	-
Ca	Cauchy number.	-
C _d	Coefficient of steady drag force.	-
C _d (max)	Maximum coefficient of steady drag force over a range of velocity.	-
C _{d'}	Coefficient of fluctuating drag force.	-
C _{ds}	Coefficient of steady drag force on a stationary cylinder.	-
C _l	Coefficient of fluctuating lift force.	-
C _l (z)	Local coefficient of fluctuating lift force.	-
C _{le}	Effective lift coefficient.	-
c	Coefficient of viscous damping.	-
c	Chord length of fairing or splitter plate.	m
c/t	Chord to thickness ratio of fairing.	-
D	Diameter.	m
DFM	Drag force model.	-
DWF	Deep water flume.	-
E	Young's modulus.	-
E	Error.	-
F	Force.	N
F _l (z)	Local maximum excitation force per unit length.	N/m
F _T	Total drag force.	N
FIV	Flow induced vibration.	-
f	Frequency.	Hz
f(z,t)	Local instantaneous excitation force per unit length.	N/m
f _n	Natural frequency of structural vibration.	Hz
f _v	Frequency at which pairs of vortices are shed.	Hz
G	Gap size.	m
g	Acceleration due to gravity.	m/s ²
I	Second moment of area.	m ⁴
K	Thousand.	-
K _s	Stability Parameter.	-
L	Length of cylinder.	m
LSP	Linear shear velocity profile.	-
M	Million.	-
MEOE	Maximum expected overall error.	-

M_r	Mass ratio.	-
m	Metre.	-
m	Mass per unit length.	kg/m
$m(z)$	Local mass per unit length.	kg/m
m_e	Effective mass per unit length.	kg/m
NPV	Net present value.	-
OAR	Open area ratio.	-
Re	Reynolds number.	-
RVCF	Reduced Velocity correction factor.	-
RVRR	Reduced Velocity response range.	-
SDO	Suppression device offset.	-
SSP	Stepped shear velocity profile.	-
St	Strouhal number.	-
T	Axial tension.	N
t	Time.	s
U	Free stream velocity.	m/s
U_r	Relative velocity between cylinder and free stream.	m/s
U_{ref}	Reference free stream velocity depending on flow profile.	m/s
U.T.S	Ultimate tensile strength.	N/m ²
V_r	Reduced Velocity.	-
W	Work done by fluid on cylinder over one vibration cycle.	J
W_r	Wake response parameter.	-
$w(z)$	Local work done by fluid over one vibration cycle.	J
Y	Maximum cross-flow displacement.	m
$Y_1(z)$	Local maximum cross-flow displacement.	m
Y/D	Reduced Amplitude.	-
$Y_{EFF}(\text{Max})$	Modified Reduced Amplitude.	-
Y_β/Y_o	Normalised shear parameter.	-
y	Cross-flow displacement varying, in general, with t and z.	m
z	Distance along cylinder's longitudinal axis.	m

Greek symbols

α	Velocity ratio used to define shear flow.	-
β	Steepness parameter defining degree of linear shear.	-
δ_s	Logarithmic decrement of structural damping.	-
ζ	Damping ratio.	-
μ	Fluid viscosity.	Ns/m ²
ρ	Density.	kg/m ³
$\psi(z)$	Mode shape.	-
ω	Frequency.	r/s

CHAPTER 1

INTRODUCTION AND OBJECTIVE

1 BACKGROUND.

If a fluid is forced to flow around a solid body, some form of pressure field is inevitably established in the vicinity of the body. In many practical situations this pressure field is unstable, causing the body to experience a time varying force which can lead to a Flow Induced structural Vibration (FIV).

Vortex induced vibration is just one of several different types of FIV, however, it receives more attention than any other type for three reasons:-

- It is ubiquitous. Vortex induced vibration can occur in many different fields of engineering, and over a wide range of operating conditions.
- The mechanisms involved in vortex induced vibration are complex, providing many intellectual challenges; some are now well understood but many remain to be fully explained.
- Vortex induced vibration causes serious operational problems in structures and systems which, if left unchecked, can lead to catastrophic failure. In many situations it is not just an irritating occurrence; it is a problem which demands a practical solution.

The phenomenon is characterized by the periodic formation, and subsequent shedding, of vortices from alternate sides of the trailing edge (or section) of the solid body. The vortices have a diameter of the same order of magnitude as that of the body from which they are shed. Since the pressure in the vicinity of each vortex is relatively low, the alternate shedding results in an alternate pressure force on the body, and it is this alternating force which provides the energy which is necessary to sustain the structural vibration.

Nowadays, vortex induced vibration is usually associated with a piece of manmade equipment and is regarded as a problem to be overcome, but the phenomenon is quite wide spread in the natural world which is, after all, characterized by the interaction of fluids (usually air or water) and solid bodies.

As a result of its natural occurrence, the subject has a very long history although, prior to Strouhal's work published in 1878, it is a history of observed effects with little understanding of the mechanisms involved. For example, the Aeolian harp has been in existence since ancient times and, in the fifteenth century, Leonardo da Vinci made

detailed sketches of the vortices which were formed as a stream of water flowed around a stationary wooden post.

As mentioned above, the scientific study of vortex induced vibration began in 1878 when the now famous Czech scientist, Strouhal, carried out experiments to establish the effect of a number of fundamental parameters on the frequency of vortex induced vibration.

Since Strouhal, many others have researched the phenomenon and today there are many hundreds of references on the subject, some of which are cited in the Chapters that follow. Despite the large amount of work which has been undertaken, our understanding of the subject is still far from complete.

On rare occasions, vortex induced vibration can be beneficial; for example, it has been used as the basis of a fluid velocity meter (Fisher & Porter, 1978), however, in most instances where it occurs within or around a manmade structure it is detrimental and may cause severe operational problems as well as structural failure, usually by the mechanism of fatigue.

Vortex induced vibration is, therefore, regarded as a serious problem to be overcome in many fields of engineering. The occurrence of problems attributable to vortex induced vibration has increased in recent years because of the trend, in many different branches of engineering, toward more efficient and cost effective design with smaller factors of safety. Structural components have tended to become lighter and more flexible, whilst flow velocities have tended to increase (Weaver, 1985).

Such trends are nowhere more apparent than in the offshore oil industry where ever more slender structures are used to exploit oil reserves in increasingly hostile environments. Drilling now takes place in water depths of up to 1000 m and ocean currents in excess of 2 m/s have been encountered. In this type of environment vortex induced vibration is a serious problem which has to be controlled if the oil reserves are to be successfully recovered.

In any kind of problem solving it is obviously important to study the problem itself as well as to study different methods of solution. The majority of work pertaining to vortex induced vibration has been devoted to the problem, rather than to developing and understanding the various solutions; this is particularly true in the area of mathematically modelling. As a consequence, whenever a new vortex induced vibration problem emerges, the method of solution usually relies heavily on physical testing, often using scaled models and prototype structures, rather than on established and reliable analytical techniques. This procedure can be expensive and time consuming.

There are various solution options available to the engineer faced with a vortex induced vibration problem and the type of solution adopted will depend very much on the particular field of engineering in question.

For reasons which are explained in Chapter 3, in the offshore oil industry the solution of serious vortex induced vibration problems are almost always overcome by fitting some kind of fluid dynamic suppression device; helical strakes, perforated shrouds and fairings have all been used.

Due to a lack of coordinated research and analytical model development it is not generally possible to predict the performance of a fluid dynamic suppression device, without resorting to scale model testing.

As a result, the solution of vortex induced vibration problems in the offshore oil industry is often carried out inefficiently with a great deal of wasted time and money.

2 OBJECTIVE.

The primary objective of this project is to increase the current understanding of the behaviour of fluid dynamic suppression devices and develop analytical techniques which will enable device performance to be reliably predicted, without the need for expensive and time consuming model and prototype testing.

It is hoped that, by achieving this objective, the efficiency with which vortex induced vibration problems are solved in the offshore oil industry will be significantly increased.

In order to be of general use, the analytical techniques should be capable of modelling a wide variety of real offshore ocean conditions such as:-

- Sheared current profiles.
- Partial suppression device coverage.
- Sub-critical and critical flow regimes.
- Multi-modal vibrations.

3 OUTLINE OF THE PROJECT.

The strategy which was adopted in order to achieve the project objective involved the following procedure;

- (a) A literature review was carried out to assess the current level of understanding of the phenomenon of vortex induced vibration and its suppression. Thus, the strengths and weaknesses in the technical knowledge pertaining to the suppression of vortex induced vibration were identified. The review is presented in Chapters 2 and 3 of the thesis.

- (b) An analysis of the economic effects of vortex induced vibration on the exploitation of offshore oil reserves was carried out. This served jointly to demonstrate the practical importance of vortex induced vibration and, hence, the industrial need to suppress it, and to provide a means of assessing the merits of the different solution options. The economic analysis is presented in Chapter 4.
- (c) Since the literature review revealed a lack of analytical models capable of predicting the performance of suppression devices, two new models were developed; one to predict the amplitude response and the other to predict drag force. Model development is described in Chapter 5.
- (d) An experimental test programme was devised and carried out to obtain technical information in an attempt to fill the gaps in current knowledge (as identified in 'a' above) and provide vital data for the calibration of the new analytical models. The test programme is described in Chapter 6 and a comparison of experimental results with those predicted by the analytical models is presented in Chapter 7.
- (e) As an alternative means of achieving the objective of improving the efficiency with which vortex induced vibration problems are solved, an investigation into the possibility of using novel designs of suppression device was carried out. Two new devices were developed which offer attractive features for offshore use. The novel devices are described in Chapter 8.

CHAPTER 2

THE PHENOMENON OF VORTEX INDUCED VIBRATION

1 A CLASSIFICATION OF FLOW INDUCED VIBRATION.

Whenever a fluid flows around a solid structure some form of pressure field is established in the vicinity of the structure. For a variety of reasons, the pressure field may contain a significant oscillatory component and, if the structure is sufficiently flexible, this may result in a flow induced structural vibration.

During flow induced vibration, energy is extracted from the flow and used to generate or sustain the structural motion. There are many different physical examples of flow induced vibration (FIV) but, by classifying them according to the nature of the interaction which occurs between the fluid excitation force and the structural motion, it is possible to categorise all types of FIV into just three groups (Weaver 1985):-

- Forced vibration.
- Structurally excited vibration.
- Structurally controlled vibration.

The forces associated with the fluid-structure system can be divided into inertial, structural and fluid-dynamic components. These forces can be represented by the general differential operators I, S and F each of which, in general, will vary with time (t) and structural displacement (y) (Weaver 1985).

The general equation of motion for the structure can be written as:-

$$F(y, t) + S(y, t) = I(y, t) \quad (1)$$

Flow induced vibrations are classified according to the form of the fluid-dynamic force operator, F, as described below.

The distinctive feature of **forced vibration** is that the fluid force does not depend on the motion of the structure. For many practical cases equation 1 becomes:-

$$F(t) + S(y) = I(y) \quad (2)$$

Structural buffeting by a random turbulent flow is a common example of a forced vibration.

For a vibration to be classed as **structurally excited** the fluid force must depend on the structural motion and, further, must completely disappear when the structure is stationary. In practice, equation 1 usually becomes:-

$$F(y, t) + S(y) = I(y) \quad \text{with } F(0, t) = 0 \quad (3)$$

One of the most dramatic examples of structurally excited vibration is the bending-torsion flutter of aircraft wings.

Structurally controlled vibrations are defined as those in which the fluid force exists even when there is no structural motion and, at certain amplitudes of vibration, the structural motion controls the fluid force. In practice equation 1 usually becomes:-

$$F(y, t) + S(y) = I(y) \quad \text{with } F(0, t) \neq 0 \quad (4)$$

Vortex induced vibration is the classic example of this category.

The remainder of this chapter describes and explains the phenomenon of vortex induced vibration, initially by discussing vortex shedding from a stationary cylinder and then by moving on to consider the complicating effects of structural motion.

2 VORTEX SHEDDING FROM A STATIONARY CYLINDER.

The circular cylinder is one of many different bluff body shapes which will shed vortices when placed in a flowing fluid. Because of its significance in engineering structures the circular cylinder has received far more attention than any other body shape in vortex shedding research and is also the subject of this thesis.

2.1 Flow Patterns Around the Cylinder.

Consider the simple case of a smooth, rigid cylinder of infinite length which is positioned with its longitudinal axis orthogonal to the flow. If the flow is uniform and laminar then the independent variables which affect the flow pattern and, consequently, the vortex shedding are:-

- Free stream velocity (U).
- Fluid density (ρ).
- Fluid viscosity (μ).
- Cylinder diameter (D).

The Reynolds number is the only independent dimensionless group which can be formed from these variables and, therefore, it alone is sufficient to fully describe the flow around the cylinder.

There are six distinctive types of flow regimes which can occur around the cylinder (Fig. 2.1). These were first presented by Lienhard (1966) in his review, although the work is attributed to Marris, Foppl, Kovasznay, Tritton and Roshko (see Lienhard, 1966 for references).

At Reynolds numbers below 5 the flow is unseparated, the streamlines are symmetric and appear identical to those of a potential flow. As the Reynolds number increases, however, the streamlines at the rear of the cylinder widen and a stable pair of Foppl vortices are formed.

Between Reynolds numbers of 5 and 40 the Foppl vortices elongate in the direction of flow and eventually become unstable at a Re of about 40. This wake instability invariably affects one of the Foppl pair before the other causing it to be shed from the cylinder. As it leaves, the remaining vortex becomes momentarily stable although after a short period, during which it grows and moves toward the centre rear of the cylinder, it too is shed.

So begins the periodic shedding of vortices although until a Reynolds number of about 90 is reached the periodicity of the vortex shedding is controlled by the wake instability rather than by the shedding process itself.

Throughout the Reynolds number range 0 to 150 the flow remains laminar around the cylinder and in the vortex street. As the Reynolds number rises from 150 there is a transition to turbulent flow which starts downstream in the vortex street but moves upstream if the Reynolds number is increased. The precise point of the transition to turbulent flow is used to define the remaining three flow regimes.

At $Re = 300$ the entire vortex street is turbulent and there is little significant advancement of the transition point until $Re = 3 \times 10^5$ when the transition to turbulence reaches back upstream to the flow separation point on the cylinder causing the hitherto laminar boundary layer to become turbulent. This Reynolds number marks the start of the **critical regime** which continues until $Re = 3.5 \times 10^6$ and is characterised by the **absence** of periodic vortex shedding. The wake narrows, becomes highly turbulent and contains aperiodic pressure fluctuations which cover a wide frequency bandwidth (Lienhard 1966).

When $Re > 3.5 \times 10^6$ the flow regime is **supercritical** and vortex shedding is re-established, although the wake remains narrower than it was in the sub-critical regime.

In the offshore environment the Reynolds number frequently lies in the upper sub-critical and critical range and so these flow regimes are of particular interest.

2.2 The Vortex Shedding Process.

The vortex shedding process can be divided into three parts:-

- Vortex initiation.
- Vortex growth.
- The shedding itself.

The fundamental cause of vortex initiation is boundary layer separation which is a result of fluid friction and the adverse pressure gradient caused by the rear half of the cylinder's surface.

The potential flow of an ideal fluid around an infinitely long cylinder would be streamlined, symmetrical and would contain no vorticity (Fig. 2.2). Vortex shedding would not occur, indeed, the cylinder would not generate any form of wake and would not experience any fluid drag, regardless of the Reynolds number.

A single streamline of such a flow is shown in Fig. 2.3a. In moving from point A to B the fluid loses pressure energy but gains an equal amount of kinetic energy. From B to C this energy transformation is reversed and, therefore, the fluid velocity at C is the same as that at A.

Replacing the ideal fluid with a real one causes a boundary layer to form within which energy is lost due to fluid friction. Consequently, somewhere between points B and C the fluid in the boundary layer loses all its kinetic energy and comes to rest causing the boundary streamline to separate from the cylinder (point D, Fig. 2.3b).

Point D lies in a region which has an adverse pressure gradient and, therefore, the fluid immediately downstream of D will have a higher static pressure. This causes flow reversal to occur just downstream of the separation point and a vortex is initiated which marks the start of the cylinder's wake. Prandtl & Tietjens (1957) photographed this phenomenon by seeding the fluid with fine particles which show up under illumination and give an indication of the local fluid velocity (Fig. 2.4).

Vortex growth and the structure of the cylinder's wake depends on the Reynolds number (Section 2.1), however, over a wide range of Reynolds number large vortices develop and are subsequently shed periodically from alternate sides of the cylinder to produce the well known Karmen Vortex Street (Fig. 2.5).

The mechanisms which are involved in the development of the vortex street are complex and, even after a century of research, they are not completely understood although there have been many attempts at providing a physical description.

In a recent review Bearman (1984) includes a detailed description of vortex growth and shedding which emphasises the importance of the mutual interaction between the two separated shear layers.

A vortex in the cylinder's near wake grows as it receives circulation from the free shear layer to which it is attached. Growth is halted only when the vortex becomes large enough to capture the shear layer on the opposite side of the cylinder. This shear layer contains circulation of opposite sign which acts to sever the vortex from its own free shear layer with the result that the vortex is shed and moves off downstream.

2.3 Vortex Shedding Frequency and Correlation Length.

The scientific study of vortex shedding began in 1878 when Strouhal discovered that the frequency of vortex shedding, f_v , could be expressed in terms of the free stream velocity, U , and the cylinder diameter, D :-

$$f_v = \frac{St \cdot U}{D} \quad (5)$$

In general the Strouhal number, St , depends on the Reynolds number (Fig. 2.6), however, over a wide range of Reynolds number the Strouhal number is constant and has a value of 0.21. Note that the numbers one to six marking ranges of Reynolds number in Fig. 2.6 correspond with those of Fig. 2.1. A comparison of these two figures reveals the strong connection between the flow pattern around the cylinder and it's Strouhal number.

One might expect that, provided the incident flow was uniform, the flow pattern around the cylinder would not vary along it's length and would, therefore, be two dimensional. In fact this is not the case as vortices are shed in cells. Although the Strouhal number and, therefore, the vortex shedding frequency associated with each cell is constant, there are differences between the phase of the shedding from each cell.

The length of each cell is known as the correlation length and it depends primarily on the Reynolds number, as summarised by King (1977).

Over a wide range of Reynolds number (150 to 100000) the correlation length lies between 2 and 3 cylinder diameters.

2.4 Forces on the Cylinder due to Vortex Shedding.

The periodic shedding of vortices creates an asymmetric and periodic pressure distribution around the cylinder. This is illustrated in Fig. 2.7 which shows the pressure distribution around the cylinder at three sequential instants in time which together cover half the vortex shedding period.

Pressure at the rear of the cylinder is lowest in the vicinity of the attached vortex and rises locally when the vortex is shed.

The cylinder is, therefore, subjected to a resultant force, F , which fluctuates periodically in both direction and magnitude. It is apparent from Fig. 2.7 that when F is resolved in-line with the flow it contains both steady and fluctuating components:-

$$F_{dt} = F_d + F_{d'} \quad (6)$$

Where:

F_{dt} = Instantaneous total value of the in-line force.

F_d = Steady component of the in-line force.

$F_{d'}$ = Fluctuating component of the in-line force.

The steady and fluctuating drag coefficients are defined :-

$$C_d = \frac{2F_d}{A \cdot \rho \cdot U^2} \quad (7)$$

$$C_{d'} = \frac{2F_{d'}}{A \cdot \rho \cdot U^2} \quad (8)$$

Where:

A = Frontal area of cylinder.

ρ = Fluid density.

U = Free stream velocity.

Resolving F in the cross-flow direction and repeating the above procedure gives:-

$$F_{1t} = F_1 + F_{1'} \quad (9)$$

$$C_1 = \frac{2F_1}{A \cdot \rho \cdot U^2} \quad (10)$$

$$C_{1'} = \frac{2F_{1'}}{A \cdot \rho \cdot U^2} \quad (11)$$

The vortex shedding frequency, f_v , is defined as the frequency at which pairs of vortices are shed. The lift force therefore fluctuates at the vortex shedding frequency, however, the drag force fluctuates at twice this frequency because its periodicity is associated with individual rather than pairs of vortices.

It is usual to express the fluctuating force components of both lift and drag in terms of their r.m.s. value. The steady component of lift is zero unless an asymmetry is introduced in the flow or structure. For this reason the steady component of lift is commonly ignored, and the fluctuating component is denoted simply by C_l , rather than by $C_{l'}$. This practice has been adopted in all subsequent chapters of the thesis.

Because of the intimate connection between the Reynolds number and the flow patterns around the cylinder it is not surprising that the Reynolds number has a profound effect on the value of the lift and drag coefficients. Figs. 2.8 to 2.10 show how the coefficients C_d , $C_{d'}$ and C_l vary with Reynolds number. The importance of the three flow regimes (sub-critical, critical and super-critical) on the value of the coefficients can be clearly seen.

2.5 Complicating Factors.

The discussion so far has centred upon a very simple and, in some ways, idealised situation; the cylinder has been of infinite length, it has had a smooth exterior surface and it has been positioned normal to the flow. The free stream flow has been of uniform velocity and non-turbulent. In a marine environment many of these conditions may not exist and it is, therefore, useful to discuss their effect on vortex shedding and on the fluid loading on the cylinder.

2.5.1 Cylinder end effects and aspect ratio.

The presence of a cylinder end causes three dimensional flow to occur in the vicinity of the end which reduces the Strouhal number by an amount which depends on the cylinder's aspect ratio (i.e. L/D). Gowda (1975) found that with an Aspect Ratio of 50 or more the end effect was insignificant but at lower Aspect Ratios the Strouhal number was reduced. For example, with an Aspect Ratio of 15 the Strouhal number was 35% lower than for the infinite cylinder case.

2.5.2 Sheared incident flow.

The effect of a sheared incident flow profile where the plane of the shear is parallel to the cylinder's axis is of practical importance. For example, this situation often arises in offshore engineering where marine risers (vertically positioned cylindrical structures) are subjected to ocean currents which vary with depth.

Shear flow contains vorticity with an axis of rotation which is normal to the shear plane, whereas the vortices created by the cylinder have their axis parallel to the cylinder and, hence, parallel to the shear plane. This results in a complicated three dimensional interaction between two sets of orthogonal vortices.

An additional complication arises because, when the flow is sheared, the velocity and hence the stagnation pressure at the front and rear of the cylinder varies along the length of the cylinder. This causes axial flow along the front and back of the cylinder (Masch & Moore 1960).

The degree of shear is characterised by the steepness parameter, β as defined in the following equation:-

$$\beta = \frac{D}{U_{ref}} \times \frac{dU}{dz} \quad (12)$$

Where:

- D = Cylinder diameter.
- U_{ref} = Reference velocity, usually defined as the mid-span or mean value.
- dU/dz = Velocity gradient.

In a recent review Griffin (Sept. 1985) has summarised the work which has been carried out to investigate the effects of shear on the flow around circular cylinders. This work has been entirely experimental due to the complexity of the highly three dimensional flow. Many of the experiments were conducted with low aspect ratio cylinders ($L/D < 20$) rendering the results of limited use but, despite this, some trends have become apparent.

The presence of shear in the incident flow decreases the vortex shedding cell length and tends to reduce the definition of cells and their boundaries. Within each cell the Strouhal number and base pressure coefficient remain constant provided they are based on the local velocity.

2.5.3 Surface roughness and free stream turbulence.

The flow regime around a cylinder has been shown to depend on the Reynolds number. In physical terms it is the position and nature of flow separation which determines the flow regime and both these factors are influenced by the surface roughness of the cylinder and the free stream turbulence as well as by the Reynolds number.

Fage & Warsap (1929) found that an increase in free stream turbulence translated the C_d - Re curve toward a lower Reynolds number without appreciably altering its form (Fig. 2.8). This effect was due to the destabilising influence of the turbulence on the

boundary layer.

Miller (1976) conducted tests on sand roughened cylinders and found that roughness translated the C_d - Re curve toward a lower Reynolds number due to the destabilising influence of the rough surface on the boundary layer. Roughness also increased values of C_d , particularly in the critical and sub-critical regimes. This was due to the increase in skin friction, the increase in cylinder diameter and to the progressive disappearance of the separation bubble which is associated with the large reduction of drag which occurs during the critical flow regime.

3 VORTEX SHEDDING FROM A FLEXIBLE CYLINDER.

In the previous section it was shown that vortex shedding causes a stationary cylinder to experience a periodically fluctuating force which has components in both the in-line and cross-flow direction. If the cylinder is sufficiently flexible then this force may produce a flow induced vibration.

If the amplitude of vibration is small (typically less than 0.05 diameters) then the motion does not significantly affect the fluid force and a simple **forced vibration** occurs, however, for larger amplitudes the motion of the cylinder has a strong influence on the fluid force and the vibration becomes **structurally controlled**.

The hydroelastic interaction that occurs between the structure and the fluid is non-linear and makes the vortex shedding phenomenon an order of magnitude more complex than it is for the stationary cylinder case.

3.1 The Effects of Vibration on the Vortex Shedding Process.

3.1.1 Hydrodynamic forces on the cylinder.

The fluctuating force exerted by vortex shedding from a stationary cylinder may be sufficient to initiate a vibration but it is not large enough to sustain a vibration of significant amplitude (Sarpkaya, 1979). Large amplitude vibrations occur because the fluctuating force is amplified by the motion of the cylinder.

A vibrating cylinder forms stronger vortices than a stationary one and the formation occurs closer to the rear of the cylinder. Both these effects increase the hydrodynamic forces on the cylinder (Sarpkaya 1979).

The mean drag coefficient is increased by the vibration, sometimes by as much as 270%. Griffin (June 1985) has analyzed the results of many independent investigations and developed an empirical formula for predicting the mean drag on a cylinder undergoing vortex induced vibration:-

$$\frac{C_d}{C_{ds}} = 0.124 + 0.933 W_r \quad (13)$$

Where:

C_d is the drag on the vibrating cylinder.

C_{ds} is the drag on an equivalent stationary cylinder.

W_r is the wake response parameter as defined by the following equation:-

$$W_r = \frac{(1 + 2Y/D)}{St \cdot V_r} \quad (14)$$

(This method of predicting the drag amplification due to vortex induced vibration is further discussed in Chapter 5.)

The effective fluctuating lift coefficient also increases with amplitude until it reaches a peak value of about 0.5 when the Reduced Amplitude is about 0.6. Further increase in amplitude reduces the lift coefficient until it becomes zero at a Reduced Amplitude of between 1.5 and 2.0 (Fig. 2.11, King (1974), See also Chapter 5, section 3.3.3)

Vortex induced vibration is, therefore, self limiting and, no matter how compliant and lightly damped the cylinder may be, its Reduced Amplitude will not appreciably exceed 2.0.

Approximately ninety percent of the total amplification in fluctuating force on the vibrating cylinder is due to the increase in vorticity close to the rear of the cylinder as described above; the other ten percent is caused by an increase in correlation length (Sarpkaya 1979).

3.1.2 Correlation length.

Even small amplitudes of vibration cause a significant increase in correlation length, particularly when the flow has a low turbulence intensity, as shown in the table below:-

Reduced Amplitude	Correlation length (diameters)	
	Low turbulence	High turbulence
0.05	3.5	2.5
0.10	40.0	10.0

From Sarpkaya (1979) $Re = 1.9 \times 10^4$

The minimum Reduced Amplitude required to cause a significant increase in correlation length is termed the threshold amplitude (King, 1977). For cross-flow vibration the threshold amplitude lies in the region 0.05 to 0.10.

For in-line vibration the threshold amplitude is almost an order of magnitude smaller, lying in the range 0.01 to 0.02 (King, 1977).

3.1.3 Lock-on.

For a stationary cylinder the frequency of vortex shedding (f_v) varies linearly with free stream velocity according to the Strouhal relationship (Fig. 2.12), however, this is not the case for a cylinder which is undergoing vortex induced vibration. In the latter case as the velocity increases and f_v nears the cylinder's natural frequency (f_n) the cylinder will start to vibrate.

If the velocity is increased further the cylinder will continue to vibrate and shed vortices at a frequency close to f_n , thus departing from the Strouhal relationship. This phenomenon is known as lock-on and continues over a range of velocity which is dependent on the amplitude of response. For large amplitude vibrations lock-on continues until $f_v = 1.4 f_n$ (King 1977, Sarpkaya 1979).

At the end of the lock-on range the frequency of vortex shedding will return in a discontinuous jump to its natural value given by the Strouhal relationship or to the start of another lock-on range if there is another structural natural frequency close by.

3.2 Hydroelastic Parameters.

One dimensionless group (the Reynolds number) is sufficient to determine the nature of vortex shedding from a stationary cylinder. The presence of oscillatory motion complicates this simple relationship considerably and introduces several additional independent variables.

Consider the simple case of a smooth, infinitely long cylinder which is completely immersed in a non turbulent flowing fluid and is positioned as shown in Fig. 2.13. The independent variables which affect the dynamic behaviour of the cylinder are :-

- Free stream velocity (U).
- Fluid density (ρ).
- Fluid viscosity (μ).
- Cylinder diameter (D).
- Cylinder density (ρ_c).
- Modulus of elasticity (E).

Free surface and gravitational effects are not relevant to the particular case under

consideration.

Dimensional analysis of these variables leads to the following equation (King 1974, Walshe 1967):-

$$\frac{Y}{D} = f(Re, Ca, \delta_s, M_r) \quad (15)$$

In equation 15, δ_s is the logarithmic decrement of structural damping. The Cauchy number (Ca) and the Mass ratio (M_r) are defined below.

$$Ca = \frac{E}{\rho \cdot U^2} \quad (16)$$

$$M_r = \frac{m_e}{\rho \cdot D^2} \quad (17)$$

Note that m_e is the effective mass per unit length of the cylinder and includes the hydrodynamic added mass.

If the cylinder vibrates in a normal mode it can be represented by a single degree of freedom mass-spring-damper system. Let the natural frequency of the system be f_n and let the periodic vortex shedding force be represented by a harmonic excitation force with a frequency f_v .

In addition to the fluid excitation force, there are three other forces which act on the cylinder:-

- The dissipative damping force.
- The mechanical (spring) restoring force.
- The inertia force.

The relative importance of the non-dimensional groups in equation 15 depends on the frequency ratio f_v/f_n and on the value of Y/D .

Whilst the amplitude of vibration remains below the threshold level the fluid excitation force depends on the Reynolds number which, as described above, can affect the flow regime around the cylinder and the correlation length.

This dependence on Reynolds number is, however, reduced when the cylinder vibrates at higher amplitudes (King 1974). Although regular vortex shedding from a stationary

cylinder ceases during the critical regime this is not the case for a vibrating cylinder. Humphries & Lamb (1986) carried out tests on two large scale cylinders (6m long x 0.076m diameter) and found that vortex induced vibration persisted through the critical regime.

The Cauchy number is a measure of the relative importance between the fluid excitation force and the mechanical force which acts to restore the cylinder to its equilibrium position. When f_v/f_n is less than 1 the excitation force is balanced principally by the mechanical restoring force (Den Hartog 1947) and the Cauchy number is, therefore, important in determining the behaviour of the cylinder.

At resonance ($f_v = f_n$) the mechanical restoring force and the inertia force become unimportant because they cancel one another out and the fluid excitation force is balanced by the damping force alone. Scruton (1963) showed that the amount of energy dissipated by damping is dependent on the Stability Parameter (K_s) which is an amalgamation of the mass ratio and the logarithmic decrement of damping:-

$$K_s = \frac{2 \cdot m_e \cdot \delta_s}{\rho \cdot D^2} \quad (18)$$

King (1974) has shown that the maximum amplitude response due to vortex induced vibration depends on the Stability Parameter, however, at low values of mass ratio the response is affected separately by the structural damping and the mass ratio (Bearman 1984 and Sarpkaya 1979).

Above resonance ($f_v > f_n$) the fluid force is balanced principally by the inertia force and, therefore, the mass ratio is important in determining the level of vibration that occurs.

The frequencies f_v and f_n are not independent variables and, therefore, are not represented in the dimensional analysis presented above, however, the frequency ratio clearly has a profound effect on the vortex induced vibration phenomenon. The frequency ratio can be written :-

$$\frac{f_v}{f_n} = \frac{St \cdot U}{f_n \cdot D} \quad (19)$$

The group ($U/f_n D$) is known as the Reduced Velocity (V_r) and, since the Strouhal number is relatively constant over a wide range of Reynolds number, the Reduced Velocity has an important affect on the cylinder's dynamic response. The natural frequency used to calculate the Reduced Velocity is that for the case where the cylinder is immersed in the stationary working fluid and is, therefore, affected by the hydrodynamic added mass.

In summary, therefore, there are four dimensionless groups which are important in determining the behaviour of a cylinder undergoing vortex induced vibration:-

$$\frac{Y}{D} = f(Re, Ca, K_s, V_r) \quad (20)$$

For low values of mass ratio, however, the Stability Parameter must be expanded into its two constituent dimensionless numbers, the mass ratio itself and the logarithmic decrement of structural damping.

3.3 Modes of Vibration.

Vibration due to vortex shedding can have components in both the in-line and cross-flow directions. Pure in-line motion can occur with two different flow regimes which correspond with two different ranges of Reduced Velocity (King, 1977).

Although vortices are never shed in pairs from a stationary cylinder, this is possible when the cylinder is flexible. King (1974) has shown that in-line motion associated with the shedding of pairs of vortices occurs over the Reduced Velocity range $1.25 < V_r < 2.5$. Pure in-line vibrations can also be excited by alternate vortex shedding over the Reduced Velocity range $2.7 < V_r < 3.8$.

Pure in-line vibrations never exceed an amplitude of 0.2 diameters and they only occur when the cylinder has a low value of Stability Parameter (King 1974, Fig. 2.14).

Cross-flow vibrations (in water) can occur over a wide range of Reduced Velocity ($4.5 < V_r < 10$) and can reach amplitudes of up to 2.0 diameters. Maximum amplitudes fall in the range $6.5 < V_r < 8.0$ (King, 1977). Significant levels of vibration can occur over a much higher range of Stability Parameter than is the case for in-line vibrations (Fig. 2.15). During cross-flow vibrations there is an associated in-line component at twice the frequency, and this coupled motion appears as a figure of eight when viewed in plan.

In general, cross-flow vibrations are of far more practical importance than in-line vibrations and this is also true in the offshore engineering field.

Note that, in subsequent chapters, the range of Reduced Velocity over which significant cross-flow vortex induced vibrations occur is denoted by the term 'Reduced Velocity Response Range' (RVRR).

3.4 Complicating Factors.

3.4.1 End effects and aspect ratio.

The presence of a cylinder end decreases its added mass and disrupts the vortex shedding by introducing a third dimension to the flow, however, King (1985) has shown that these effects are only significant for aspect ratios below five.

Aspect ratio affects the nature of a cylinder's vortex induced vibration indirectly via its affect on damping and the cylinder's stiffness and natural frequencies. For a particular mode and amplitude of vibration, the logarithmic decrement of damping decreases as the Aspect ratio increases.

High Aspect ratio cylinders tend to be very compliant, and will have a large number of closely spaced natural frequencies, making them susceptible to vortex induced vibration.

3.4.2 Multi-modal vibration.

Lock-on occurs (in water) over a Reduced Velocity range of 4.5 to 10. If a cylinder has closely spaced natural frequencies it is possible that a given flow velocity will produce lock-on in two or more modes simultaneously.

Little work has been done to investigate the multi-modal vibration of cylinders, however, marine cables are known to suffer from this kind of vibration and have been the subject of much research.

3.4.3 Sheared incident flow.

Recent reviews by Humphries (1987) and Griffin (Sept. 1985) highlight the lack of work which has been carried out to investigate the effects of shear on the vortex induced vibration of flexible cylinders and the consequent incomplete understanding of the phenomenon.

Of the small amount of research which has been carried out, much is of limited use because it involved arbitrary values of shear parameter, low Reynolds number and cylinders of low Slenderness Ratio and, therefore, significant end effects. Furthermore, most of the experiments have been conducted with a cylinder constrained to vibrate under controlled conditions which may not be representative of a freely vibrating cylinder.

Nevertheless, some interesting facts have emerged. The strong vortex correlation which exists along the length of a vibrating cylinder in a uniform flow is reduced when the flow is sheared, but vortices are still shed in cells.

The lock-on phenomenon and associated large amplitude vibration response can still occur when the flow is sheared. Woo et al (1983) found that when a slender cable vibrated in a sheared flow the frequency spectra of the vortex shedding contained peaks corresponding to the structural natural frequencies and to the natural vortex shedding frequency.

Humphries (1987) found that shear flow decreased the maximum amplitude of vibration but slightly increased the range of Reduced Velocity over which the vibrations occurred. The shear parameter, β , was used to quantify the degree of shear and its effect on vibration amplitude is shown in Fig. 2.16.

3.4.4 Surface roughness and free stream turbulence.

From the limited amount of work which has been carried out it appears that the effects of free stream turbulence on vortex induced vibration are small. Walshe (1967) found that increasing turbulence brought about an early transition to the critical flow regime which increased the Strouhal number and consequently shifted the range of Reduced Velocity over which vortex induced vibration occurred. Humphries (1987) compared the vortex induced vibration of hydroelastically similar cylinders at two different scales and two different turbulence intensities (1 % and 5%). The results were similar, indicating that the effects of turbulence were not significant.

In tests on an artificially roughened cylinder to simulate marine growth Lamb & Walker (1987) found that roughness decreased the amplitude of vortex induced vibration. It was suggested that the reduction was due in part to a disruption of the boundary layer separation points and also to the promotion of super-critical flow brought about by the rough surface.

CHAPTER 3

THE SUPPRESSION OF VORTEX INDUCED VIBRATION

1 CONTROLLING VORTEX INDUCED VIBRATION.

The amplitude of vortex induced vibration in a cylindrical structure depends, in general, on four non-dimensional groups (chapter 2, section 3.2):-

- Reynolds number.
- Cauchy number.
- Reduced Velocity.
- Stability Parameter.

In theory, vortex induced vibration can, therefore, be controlled by manipulating any of these groups, without having to alter the fundamental design of the cylindrical structure. In the field of offshore engineering, however, it is usually impossible to achieve a satisfactory level of control in this way. The reasons for this are explained below.

In the offshore environment Reynolds numbers generally lie in the upper sub-critical or critical regime and, since vortex induced vibration persists throughout these regimes, the Reynolds number cannot be used as an effective means of control. In addition, the only influence that the offshore engineer has over the Reynolds number is via the cylinder diameter, but this is fixed by operational considerations (for example, the standard diameter for a marine drilling riser is 21 inches).

Where the Reduced Velocity lies in the range 4.5 to 10 there is, by definition, a close match between the natural frequency of vortex shedding and the structures natural frequency. Under this condition of resonance the Cauchy number has no influence on the vibration of the cylinder (Chapter 2, section 3.2) and cannot, therefore, be used as a means of control.

In a deep water riser Reduced Velocity can be controlled, via the risers natural frequency, by manipulating the axial tension. The riser will have closely spaced natural frequencies which means that lock-on may occur over a wide range of velocity. The only way to reliably de-tune this kind of multi-modal system is to apply sufficient tension to ensure that the fundamental frequency is well above the vortex shedding frequency. Such tensions are usually extremely large and impossible to achieve in practice.

In shallow waters the drilling riser (i.e. conductor pipe) is not usually tensioned and, since the riser diameter is fixed, the offshore engineer has no control over the Reduced Velocity.

Marine risers generally have a Stability Parameter of about 0.5 (Chapter 4) which, under lock-on conditions, will produce a vibration amplitude in the region of 1.5 cylinder diameters. The offshore engineer can increase the Stability Parameter by increasing the structural damping and mass, however, it is difficult to do better than double the damping and the effective mass can only be increased by a few percent. This means that the Stability Parameter cannot be increased sufficiently to significantly reduce the level of vortex induced vibration.

The inability to control vortex induced vibration in a drilling riser by manipulating any of the four fundamental dimensionless groups leaves the offshore engineer with a choice of three alternatives :-

- Operate despite the presence of vortex induced vibration. Problems may be reduced by improving the design of the structure and by using better quality materials.
- Halt operations during unfavourable sea states. Down time may be acceptable if vortex induced vibration only rarely becomes a problem, for example, during exceptional sea conditions.
- Control the vortex induced vibration by fitting some kind of suppression device.

It is shown in Chapter 4 that, where vortex induced vibration is a serious threat to operations, the first and second of these alternatives are usually rejected on economic grounds, and the only satisfactory option, therefore, is to fit a suppression device.

2 CLASSIFICATION OF SUPPRESSION DEVICES.

Suppression devices fall into one of two broad categories; mechanical or fluid-dynamic. The latter rely on the surrounding fluid for their operation, whereas the former do not, and would work equally well in vacuo.

2.1 Mechanical Devices.

By definition, mechanical devices have no direct effect on the fluctuating fluid force which causes vortex induced vibration. They work by altering the structures response to the fluid force, either by shifting the structural natural frequency away from the vortex shedding frequency (i.e. De-tuning the structures natural frequency), or by increasing the amount of damping in the structure. It follows that mechanical devices influence vortex induced vibration via their effect on the Reduced Velocity, Cauchy number and/or Stability Parameter.

De-tuning usually requires a substantial increase in the structural natural frequency. For example, guy wires and/or internal bracings are used on many land based transmission masts to increase the structural stiffness by several orders of magnitude. In general, bracing wires are unacceptable on marine drilling risers, particularly in deep water,

where a large number of wires would be needed along the length of the riser. Such wires would make many of the routine operations, such as running a casing or drillstring, impractical.

Both active and passive mechanical devices can be used to increase damping. Active dampers are rare but have been used to control the vortex induced vibration of stacks on land. These complex devices detect structural motion and attempt to produce an equal but opposite motion by accelerating a mass; this is usually achieved electromechanically (Walshe & Wootton 1970). This type of device is unsuitable for the hostile environment found offshore where simplicity, reliability and ruggedness are essential requirements for any hardware.

Passive mechanical dampers are more common and include:-

- Impact dampers: The most widely used example is the chain type where a rubber covered chain hangs against the structure and dissipates energy during vibration by impacting with the structure (Walshe & Wootton 1970).
- Coulomb Friction Dampers: These were successfully used on the Forth Road Bridge during construction. Masses were attached to the bridge by wire ropes and were free to slide up and down inclined ramps. Vibration of the bridge was effectively suppressed by the friction between the masses and the ramps.
- Hysteretic Friction Dampers: Material with high internal damping can be attached to parts of the structure which experience large amounts of strain during vibration (Walshe & Wootton 1970).

Passive dampers usually increase the inherent structural damping by a factor of two or three. This increase may be sufficient to significantly reduce vibration amplitudes in a low density fluid (e.g. air) where Stability Parameters are already high, but it is rarely sufficient to be effective in water.

In general, therefore, mechanical suppression devices are not suitable for use on marine drilling risers, and the only solution is to attack vortex induced vibration at its source by reducing the fluctuating fluid force. By definition, this necessitates the use of a fluid dynamic device.

2.2 Fluid Dynamic Devices.

Fluid dynamic devices suppress vortex induced vibration by modifying the flow pattern in the vicinity of the structure in such a way as to reduce or disrupt the fluctuating component of the fluid force. They can be used to alter the frequency of the fluctuating force but usually operate by reducing its local and/or global strength (i.e. by reducing local vorticity and/or the correlation of vorticity along the length of the cylinder).

Suppression is, therefore, achieved by fundamentally altering the fluid dynamics in the vicinity of the cylinder, but without necessarily altering any of the four important non-dimensional groups (Re , Ca , V_r , K_s).

There are a number of secondary effects associated with fitting fluid dynamic devices. The structures added mass is usually affected and this alters the natural frequency and the Stability Parameter. Hydrodynamic damping can also be affected by the presence of a fluid dynamic device.

Many different types of fluid dynamic device have been developed and a review of the research into their performance, which spans nearly forty years, is given below.

3 REVIEW OF FLUID DYNAMIC SUPPRESSION DEVICES.

The most interesting and informative means of classifying fluid dynamic suppression devices would probably be to use the fundamental mechanism of their operation as the criterion, however, almost all the devices operate by using more than one basic mechanism which makes this method of classification impractical.

Instead, the devices are classified below according to the position at which they are located relative to the cylinder, and an account of the fundamental mechanisms involved in their operation is dealt with separately. This is similar to the approach adopted by Every (1982) and Zdravkovich (1981) in their reviews of suppression devices. Three broad classes of device emerge using this criterion:-

A **topographic device** is defined as any device which modifies the form of the cylinders surface without being situated predominantly in the wake of the cylinder. Suppression is achieved by interfering with boundary layer separation in order to disrupt the initiation and formation of vortices. Topographic devices are usually homogeneous and their performance is, therefore, not affected by the direction of the incident current (i.e. they are omnidirectional devices).

A **shroud** is defined as any device which is situated at a distance from the cylinder so as to allow the passage of fluid between the shroud and the cylinder. Devices which are situated predominantly in the wake of the cylinder are excluded from this category. Shrouds usually affect all three stages of the vortex shedding process (vortex initiation, formation and shedding) and are usually omnidirectional devices.

A **wake device** is defined as any device which lies, or is designed to lie, predominantly in the wake of the cylinder. They are, therefore, unidirectional devices although in some cases omnidirectionality is achieved by allowing the device to rotate about the cylinder. The primary means of suppression is by interference with vortex shedding and the latter part of the formation, although in some cases (for example a well designed fairing) vortices are prevented from occurring altogether.

In order to compare the performance of different devices it is useful to introduce the parameter "Amplitude Suppression Ratio" (ASR):-

$$ASR = \frac{Y_b - Y_s}{Y_b} \quad (1)$$

Y_b = Maximum amplitude of vibration of the bare cylinder, in the cross-flow direction, over a defined range of Reduced Velocity.

Y_s = Maximum amplitude of vibration of the cylinder with the suppression device fitted, in the cross-flow direction, over the defined range of Reduced Velocity.

In most cases, the defined range of Reduced Velocity is about 4.5 to 10.

Where possible each entry in the following review contains information on the following key areas:-

- Suppression Ratio and Reduced Velocity.
- Stability Parameter.
- Reynolds Number.
- Drag Coefficient.
- Salient features of experimental apparatus.

If any of these areas are not covered below it is because the relevant information was not given in the original paper.

3.1 Topographic Devices.

3.1.1 Strakes.

Strakes are believed to date back to the first World War when rope was helically wound around submarine periscopes to stop them vibrating. Since then they have become very widespread on land based stacks but, ironically, are less commonly used on underwater structures. They have the advantage of being an omnidirectional device but suffer from having a high drag coefficient.

The sharp edge of the strake forces flow separation to occur at the strake rather than from the surface of the cylinder. The helical application of the strake ensures that, even though vortices are still shed in the wake of the cylinder, their axial correlation is disrupted resulting in a reduction in the global fluctuating lift force and, hence, in the severity of the vortex induced vibration.

Much of the experimental work to determine the optimum design of strakes was done in the late 1950's following the 1956 Clean Air Act which demanded taller chimneys. Consequently, almost all the experiments were conducted in wind tunnels.

In 1957 Scruton & Walshe developed an effective sharp edged rectangular strake for controlling wind induced vibration and recommended that a three start helix be used with a pitch of $15D$. Increasing the strake height from $0.03D$ to $0.12D$ progressively decreased the likelihood of instability and increased the Reduced Velocity at which peak amplitudes occurred. At the lowest value of damping tested ($K_s=1.3$) the $0.12D$ high strake achieved a suppression ratio of 0.93.

In 1959 Woodgate & Maybrey conducted a series of wind tunnel tests which demonstrated that the optimum design of a rectangular strake was three start, $5D$ pitch and $0.09D$ strake height (Fig. 3T.1). This design is embodied in current ESDU recommendations (1981).

The criterion for optimisation used by Woodgate and Maybrey was the minimisation of the aerodynamic excitation force which was measured indirectly, via its effect on the model's logarithmic decrement of decaying vibration under the action of applied eddy current damping. Results are quoted at two levels of vibration amplitude, $0.01D$ and $0.1D$.

Cowdrey & Lawes (1959) measured the drag coefficient of two three start, $5D$ pitch helical strakes having strake heights of $0.06D$ and $0.12D$. The measured coefficients were 1.35 and 1.45 respectively and were constant throughout all three Reynolds number regimes ($1 \times 10^5 < Re < 4 \times 10^6$).

Wootton (1978) also states that the drag coefficient is independent of Reynolds number and has a value of 1.3 (Three start helix, $5D$ pitch, $0.1D$ strake height).

Vickery & Watkins (1964) carried out a series of wind and water tunnel tests ($10^4 < Re < 10^5$) and concluded that strakes were equally effective in air and water. For Stability Parameters of 3 or more there was no benefit to be gained from increasing the amount of strake coverage above 50% (Fig. 3T.2). Note that this conclusion was based on lumped antinodal device coverage. Their experiment did not investigate the effect of strake coverage on cylinders having a Stability Parameter less than one. From these results (Fig 3T.2) it is evident that the performance of the strakes depended on the Stability Parameter of the cylinder; generally, the ASR decreased with increasing Stability Parameter.

Ruscheweyh (1972) established that for land base stacks susceptible to fundamental mode vibration it was best to cover the top third of the structure with strakes. This optimisation was based on the criterion of minimum bending moment at the base of the structure.

More recently (1979) Every & King carried out a series of small scale experiments in water on a cantilevered cylinder having a Stability Parameter of 0.44 ($10^4 < Re < 10^5$). Helical pitch and strake coverage were investigated over a limited range and the results are shown in Figure 3T.3A. Increasing the cylinders Stability Parameter from 0.44 to 1.7 decreased the effectiveness of the strake (Fig. 3T.3B), however, increasing it still further ($K_s=5.37$) increased the strakes effectiveness returning it to its original value for the $K_s=0.44$ case.

The sharp edged rectangular strake has become the most widely used design, but there are instances where it is more convenient to fit other designs. Several researchers have investigated the performance of strakes having a circular cross-section because of the practical convenience of this simple design. In 1956 Price carried out experiments on a six start wire helix but found the device ineffective over a range of pitch and wire diameter.

Considerable work in this field has been conducted by Nakagawa et al (1965), but the most comprehensive investigation was that of Weaver who varied the number of windings, cross-sectional diameter, pitch and coverage (Figs. 3T.4 & 3T.5). The tests were carried out in a wind tunnel where Reynolds numbers were in the range 10^4 to 1.4×10^5 . The optimum configuration, based on the criterion of minimising the fluctuating lift force on a stationary cylinder, was found to be a 4 start helix, 0.1D cross-sectional diameter and 12D pitch. An ASR of 0.9 was achieved with only 20% coverage, 0.98 with 40% coverage and there was, of course, little improvement with a greater amount of coverage.

These impressive ASRs were achieved for relatively stiff cantilevers. Weaver does not recommend the use of this particular suppression device where the Slenderness Ratio exceeds 20 because "test members having small diameters and large non-dimensional amplitudes demonstrated an ability to persist in fairly large amplitudes of oscillation with spoilers attached, whereas vibrations in other types of members were readily suppressed with spoilers".

An herring-bone strake arrangement has been investigated by Sallet & Berezow (1972). The device was tested on a model cylindrical buoy whose only support was that of a single mooring cable (Fig. 3T.6.). Both the density of the wires and the herring-bone angle were varied; the results are shown in Fig. 3T.7.

In 1979 King & Every carried out model tests in water on a 3 start, round section helical strake having a cross-sectional diameter of 0.05D and a pitch of 4.5D. They found that the ASR was 0.57 with 50% device coverage. The Stability Parameter of the cylinder for this test was 0.44. Further configurations (e.g. strake diameter 0.1D) were not investigated because of practical constraints.

Halkyard (1987) carried out large scale tests at sea to investigate the performance of a round section strake formed by wrapping a rope around a bare cylinder. The strake had two starts, an 18D pitch and a height of 0.1D. Tests were performed in the upper sub-critical Reynolds number range. The bare cylinder strummed in the cross-flow direction and the peak amplitude of 1.0D occurred at a Reduced Velocity of 6.5. The strake achieved a ASR of 0.80 and shifted the peak response to a higher reduced velocity ($V_r=9.0$). The strake also eliminated all in-line oscillation which reached a peak of 0.25D for the bare cylinder (Fig. 3T.8).

Exxon used round sectioned helical strakes to guard against possible vibrations in a 20 inch drill casing during running and stabbing operations (Gardner T.N & Cole M.W., 1982). No details of the strake dimensions are given but the device proved effective in laboratory trials and on site.

3.1.2 Fins.

A fin is defined here as any linear protuberance which lies parallel to the cylinder's longitudinal axis. In 1955 Baird investigated the use of sawtooth fins as a solution to a wind induced pipeline oscillation problem (Fig. 3T.9). Wind tunnel tests were carried out on configuration 1 which showed that oscillations were "completely eliminated". Because of the cost of the fin and the possibility of harmonic modes occurring, the fins were actually installed on site in configuration 2. Site measurements subsequently showed the suppression device to be fully effective.

The first person to formally investigate and report the use of straight fins as a suppression device was Price in 1956. Using a small water flume (0.23 m wide x 0.25 m deep) he fixed the water velocity at 0.2 m/s ($Re=4.6 \times 10^3$) and measured vibration amplitudes over a range of model support stiffness.

Arrangements of four and eight radial fins were investigated for different angles of incident flow (Fig. 3T.10). The results are summarised in the table below:-

No. of fins	Incident flow angle	ASR (1)
8	0.0	0.40
8	22.5	-0.07
4	0.0	(2)
4	45.0	(3)

Notes:-

- (1) ASR varied with support stiffness; the quoted values are for the worst case.
- (2) This configuration was found to be "ineffective" but no quantitative information was presented.
- (3) This configuration was found to be "effective" but no quantitative information was presented.

As part of the same experimental programme Price tested a single configuration of three wire fins (Fig. 3T.10) and found the ASR to be 0.43. Because Price was searching for an effective omnidirectional device he did not test fins comprehensively and quickly dismissed them in favour of shrouds.

In 1978 Mahrenholtz & Bardowicks conducted wind tunnel tests at sub-critical Reynolds number to investigate the effects of five equispaced radial fins (Fig. 3T.11). Amplitude response, drag and lift were all found to be dependent on the flow angle, theta. The device was ineffective as a suppressor; it actually worked as a vibration amplifier for

most values of flow angle.

In 1978 Gartshore used a cantilever model in a wind tunnel (0.69 m deep x 0.9 m wide, $1.5 \times 10^4 < Re < 1.2 \times 10^5$) to investigate ordinary and saw-toothed fins (Fig. 3T.12). Tests were carried out using low and high levels of turbulence in the incident flow (0.1 % and 14 % respectively). Gartshore's results are summarised in the table below:-

Ordinary fins			Saw-tooth fins		
Turb. level	ASR		Turb. level	ASR	
	$\theta=0.0$	$\theta=90.0$		$\theta=0.0$	$\theta=90.0$
0.1	-3.0	-1.5 (1)	0.1	(2)	(2)
14.0	0.28	-3.8 (1)	14.0	0.38	(3)

Notes:-

- (1) No maximum amplitude was reached for these tests.
- (2) No data given.
- (3) This value was large and negative although the precise value was not given.

The addition of fins altered the shape of the amplitude response curve but, in all cases, significant responses were confined to the Reduced Velocity range 3.5 to 7.0.

Novak (1966) used staggered radial fins arranged in a helix to suppress the flow induced vibration of a Television mast (Fig. 3T.13). Wind tunnel tests on a 1:10 scale model showed the fins to be effective at sub-critical Reynolds number, however, at super-critical Reynolds number vibration amplitudes approached the bare cylinder values. The radial fins were used on the full scale mast and performed satisfactorily.

Several researchers have investigated the effect of placing "tripping wires" along the length of a circular cylinder. These wires can be used to trigger turbulence in the boundary layer at Reynolds numbers significantly below the value which is normally associated with the critical transition. The effect of the tripping wires is therefore to shift the C_d - Re curve towards lower Reynolds number.

Fage & Warsap (1929) showed that tripping wires having diameters ranging from 3 % to 57 % of the boundary layer thickness progressively reduced the drag coefficient of a cylinder by prematurely triggering the transition to the critical regime.

Tripping wires can influence vortex induced vibration as well as drag. Price (1956) has shown that for a spring mounted cylinder in water at sub-critical Reynolds number tripping wires reduced the peak amplitude of vibration by 44 %. Price used three wires which were placed at angles of 0 and ± 60 degrees relative to the upstream stagnation point. All three wires had a diameter of 0.023 times the cylinder diameter.

Naumann (1966) used wind tunnel tests to demonstrate that relatively small diameter wires can be used to control vortex shedding from a stationary cylinder. The wires, known as separation wires, were used to fix the point of flow separation and subsequent vortex formation along the length of the cylinder. Arrangements of broken wires were found to disrupt separation and eliminate regular vortex shedding. Naumann used a particular arrangement of three separation wires to determine the critical angular difference necessary to eliminate regular vortex shedding (Fig. 3T.14).

3.1.3 Discontinuous protuberances.

A variety of protuberances have been arranged in a non-linear position on the surface of a cylinder in an attempt to suppress vortex shedding and/or reduce the drag coefficient.

Mizuno (1970) carried out a wind tunnel investigation into the effect on flow separation of attaching a series of spheres to the surface of a cylinder (Fig. 3T.15). He found that the spheres could have a profound effect on the separation line running along the length of the cylinder. Sinusoidal and irregular wavy separation lines could be produced depending on sphere spacing (L/D) and Reynolds number. A critical sphere spacing was identified ($L/D=1.4$) above which no interaction occurred between the wakes of adjacent spheres.

Sallet & Berezow (1972) investigated the vibration response of a submerged moored buoy using model tests in a towing tank. This freely supported system vibrated with zero to peak amplitudes of up to three diameters before suppression devices were attached. Several configurations of cylindrical studs were fitted to the surface of the cylinder in an attempt to suppress vibration (Fig. 3T.16). Two other configurations were also tried using rectangular and triangular studs but the details of these arrangements are not given in the original paper. All configurations produced a degree of suppression and the best overall ASR achieved was 0.53 (for configuration no.2, fig. 3T.16).

3.1.4 Voluminous Devices.

This category includes devices which cover an appreciable part of the cylinders circumference and displace a significant amount of fluid relative to that displaced by the cylinder.

In 1956 Price used a small water flume (0.23 m wide x 0.25 m deep, $Re=4.6 \times 10^3$) to investigate the possibility of using cylindrical devices (referred to as "bushings") to suppress vortex induced vibration. He only tested one configuration which consisted of $1.25D$ bushings spaced out at regular intervals along the cylinder (Fig. 3T.17) and found that it resulted in a slight increase in vibration amplitude. The worst ASR recorded was -0.06.

3.2 Shrouds.

Many different types of shroud have been designed, as described below, but they all operate on similar principals. The leading edge of the shroud makes the flow which is incident upon the cylinder highly turbulent. This is believed to disrupt flow separation from the cylinder and, therefore, also disrupts the vortex initiation process.

Many shrouds tend to direct high pressure fluid from the front of the cylinder round to the base of the cylinder via the gap between the shroud and the cylinder. This increases the base pressure and reduces the strength of vorticity in the cylinder's wake.

Large scale vortex shedding is also reduced by the trailing edge of the shroud because each element of the shroud generates its own small scale vortex street.

As described in section 3.1 above, Price (1956) investigated several different suppression devices including helical strakes, wire fins and bushings. During these investigations he soon abandoned all these devices and turned his full attention to the perforated shroud because of its relatively good suppression performance and omnidirectionality.

Price tested several different designs of perforated shroud with round holes (Fig. 3S.1). The best results were obtained with a hole size of $0.125D$ and with the perforations arranged as a square lattice (configuration no.2). He obtained performance envelopes which showed the cylinder's Reduced Amplitude, both with and without the shroud, over a range of structural support stiffness (Fig. 3S.2). These envelopes can be represented by a single curve of ASR versus structural support stiffness (Fig. 3S.3).

All tests were carried out in a water flume having a constant velocity of 0.20 m/s ($Re=4.6 \times 10^3$) and in all the tests bar one the full length of the cylinder was shrouded. Price found that a coverage of 50% over the antinodal half of the cantilevered cylinder was ineffective at reducing vibration amplitudes (Fig. 3S.2).

In order to establish the performance of the shroud at critical and super-critical Reynolds number Price carried out a series of tests in a wind tunnel. The shroud was effective at reducing the response in both regimes, but the bare cylinder itself only vibrated with a maximum amplitude of 0.005 diameters.

Price used further wind tunnel tests to measure the drag on the shrouded cylinder throughout the three Reynolds number regimes and found that the drag coefficient was invariant with Reynolds number (Fig. 3S.4). Note that the drag coefficient is based on the cylinder, not the shroud, diameter.

In 1968 Walshe studied a variety of square holed perforated shrouds (Fig. 3S.5). The results of his wind tunnel tests ($Re=10^5$) are presented as a graph showing the variation of the inverse of the cross-flow Reduced Amplitudes D/Y with Stability Parameter (Fig. 3S.6). Walshe concluded that the optimum shroud geometry was configuration no. 1 which had an Open Area Ratio (OAR) of 20%, however, for Stability Parameters above 25 the best suppression was clearly attained by configuration no. 4 (i.e. 36% OAR,

gap=0.059D).

The results show that 25% antinodal shroud coverage was sufficient to give effective suppression in air but not in water. For Stability Parameters above 10 there was no appreciable difference between the 25% and 12.5% shroud coverage results.

It is important to note that even the lowest Stability Parameters tested were about ten times the values which are typical for offshore structures and are of limited use in this respect, however, Walshe's results do illustrate the dependence of shroud performance on Stability Parameter. They also demonstrate that shroud performance is not strongly dependent on gap size, G , in the range 0.05D to 0.12D. This lack of strong dependence is particularly marked at Stability Parameters of less than ten.

Knell (1969) measured the drag coefficient of square holed shrouds having Open Area Ratios of 20% and 36% and a Gap of 0.117D. The experiment was carried out in a wind tunnel over the Reynolds number range 8×10^4 to 3×10^6 . Knell found that the drag coefficient for both the shrouds was 0.9 and that it was invariant with Reynolds number. This value is close to that measured by Price for round holed perforated shrouds.

Wootton & Yates (1970) used the same model and test facility as Knell to establish the proportion of the total drag which was taken by the shroud itself. They concluded that the drag coefficient of the shroud (based on shroud diameter) increases as the gap between it and the cylinder increases and as its Open Area Ratio decreases. Generally, the shroud took a high proportion of the total drag; for the case of the 36% Open Area Ratio shroud having a gap of 0.12D 70% of the total drag force acted on the shroud.

The first full scale use of shrouds was at Immingham in 1972 where they were used on three 36.5 m long piles with diameters of 0.46 m, 0.6 m and 0.76 m. Wootton (1972) carried out an extensive experimental programme on the piles and used the shrouds to control vibration and hence prevent structural failure during the long periods of down time between tests. He used square holed shrouds with an Open Area Ratio of 30% and a gap of 0.1D. The shroud covered 21% of the wetted length of the pile and was positioned at its vibration antinode. The shroud was found to completely eliminate pile vibration below a Reduced Velocity of 3.8 (i.e. in-line vibrations), but its performance at higher Reduced Velocities was not investigated.

Walshe & Cowdrey (1972) carried out model tests in a wind tunnel on a pair of stacks and found that a shroud (square holed, 36% OAR, Gap=0.12D) covering 25% of the stack was ineffective at controlling turbulent buffeting.

In 1976 King & Every conducted model tests to establish the effectiveness of a shroud at suppressing the vortex induced vibration of a tripod structure. Strakes were precluded from these tests because their high drag coefficient would have caused unacceptably high bending stresses in the full scale structures.

There was a further problem regarding the fitting of strakes; it was impractical to fit the strakes to the submerged piles after they had been driven into the sea-bed and it was also difficult to drive piles which had already been fitted with strakes.

The shroud used by King & Every was of the round hole type having an Open Area Ratio of 36% and a Gap of 0.1D. The maximum prototype design velocity of 3.35 m/s corresponded to a Reduced Velocity of 3.1 which meant that cross-flow vibration would not occur but that in-line vibration may be a problem. The Stability Parameter for the piles was 0.3. King and Every found that when the shroud covered 20% of the wetted length of each tripod leg in-line vibration was effectively suppressed. An ASR of 0.94 was achieved when the shroud was placed at the centre of the pile and this only dropped to 0.90 when the shroud was moved to just below the water surface. The shroud was found to be less effective when the flow direction was such that the shroud was in the wake of a leading pile.

Every and King tackled another problem offshore which was caused by vortex induced vibration in 1979; the drill casing of the jack up drilling rig "Offshore Mercury" was suffering unacceptable vibration in tidal currents of up to 2.0 m/s ($Re=1.5 \times 10^6$ i.e. super-critical). Model tests were carried out in a water flume at sub-critical Reynolds number in order to establish the effectiveness of a variety of suppression devices including square holed perforated shrouds with an Open Air Ratio of 35% and gaps of 0.1D and 0.15D.

Tests were carried out on cantilever cylinders having Stability Parameters of 0.44 and 1.7 with the top 50% shrouded (i.e. the antinodal end was shrouded). The gap between the shroud and cylinder was varied to assess its effect on shroud performance. The results are shown in Figure 3S.7 and in summary form in the table below:-

Stability Parameter	Gap (D)	ASR
0.44	0.10	0.68
0.44	0.15	0.69
1.70	0.10	0.46

An axial rod shroud was developed by Zdravkovich in 1971. He investigated its performance as a suppression device in a small water flume at sub-critical Reynolds number and also measured the pressure distribution around a static model in a wind tunnel.

Shroud Open Area Ratio was varied from 24% to 96% , the rod size was varied from 1.08D to 1.5D and the circumferential distribution of Open Area Ratio was also altered.

For the 1.08D rod shroud the best results were achieved with high open areas. The ASR was 0.85 for the 96% OAR shroud, although peak response was probably not reached because of the restricted velocity range (Fig. 3S.8). Conversely, for the 1.5D rod shroud best results were achieved with low Open Area Ratios. The highest ASR obtained was 0.78 with an Open Area Ratio of 25%. Overall, the most effective configuration was OAR=63%, $d=1.25D$ with which an ASR of 0.93 was achieved.

Increasing Open Area in the vicinity of the stagnation point was found to improve the shrouds performance, but if this was done at any other location the shroud performance was impaired.

Zdravkovich measured the drag coefficient of two shroud configurations (OAR=63%, $d=1.25D$ and OAR=81%, $d=1.16D$) and found it to be 0.9 in both cases.

Wong (1977) investigated an axial slat suppression device which was later used to prevent possible vibration of the Edinburgh Royal Infirmary stack (Wong & Cox 1979). The experiments were carried out in a wind tunnel where the Reynolds number reached a maximum of 1.5×10^5 and were designed to optimise the geometry of the shroud. Having tested twenty different configurations Wong concluded that the best design consisted of 25 slats with an Open Area Ratio of 40% (Fig. 3S.9), however, Wong stated that the optimum configuration should be determined for each particular structure under consideration.

Wong continued his investigation using the 25 slat configuration by studying the effect of changing the amount of device coverage on the cylinders vibration response over a range of Stability Parameters (Fig. 3S.10). Unfortunately, this range did not cover Stability Parameters below 2.0.

As part of their search for a solution to the Offshore Mercury drill pipe vibration problem (discussed above), Every & King (1979) tried two different slat designs (Fig. 3S.11) and tested them on cylinders having a range of Stability Parameters. The results are shown in Figure 3S.12 and a summary of the ASRs achieved is given in the table below. Note that both the slat designs were unidirectional.

Stability Parameter	Slat type	Coverage (%)	ASR
0.44	small	50.0	0.81
0.44	large	50.0	0.85
0.44	large	100.0	0.86
1.70	small	50.0	0.74
1.70	large	100.0	0.92
5.37	large	100.0	0.98

The small in-line vibration which occurred with the plain cylinder was completely eliminated by all of the slat configurations.

3.3 Wake Devices.

3.3.1 Fairings.

Although, theoretically, fairings are an ideal solution to the vortex shedding problem because they are capable of totally eliminating the near wake vortex street, in practice they suffer from the disadvantages of unidirectionality, fitting problems and high expense. Very little model testing has been reported but several full scale offshore instances of their use have been.

In 1969 Dale carried out experiments on a strumming cable at low Reynolds number ($Re < 1.5 \times 10^3$) and found that complete coverage with a rotating fairing (chord to thickness ratio, $c/t=7$) eliminated strumming and reduced the drag coefficient to 0.45 at $Re = 1.5 \times 10^3$.

In 1975 Petrobras began planning in order to drill exploratory wells in a high current area off the coast of Brazil. The drilling contract required operation in 150 m of water with currents of up to 4 m/s which presented extreme problems due to possible vortex induced vibration of the 0.61 m diameter riser and its choke and kill lines. The high steady drag on the riser was also of concern.

Grant and Patterson (1977) and Albers & Da Silva (1977) describe the method of solution to this problem. The dual requirement of reduced vortex induced vibration and steady drag led them quickly toward the use of a fairing as the only suppression device capable of meeting their specification.

For fitting reasons the chord to thickness (c/t) ratio had to be less than three. Because of the complete lack of experimental data for fairings with such a short c/t ratio, a wind tunnel test programme was commissioned.

Initial tests were carried out on a fairing with a c/t ratio of 2.5. For zero angle of attack the drag coefficient, C_d , was 0.12 and did not vary strongly with Reynolds number, however, the fairing was not stable in this position as shown by the positive slope of the restoring moment curve in Figure 3W.1A. If free to do so, the fairing would take on an angle of attack of about 6 degrees resulting in an increase in drag coefficient to around 0.3.

The test programme continued with a fairing which had a c/t ratio of 2.0. "Fish tail" fins were added to the trailing edge of the fairing at 45 degrees to the centreline to make it weathervane and thereby minimise drag. This method proved successful and the length of fin was optimised resulting in a drag coefficient of 0.17 and a favourably sloping (i.e. stable) restoring moment curve (Fig. 3W.1B).

This design was successfully used offshore. The full scale fairing was fitted with plain bronze bearings to allow a full 360 degree rotation. Grant and Patterson used their data along with that from other sources in order to produce a graph of drag coefficient versus c/t ratio for a general design of fairing (Fig. 3W.2).

Henderson (1978) carried out a comprehensive series of wind tunnel and water flume tests on a faired towing cable in order to investigate severe and unpredictable kiting problems in the full scale cable. Friction between the cable and fairing was identified as one cause of the problem as it prevented weathervaning. Friction was increased by the weight of the fairings themselves, particularly on the lower fairings in a stacked system with no intermediate thrust bearings. In addition, once kiting had occurred, large hydrodynamic forces also contributed to the frictional force.

The proprietary cable fairing which was tested was found to be of a very poor design. Even though the fairing had a c/t ratio of four it had a very poor restoring moment (i.e. low tendency to weathervane) because of poor streamlining at the front and rear of the fairing. In order to improve the restoring moment a flat plate was attached to the trailing edge of the fairing which increased the c/t ratio to 5.5, thus moving the centre of hydrodynamic pressure further aft. Water tank trials were used to assess bearing performance. This was found to be unsatisfactory so PTFE washers were used to reduce friction on the axial thrust bearings. In ocean trials these modifications were found to produce improved and repeatable behaviour but did not eliminate kiting.

Exxon faced extreme ocean condition when they attempted to drill off the coast of Brazil in 1980 (Gardner & Cole, 1982). Surface currents were measured in excess of 2 m/s which caused excessive drag on the structure. The drill casing was strengthened and its mass was increased to reduce the steady deformation of the drill pipe caused by the drag force but nothing was done to suppress vortex induced vibration. Before the drill pipe had even reached the sea bed (water depth 430 m) it failed catastrophically due to fatigue caused by vortex induced vibration.

Although strakes were used to suppress casing vibration (section 3.1), fairings were used on the riser and on all other wires and cables exposed to the current. The riser fairing was developed in a water flume at MIT. Tail fins were fitted, which were very similar to those used by Grant & Patterson (1977), in order to improve the weathervaning performance of the fairing. After the dimensions of the fairing had been optimised the drag coefficient was measured and found to be 0.3.

Testing was continued on a larger scale at the Danish Ship Research Laboratory where a 5.5m deep towing tank was used to assess the performance of the fairings. They were found to be highly successful and so full size fairings were put through ocean towing trials as a final check prior to being used successfully during drilling operations which began in January 1982. .

Packwood & Williams (1986) reviewed several different fairing designs suitable for use in the marine environment. These included Williams own round nosed fairing ($c/t=2.5$), Smiths tear drop fairing ($c/t=2.17$) and Calkin's tear drop fairing with trailing edges ($c/t=2.5$) (Fig. 3W.3).

These low c/t ratio fairings can be highly effective at reducing drag but there performance is very Reynolds number dependent. For example, the Williams fairing operated well at $Re=1.5 \times 10^6$ but its performance suffered at Reynolds numbers below 10^6 due to increased flow separation. Another disadvantage of this type of fairing is that

the centre of pressure can be as far forward as $0.05x_c$ aft of the leading edge.

The commercially available "fathom" fairing which has been studied by Henderson (discussed above) was also reviewed and shown to be unsatisfactory because of its high drag coefficient and forward centre of pressure.

Packwood and Williams sought to develop an improved fairing design having low drag and a centre of pressure as far aft as possible. They used R.A.E.'s computer modelling and wind tunnel facilities to design an efficient fairing for low Reynolds number applications. The final design had a c/t ratio of 5.0 and a truncated trailing edge.

The fairing had a drag coefficient of 0.03 at a zero angle of attack (half that of the commercial fathom fairing) and the centre of pressure was $0.25x_c$ aft of the leading edge.

3.3.2 Splitter plates.

Roshko (1954) investigated two types of splitter plate in a wind tunnel in order to test the then current theories on wake formation behind bluff bodies.

A long splitter plate ($c=5D$) which was attached to the base of the cylinder was found to eliminate periodic vortex shedding and increase base pressures ($Re=1.45 \times 10^4$).

A short splitter plate ($c=1.14D$) was also tested with a varying gap between it and the base of the cylinder (Fig. 3W.4). As the gap between the cylinder and plate was increased vortex shedding became increasingly suppressed until, at a g/d ratio of 3.85, there was a discontinuous reversion and vortex shedding re-established itself in the near wake of the cylinder as if the plate was not present.

Dale (1969) investigated the use of splitter plates as cable strumming suppressors. The plates were found to effectively eliminate strumming provided they were placed at the strum antinodes. The best geometry was found to be $c=10D$, $L=20D$.

Sallet (1970) developed an extensive theory intended to predict the optimum length of a splitter plate. He also carried out tests on a cylindrical moored buoy in a water towing tank (described under Section 3.1) which established that at sub-critical Reynolds number a splitter plate of length $3D$ was sufficient to completely eliminate vortex induced vibration, whilst a plate of length $2D$ provided very little suppression. At super-critical Reynolds number, however, a $1.5D$ splitter plate was sufficient to completely suppress vortex induced vibration.

Zdravkovich (1981) summarised the work carried out by Appelt (1973) to determine the effect of various splitter plates on the flow around a stationary cylinder as follows:-

Short plates ($c/D < 2$) were found to significantly reduce drag (31% reduction for $c/D=1$) and shift the formation of vortices to the trailing edge of the plate. The Strouhal number varied by $\pm 10\%$ over the range $0 < c/D < 2$.

In the range $2 < c/D < 5$ vortex shedding and drag coefficient were progressively modified. For $c/D=5$ the drag coefficient was 0.8 and vortex shedding was eliminated. Increasing the length above 5D had no significant effect because the flow became re-attached to the plate at approximately 5D downstream from the base of the cylinder.

Halkyard (1987) carried out large scale tests at sea to investigate the performance of splitter plates as suppression devices for several different amounts of coverage. The plates were 3.33D long and were free to rotate around the cylinder. Halkyard found that 100%, 77% and 50% coverage completely eliminated the vortex induced strumming vibration but that 25% coverage only produced a 20% ASR (Fig. 3T.8).

3.3.3 Guide vanes.

Grimminger (1) (1945) investigated a variety of guide vanes using a water filled towing tank ($Re=1.7 \times 10^4$) and a model cylinder with a Slenderness Ratio of 24. (Fig. 3W.5).

The flat plates were successful in reducing vortex induced vibration to a "low level" and reduced the drag coefficient from 1.06 to 0.83. The contoured guide vanes (Fig. 3W.5B) performed better and reduced the drag coefficient to 0.51.

Gerrard (1) (1966) found that a perpendicular plate placed in the wake of a stationary cylinder had a significant effect on the frequency of vortex shedding. (Fig. 3W.6)

- (1) These references were not available to the author. Information was taken from the review by Zdravkovich (1981).

3.3.4 Flexible wake devices.

Due to fluid drag most flexible suppression devices naturally trail behind the cylinder to which they are attached and are, therefore, classified as Wake devices.

(a) FLAGS.

During the installation of Shells Cognac platform some of the piles were vulnerable to vortex induced vibration in both pendulum and cantilever modes (King, 1978, Fisher, 1979). King carried out model tests in a water flume at sub-critical Reynolds Number to investigate the possibility of using flags as suppression devices (Fig. 3W.7A). The longest piles had a Slenderness Ratio of 46 and were found to vibrate in the pendulum mode with a Reduced Amplitude of 1.2. Two flags (10D high x 3D long) attached at a distance of 5D from each end of the pendulum were found to completely suppress vortex induced vibration in the first two modes of vibration for both uniform and sheared incident flow.

In the cantilever mode flags were again found to be effective vibration suppressors and all the configurations which were tested had ASRs of at least 0.86 (Fig. 3W.8).

There was considerable variation in the measured drag coefficient for the different flag configurations. The lowest coefficient ($C_d=0.72$) was achieved by attaching the 10D x 5D flag to the cylinder at three points only and leaving a D/8 gap between the base of the cylinder and the leading edge of the flag.

A great variety of flexible devices have been used in an attempt to suppress the vibration of submerged cables. This vibration (known as strumming) can cause severe noise interference in communication cables such as those used with sonar "fish". It also greatly increases the drag on the cable and increases the likelihood of failure through fatigue. High drag is undesirable on the umbilical cables used with Remote Operated Vehicles (ROV) because it can severely limit the capabilities of the ROV (i.e. Max. speed, depth and operating area).

Many of the suppression devices have been developed for the Navy where the elimination of strumming is essential for stealthy operation although the value of the drag coefficient is relatively unimportant. Consequently, some of the flexible suppression devices have very high drag coefficients and information on drag is often rather limited.

The subject of strum suppression has been extensively reviewed by Hafen & Meggitt (1977) and Every (1982). Flexible strum suppression devices may be categorised into Ribbons, Hairs and Fringes.

(b) RIBBONS.

Ribbons have been found to be effective strum suppressors with ASRs in the range 0.5 to 0.95 although they can have rather high drag coefficients ($1 < C_d < 5$) (Every 1982). The optimum dimensions are 4 to 6D long, 1 to 2D wide with a gap of 1 to 2D (Fig. 3W.7).

(c) HAIRS.

Hairs of length 6D and spaced at D/2 intervals are very effective strum suppressors with ASRs in the range 0.85 to 1.0 and, from the limited data which is available, appear to have a relatively low drag coefficient ($1 < C_d < 2$), (Hafen & Meggitt, 1977).

(d) FRINGES.

Hairs arranged in tufts are known as fringes (Fig. 3W.7). Although only limited data is available they appear to be equally as effective as hairs at suppressing strumming ($0.85 < ASR < 1.0$). The optimum configuration is a length of 4 to 6D and spacing of about 1D, (Every 1982). Fringes have higher drag coefficients than hairs and there is considerably more scatter in the results ($1.5 < C_d < 6.0$), (Hafen & Meggitt, 1977).

3.3.5 Fluid flow devices.

It is possible to improve the flow pattern around a bluff body by either injecting or removing fluid at strategic locations. The technique of sucking away the boundary layer has been reported by Prandtl & Tietjens (1957). The phenomenon of vortex shedding results directly from the vortex initiation which occurs in the boundary layer and, therefore, if this layer is removed vorticity is eliminated at its source. By removing fluid from the walls of a suddenly enlarged channel Prandtl showed that a flow pattern resembling that of a potential flow could be obtained.

Cornish (Year unknown) observed that a fixed vortex is formed close to the upstream stagnation point of a circular cylinder with an unbounded end. He reasoned that the vortex remained stable because fluid flowed out of the vortex to form trailing vortices on either side of the cylinder and this flow balanced the flow of fluid from the free stream into the vortex.

Cornish postulated that vortex shedding from the rear of a bluff body could also be stopped by extracting sufficient fluid from the wake of the body to balance the incoming flow from the boundary and entrainment layers. Experiments were carried out which demonstrated that his theory was correct. Increasing the amount of flow which was removed progressively reduced the Strouhal number until shedding ceased altogether (i.e. $St=0$) when flow balancing was achieved. Cornish applied the technique to several bluff bodies including a simple cylinder, an aerofoil and a model of a sports car.

From the data given in Cornish's paper it appears that the amount of flow removal required is of the order of 5% of the free stream flow which would pass through the projected area of the bluff body in the absence of the body (referred to from here on as the "absent body free stream flow").

Aerofoils with a blunt trailing edge can be used to fix the flow separation point and thus eliminate certain types of flutter. There is a low speed drag penalty, however, and in an attempt to reduce this Wood (1964, 1966) investigated the effects of base bleed on the flow around a blunt based aerofoil and its drag coefficient.

Wood found that increasing the bleed rate progressively increased the distance between the aerofoil and the point at which the wake burst into the vortex street. Bleed flow also decreased the strength of the vorticity in the street but did not significantly alter the characteristic dimensions of the street or the Strouhal number. An optimum bleed flow rate was determined (12.5% of absent body free stream flow) which reduced the net profile drag by 70%.

Igarashi (1978) studied the effects of cutting a slit through the centre of a cylinder on the flow patterns and drag coefficient, presumably as a means of achieving a base bleed without the need for an external flow source. Tests were carried out over a wide range of slit angle. For angles between 0 and 40 degrees a positive bleed flow occurred, the base pressure and Strouhal number were increased and the drag coefficient decreased. For angles between 60 and 90 degrees the bleed flow was negative and fluctuated as the vortices were shed. The base pressure and Strouhal number decreased and the drag

coefficient increased.

King & Miller invented and patented a novel method of suppressing vortex induced vibration which involved an air manifold. A circular manifold was placed around the circumference of the cylinder and was fed with compressed air. Naturally, the air ascended in the wake of the cylinder and induced a vertical flow which disrupted the vortex shedding correlation.

The air manifold was tested at small scale on a cylinder measuring 1.0m long x 0.032m in diameter (Re were of the order of 2×10^4). The device achieved a maximum ASR of 0.76 when the flow rate of free air was set to 62% of the absent body free stream flow (Every 1982).

Large scale tests were also carried out using a 10m long cylinder with a diameter of 0.73m (Re were of the order of 3×10^5). The best ASR achieved was 65% using a free air flow rate of 20% of the absent body free stream flow (Every 1982).

4 DISCUSSION.

Fluid dynamic suppression devices are used extensively on land based stacks and in the marine environment where they often represent the only practical and economic means of controlling vortex induced vibration. The importance of fluid dynamic suppression device is reflected in the number of references on the subject and the large amount of work which has been directed at their development over the past forty years.

Over 50 references are cited in this review, but there are believed to be many more which have not been published. For example, the M.O.D. have commissioned a considerable amount of confidential work in order to try to overcome the problem of noise generated by cable strumming and some of the work carried out by the oil companies on the suppression of vortex induced vibration in marine risers is commercially sensitive and has, therefore, remained confidential.

Approximately 50% of the work reported above is concerned with topographic devices; strakes alone account for 30%. The remaining work is shared evenly between shrouds (mainly the perforated type) and wake devices (mainly fairings and splitter plates).

4.1 Topographic Devices.

The helical strake is popular because it is simple, reliable and omnidirectional. The optimum design of the rectangular strake has been established for a short cylinder ($L/D=14$) vibrating with a low amplitude (up to $0.1D$) as :-

- Three start, $5D$ pitch, $0.1D$ height.

Pitch is the least important of the three design variables; good performance has been obtained with pitches ranging from 2.25 to 15 diameters. On a cylinder with a low

Stability Parameter (0.44) a strake with optimum dimensions can be expected to achieve a ASR of 0.74 (Every & King, 1979) (fig. 3T3).

Strake performance depends on the structures Stability Parameter (Vickery & Watkins (1964) (fig. 3T2). The vast majority of research has been carried out using cylinders having a high Stability Parameter (i.e. $K_s > 2$) which were subjected to sub-critical flows.

Performance also depends on other parameters as can be seen by comparing the results of Vickery & Watkins (1964) with those of Every & King (1979) (fig. 3T3). At a Stability Parameter of 1.7 King and Every's tests in water gave a ASR of 0.51, whereas Vickery and Watkins test in air gave a value of 0.94. The reason for this discrepancy is not clear but the principal differences between the experimental set-up in the two cases were mode shape, aspect ratio, mass ratio and damping.

For cantilevered cylinders the optimum ASR is obtained with the top half of the cylinder covered; there is no benefit from increasing the coverage.

The optimum design of round sectioned strake is:-

- Four start, 12D pitch, 0.1D height.

The criterion for optimisation was minimum fluctuating lift force on a stationary cylinder and this may, therefore, not be the optimum design for a cylinder which is undergoing vortex induced vibration. This type of device has been used successfully offshore but meaningful performance data is very limited.

Fins have been investigated by five researchers and, although there have been successful results, these were limited to cases where the vibration problem was caused by a unidirectional wind. In water, fins generally do no better than suppress vibrations by 40% and are highly unidirectional.

4.2 Shrouds.

Perforated shrouds offer omnidirectionality and a drag coefficient which is invariant with Reynolds number and has a value of around 0.9, however, their suppression performance is generally not as good as a strakes; ASRs of 0.50 are typical.

The performance of shrouds depends on Stability Parameter and the optimum design has not been rigorously established for any particular Stability Parameter range, however, the following design has become accepted as the optimum over a range of Stability Parameter :-

- Round or square holes arranged in a square lattice.
- Hole size = 0.1D.
- Gap = 0.1D.
- OAR = 30%.

Very little research has been done at low Stability Parameter (i.e. $K_s < 2$) and it is restricted to the sub-critical Reynolds number range. Price (1956) achieved an ASR of 0.50 under the following conditions :-

- Stability Parameter = 1.0 (inferred from bare cylinder response).
- Hole size = 0.125D.
- Gap = 0.125D.
- OAR = 36%.

Every & King (1979) tested a similar shroud but with the hole size and gap of 0.1D. With 50% antinodal coverage the shroud achieved a ASR of 0.68 and 0.46 on cylinders having Stability Parameter of 0.44 and 1.7 respectively.

From the limited amount of research which has taken place it appears that Axial rod and slat shrouds have similar drag coefficients to perforated shrouds and may offer significant improvements in suppression performance.

4.3 Wake Devices.

The fact that fairings can be very effective suppression devices is taken for granted and, consequently, there is very little data available on ASR. Very few researchers have investigated the effects of partial coverage and little is known about its effects on suppression or drag. Work has concentrated on the development of stubby fairings (i.e. $c/t < 3$) and, in particular, on overcoming their inherent lack of directional stability.

Fairings offer unparalleled performance in terms of both vortex induced vibration and drag reduction but there is a price; they are relatively complex to make and fit and are unidirectional. They can be made to weathervane about the cylinder's axis but this introduces further complexity and cost.

Splitter plates need to be at least 3D long to provide a high level of suppression and coverage may be reduced to around 50% before performance is impaired. These devices are much simpler than fairings but the penalty for this is higher drag ($C_d = 0.8$ is typical), however, they suffer the same major disadvantage as fairings; they are unidirectional.

Guide vanes represent a compromise between fairings and splitter plates in terms of both complexity and performance. There is no quantitative information available on their suppression performance but it is known that they reduce the drag coefficient to about 0.5

A wide variety of flexible wake devices have been developed, principally for use as cable strum suppressors. In general, they can provide effective levels of vortex induced vibration suppression but tend to increase the bare cylinder drag coefficient. The information on their performance when fitted to cylindrical structures is very limited.

Boundary layer suction and base bleed are a simple and highly effective means of suppressing vortex induced vibration and drag, however, in practice, their use is restricted to complex structures such as cars and aerofoils where it is feasible to overcome the problems of providing the means of fluid displacement and where there is sufficient scope for modifying the structures design to incorporate the fluid displacement device. In general, they are not a feasible solution to the problem of vortex induced vibration in marine risers.

Every's air manifold device could be used offshore but its use would probably be restricted to shallow waters and to short term solutions.

5 CONCLUSION.

Fluid dynamic suppression devices are often the only practical means of controlling vortex induced vibration, particularly in marine risers, and this has stimulated a large amount of research into their performance.

In terms of the three important criteria, ASR , drag coefficient and economic cost, the suppression devices which are currently available for marine use are a compromise. No single suppression device performs well in all three areas.

Much of the research into the performance of suppression devices has been uncoordinated and non-standard. The result is that it is rarely possible to directly compare the findings of different researchers without making assumptions about important parameters such as the cylinder's Stability Parameter.

A large part of the work on topographic devices, particularly strakes, is applicable to high Stability Parameter cylindrical structures such as land based stacks and is not applicable to sub-sea structures such as marine risers.

Due to a lack of basic research it is not possible, in general, to predict the performance of suppression devices under the following conditions :-

- Partial device coverage.
- Sheared incident flow.
- Critical flow regime (particularly important for partial device coverage).
- Multi-modal cylinder vibration.

CHAPTER 4

ECONOMIC ASPECTS OF VORTEX INDUCED VIBRATION AND ITS SUPPRESSION

1 INTRODUCTION.

Vortex induced vibration is a phenomenon which generates a great deal of academic interest. This is evident from the large number of references on the subject, some of which were cited in Chapters 2 and 3 of this thesis. The phenomenon involves complex fluid-structure interaction, the understanding of which presents many intellectual challenges. Indeed, even after a century of research, certain aspects of the subject remain a mystery.

But the subject is also of great practical importance and much of the impetus for research has come from commercial pressures to overcome engineering problems created by vortex induced vibration.

Vortex induced vibration has recently become of particular importance in many fields of engineering because of the trend toward more efficient and cost effective design with smaller factors of safety. Structures have become lighter and more flexible, whilst flow velocities have, in many cases, increased. This trend is nowhere more apparent than in the offshore oil industry.

The exploitation of offshore oil reserves involves the use of slender cylindrical structures, such as drilling and production risers, which can be exposed to strong ocean currents. These structures (generically known as marine risers) are compliant and their natural frequencies are often similar to the natural frequency of vortex shedding caused by the local currents. They are, therefore, susceptible to vortex induced vibration.

Marine risers are lightly damped and the mass ratio ($m_e/\rho.D^2$) is quite low, at least when compared with that of land based structures subjected to wind induced loads. Consequently, the Stability Parameter is small and, when vortex induced vibrations do occur, they can reach large amplitudes.

Recent trends in the oil industry have been toward the exploitation of marginal fields in deeper and more hostile waters. Drilling now takes place in water depths of up to 1000 m and currents in excess of 2 m/s have been encountered. In this type of environment vortex induced vibration is a serious problem and has to be controlled if the oil reserves are to be successfully recovered.

2 OBJECTIVE.

The purpose of this chapter is twofold:-

- To demonstrate the practical importance of vortex induced vibration and the need for its suppression.
- To provide a means of appraising the various solutions which are available to combat vortex induced vibration.

This second objective is important because, without some quantitative means of appraising the different solution options, it is impossible to decide which method is best suited to a particular situation and difficult to assess the merit of new ideas or methods which may be put forward in an attempt to improve on existing solutions.

3 METHOD.

Ever since the search for oil moved offshore it has been the oil industry which has suffered most heavily from vortex induced vibration. Because of this, it was thought that the best way of meeting the objectives set out above would be to concentrate the analysis on the offshore oil industry.

In the oil industry, just as in any other commercial venture, the criterion by which the merits of different events or options are judged is a financial one. It was, therefore, decided that the best way of assessing the practical importance of vortex induced vibration was by analyzing the economic effects it has on offshore oil production, and that the same financial criterion should be used to appraise the different solution options which are available to combat vortex induced vibration.

The economic analysis was divided into two distinct areas:-

- The basic economics associated with the exploitation of a typical offshore oil field.
- The economic effects of vortex induced vibration on the offshore oil industry.

4 BASIC ECONOMICS OF OFFSHORE OIL FIELD EXPLOITATION.

Before presenting the economic analysis, it was felt that a brief introduction to the technical operations which are involved in offshore oil field development would be helpful, and this is given in sections 4.1 and 4.2 below.

4.1 The Principal Phases of Offshore Oil Field Exploitation.

Offshore oil field exploitation generally involves four distinct phases :-

Phase 1: Initial search.

The initial search for oil involves a variety of information and data gathering activities, the most important of which is a seismic survey which is usually carried out from a specially equipped ship.

Geologists and geophysicists are heavily involved during this phase, particularly with the data processing, analysis and interpretation. In essence, they are searching for sedimentary rocks of a particular age with an anticlinal structure which is necessary for the formation of an oil or gas reservoir.

Phase 2: Appraisal drilling.

Like Phase 1, this phase also involves data gathering, processing and analysis, but the area of the search is more focused as a result of the deductions made during Phase 1.

Typically, between ten and twenty appraisal wells are drilled (this is known as wildcat drilling) in order to assess the oil field's potential. This is usually carried out using a floating vessel such as a drill-ship or a semi-submersible drilling rig.

Again, geologists and geophysicists are required to study the core samples which are recovered from the wells. They then interpret the information in order to predict the size and shape of the oil field and the quality and realisable quantity of the oil contained in the field.

Phase 3: Development.

Field development involves a large number of complex and expensive operations:-

- Designing, manufacturing and constructing the oil platform.
- Development drilling. This may involve up to forty individual production wells which are drilled from the main platform.
- Designing, manufacturing and constructing the topside equipment.
- Installing all the necessary pipelines, risers and onshore terminals.

Phase 4: Production.

This phase includes all the operations which are necessary to operate the platform and produce the crude oil. The principle cost items faced by the operator during this phase are:-

- Labour, supplies and power.
- Maintenance and inspection.
- Field licence fees.

4.2 Offshore Drilling Techniques.

Drilling plays an important part in the discovery and development of an offshore oil field and, as explained later in this Chapter, vortex induced vibration affects drilling operations more than any other phase of offshore field development. For these reasons a more detailed description of the drilling operation is called for.

4.2.1 Drilling platforms.

Drilling can take place from a variety of surface platforms depending, primarily, on the depth of water and on the purpose of the well (i.e. appraisal or development) :-

(a) JACK-UP PLATFORM.

Jack-ups are used in water depths of between 15 and 120 m. After being towed into position their legs are hydraulically extended so as to jack the platform several metres above the sea surface to give a stable drilling base.

(b) DRILL-SHIPS.

Drill-ships are highly manoeuvrable and can operate in water depths of up to 800 m using anchoring lines to hold station whilst drilling. More sophisticated dynamically positioned ships can operate in still deeper waters.

(c) SEMI-SUBMERSIBLES.

Like drill-ships, semi-submersibles use either anchoring lines or dynamic positioning systems to hold station. They can operate in water depths of up to 1000 m and offer a more stable drilling platform than a drill-ship.

(d) FIXED PLATFORMS.

Although most appraisal drilling is carried out using one of the three methods listed above, 85 % of development drilling takes place from fixed platforms in water depths of up to 400 m. Deviated drilling allows a wide area to be accessed from a single platform.

4.2.2 Drilling operations.

Drilling the well is accomplished by rotating a drill-string from the surface which has a drilling bit attached to its lower end. During drilling, fluid (mud) is pumped from the surface down through the drill-string to lubricate the bit, remove the cuttings, support the sides of the well and control any back pressure from the down hole fluids.

The first hole to be drilled usually has a diameter of 36" and extends just a few metres below the sea-bed. A liner, called the conductor pipe, is inserted into the hole before drilling continues.

A 30" diameter hole is drilled next by a string which passes through the conductor pipe and reaches a depth of up to 100 m. The drill- string is removed from the hole and a 30" liner (liners are called casing strings) is inserted and cemented in position.

Drilling and lining continues in this telescopic fashion until, at the bottom of the well, which can be as much as 6000 m (4 miles) below the sea-bed, the hole diameter has reduced to 7".

4.2.3 Marine risers.

The type of engineering hardware used which bridges the gap between the surface platform and the sea-bed depends on the water depth. There are two basic systems in common use.

(a) SHALLOW WATER - CONDUCTOR PIPE.

In water depths below 120 m where rigid platforms are used a 30" conductor pipe is lowered from the surface and stabbed into the sea-bed. The pipe extends from the sea-bed to the surface platform and remains in position throughout the entire drilling operation.

All the drilling and lining operations are carried out through the conductor pipe and the pipe is also used to return the drilling mud to the surface platform.

Control valves are used in case of emergency to prevent high pressure fluids being blown back up the well. These valves are positioned on the platform at the top of the conductor pipe and are known, collectively, as the 'christmas tree'.

Conductor pipes usually have Stability Parameters in the range 0.4 to 0.8 (Every & King, 1979, Lamb, 1990) so, when vortex induced vibrations occur the amplitude of vibration tends to be large (chapter 2).

(b) DEEP WATER - MARINE DRILLING RISER.

Deeper waters dictate a different approach to marine operations, although the operations which are carried out beneath the sea-bed are very similar to the shallow water case.

The first operation is the lowering and locating of a wellhead template which sits on the sea-bed. A primary guidance system is then installed which usually consists of four wires which run from the wellhead to the surface platform.

The large diameter holes (36", 30" and 20") are drilled using a drill-string which has no outer casing and is located solely by the wire line guides. In the absence of any outer casing there is no alternative but to allow the drilling mud and well cuttings to be dispersed into the sea.

The 30" and 20" well casing strings are also run to the sea-bed using the wire guides with no conductor pipe to protect them.

The smaller diameter holes which penetrate deep into the earth's crust cannot be drilled in this way because of the risk of a well blow out and a marine drilling riser is, therefore, employed at this stage.

The riser, which has an outer diameter of 21", allows the mud to be recycled and contains any well products which would otherwise pollute the sea. It also acts as the guide for running the drill-string and casings, thus replacing the wire line system. Because of the risk of the riser failing the control valve assembly for preventing well blow back is positioned at the wellhead where it is known as a blow off preventer (BOP) stack.

Two hydraulic lines (choke and kill lines) are run along the sides of the marine drilling riser to allow the BOP stack valves to be controlled from the surface.

Deep water risers are kept under axial tension by hydraulic units located on the surface platform and, where a floating platform is used, the tensioning unit also incorporates a heave compensation mechanism.

Cylindrical buoyancy modules are often attached to the riser, covering most of its length, to reduce the effective weight of the riser in situ. This reduces the amount of top

tension required and also offers some protection to the riser and its choke and kill lines (Buoyancy modules are discussed in more detail in chapter 8).

With buoyancy modules attached, the Stability Parameter of a deep water marine riser is about 0.3, but the value is increased by a factor of about two if modules are not used because of the increase in the riser's effective density.

4.3 Economic Analysis.

For the purpose of this economic analysis, the end product is taken to be crude oil and so none of the refining or so called "downstream" operations are considered. The "upstream" operations necessary for the production of crude oil can be split into four main phases as discussed in section 4.1 above. The costs of these four phases are discussed in the following section.

4.3.1 Costs.

The principle costs associated with the four phases of field development are as follows:-

(a) INITIAL SEARCH.

The surveying itself costs between £400 and £800 per kilometre (Offshore Engineer, Feb. 1989) and the search for a reservoir requires many hundreds of kilometres to be covered in a grid like mesh. The average total cost of searching for and finding a North Sea oil reservoir is £60 M (Offshore Engineer, Jan. 1989).

(b) APPRAISAL DRILLING.

Each appraisal well costs over £2 M to drill and the average total cost of this phase for a North Sea field is £93 M (Offshore Engineer, Jan. 1989).

(c) DEVELOPMENT.

The average cost of field development in the North Sea is £1,340 M (Offshore Engineer, Jan. 1989).

(d) PRODUCTION.

Unlike the first three phases, the cost of production is a variable cost and is, therefore, measured per unit of oil produced i.e. per barrel. The average cost of production in the North Sea is £0.51 per barrel (Offshore Engineer, Jan. 1989) and, since a typical North Sea field contains 350 M recoverable barrels of oil, the total cost of production over the life of the field is £179 M (Offshore Engineer, Jan. 1989).

The above costs can be summarised as follows:-

Operation	Average cost for a North Sea field	
	(£M)	(% of total)
Initial search	60.0	3.6
Appraisal drilling	93.0	5.6
Development	1340.0	80.1
Production	179.0	10.7
Total	1672.0	100.0

The table above shows the considerable investment that is made by the oil companies during the early stages of a field's life; the three discovery and development phases together make up 90% of the total cost of producing the crude oil.

4.3.2 Revenues.

The revenues from oil field development are directly related to the rate of oil production and the price of oil. During the development phase of a field's life, oil production rises rapidly and peak production is usually reached two or three years into the development programme. Thereafter, the natural tendency is for the reservoir production rate to decline with time due to a reduction in reservoir pressure, however, efforts are made to keep the actual rate of production stable by using three techniques:-

- Water injection to boost reservoir pressure.
- Gas lift.
- Increasing the number of production wells by continued development drilling.

In the North Sea an average field contains 350 M barrels of recoverable oil and the average rate of production over the life of the field is 91,000 barrels per day (Offshore Engineer, Jan. 1989). It follows from these figures that the average life of a North Sea oil field is 10.5 years.

Since the oil price crash of 1985 prices have become remarkably stable. During the period 1986 to 1988 the price of a barrel of crude oil fluctuated between \$15 and \$18. The average price over the period May 1988 to April 1989 was \$17.6 per barrel (Financial Times, April 1989) and this price is predicted to vary little over the next year or so.

At the time of this analysis, the dollar/sterling exchange rate was 1.71 (Financial Times, April 1989) and, therefore, the average revenue from a North Sea field was :-

$$\text{Daily revenue} = 91,000 \times (17.6 / 1.71) = \text{£}937 \text{ K.}$$

$$\text{Total field revenue} = 350 \times 10^6 \times 17.6 / 1.71 = \text{£}3602 \text{ M.}$$

4.3.3 Analysis of costs and revenues.

At this stage it is possible to assess the actual worth of exploiting a typical North Sea oil field using the Net Present Value (NPV) method. The results of the NPV analysis are also used in section 5.3 below, where they are required to compare the costs and benefits of a variety of solutions to the problems caused by vortex induced vibration.

$$NPV = -C + \sum_{i=1}^I \frac{NR_i}{(1 + k)^i} \quad (1)$$

where:-

- C = Initial capital outlay of the project.
- i = Particular year of project life.
- I = Project duration in years.
- NR_i = Net revenue in year i.
- k = Cost of capital.

Assuming the cost of capital is 10%, and using the average value of costs and revenues given above, the NPV of a typical North Sea oil field can be established:-

$$C = 60 + 93 + 1340 = \text{£}1493 \text{ M.}$$

The net revenue per barrel of oil produced is the price minus the direct cost of producing it i.e. $\text{£}10.29 - \text{£}0.51 = \text{£}9.78$.

Assuming a flat rate of production of 91,000 barrels per day which equals 33.2 M barrels per year:-

$$NR_i \text{ (for all } i) = \text{£}9.78 \times 33.2 = \text{£}324.8 \text{ M.}$$

The present value of these revenues assuming a 10 year project life is given in the table below:

Year	1	2	3	4	5
PV (£M)	295.3	268.4	244.0	221.8	201.7
Year	6	7	8	9	10
PV (£M)	183.3	166.7	151.1	137.7	125.2

In 1989 the Net Present Value of exploiting a typical North Sea oil field was therefore:-

$$\text{NPV} = -£1493 \text{ M} + £1996 \text{ M} = £503 \text{ M}.$$

5 THE ECONOMIC EFFECTS OF VORTEX INDUCED VIBRATION.

5.1 A Classification of Potential Problems.

Vortex induced vibration can cause a wide variety of problems in engineering structures, but these can be classified into one of only four types.

(a) STRESS.

The structural oscillation caused by the vortex shedding produces internal cyclical stresses which can lead to failure by the mechanism of fatigue. This is the most common form of failure but catastrophic failure due to buckling can also occur as a result of severe steady bending stresses in the structure which are caused by the increased hydrodynamic drag on the structure which accompanies vortex induced vibration (see b below).

(b) DRAG.

The drag coefficient of a stationary structure can be increased significantly by vortex induced vibration. For example, that of a circular cylinder is amplified by a factor of up to 2.7 by large amplitude vortex induced vibration (Griffin, June 1985).

An increase in drag is usually undesirable. As well as increasing the internal loads, as discussed in 'a' above, it increases the loads imparted to adjacent components and may, therefore, affect their design and sizing.

(c) DISPLACEMENT.

Vortex induced vibration can cause both steady and dynamic structural displacement and this may adversely affect the performance of the structure.

Even if the component which is exposed to the vortex induced vibration is not itself adversely affected, problems can still arise due to the transmission of the vibration to adjacent components.

(d) NOISE.

The wake of a body undergoing vortex induced vibration is highly disturbed and can be detected acoustically. This can be a serious problem for certain military operations which depend on stealth for their success.

5.2 Effects of Vortex Induced Vibration on the Offshore Oil Industry.

A recent questionnaire and literature survey (Lamb 1989) has shown that vortex induced vibration has caused problems for the oil industry in three of the four potential problem areas: stress, drag and displacement.

Three of the four main phases involved in field exploitation were affected: appraisal drilling, development and production. A summary of the problems suffered by the oil industry as a result of vortex induced vibration is given below:-

(a) STRESS.

Vibration induced fatigue failure is a potential problem in both drilling and production risers, although actual catastrophic failure has only occurred in drilling risers.

Vortex induced vibration can affect conductor pipes and marine drilling risers and can, therefore, delay drilling operations in all water depths.

In the worst reported case, fatigue failure caused by vortex induced vibration resulted in the loss of an entire drill-string and the ultimate abandonment of the well, although drilling in the same area did re-commence a year later following extensive re-design of the drilling equipment.

The choke and kill control lines which are attached to deep water drilling risers can also suffer from vortex induced vibration. This can cause damage to the lines themselves and to the parent riser through transmitted loads and, in some cases, through impact.

During the construction of oil platforms, piles are sometimes used which are temporarily positioned as cantilevers either extending up from the sea bed or hanging down from a surface structure. The vortex induced vibration of such piles has been reported, but failure was avoided by design modifications.

The high drag forces associated with vortex induced vibration cause potential problems of high steady bending stress in drilling and production risers, although there have been no reported failures of this kind. Riser buckling due to these forces was identified as the dominant failure mode during attempts to drill in hostile waters off the coast of Brazil, but actual failure was avoided by fitting fairings.

(b) DRAG.

The amount of axial tension applied to a marine riser to reduce bending is dependent on the hydrodynamic drag force on the riser. An increased drag coefficient caused by vortex induced vibration may, therefore, require larger and more expensive tensioning equipment to be installed.

Where a dynamically positioned vessel is used its station holding ability may be threatened by the increased drag on the riser caused by vortex induced vibration. Increased power will be required to hold station and, in some cases, a larger and more expensive type of surface vessel or structure may be necessary.

(c) DISPLACEMENT.

Stabbing operations such as the locating of a drill-string into a sea bed template require accurate positioning and, therefore, displacements caused by vortex induced vibration can jeopardise the operation and may damage equipment.

5.3 The Economic Cost of Vortex Induced Vibration.

There are a large number of different problems which can, and have, occurred as a result of vortex induced vibration during the exploitation of offshore oil fields, as discussed in section 5.2. For a complete analysis, each type of problem would have to be studied independently and it is likely that the most cost effective type of solution would vary from case to case.

It would be tedious to carry out such a comprehensive study and, more importantly, unnecessary because the vast majority of significant vortex induced vibration problems occur during the operation of drilling, as outlined in section 5.2.

In particular, it is the drilling of appraisal and initial development wells, using drill-ships and semi-submersibles, which are most often adversely affected by vortex induced

vibration. This area alone has, therefore, been selected for economic analysis.

5.3.1 The cost of drilling a well.

Before going on to discuss the costs incurred due to vortex induced vibration, it is necessary to assess the costs involved in drilling a typical 'problem free' well.

A detailed study of the operations and costs involved has been completed based on information supplied by Shell and Astec Developments Ltd (Lamb, 1989) and a summary of the costs which were identified in this study is presented below :-

Operation	Unit Cost (£K)	Total Cost (£K)
Rig hire	36.0 per day	828.0
Labour	2.2 per day	50.6
Materials	1568.0 per well	1568.0
Total cost of drilling a well		2446.6

Note: In the table above total cost figures are based on a typical well completion period of 23 days.

5.3.2 The cost of downtime.

Most of the solutions to the problems caused by vortex induced vibration involve a certain amount of rig down time and it is, therefore, necessary to calculate the cost of lost production (i.e. the opportunity cost), which will occur as a result. Of course, production is seldom lost, but delayed, and the cost incurred is therefore directly related to the cost of the capital.

A typical oil field has a productive life of 10 years and so the cost of one day of downtime at the beginning of the project is equal to the difference between one days net revenue in year 1 and the present value of one days net revenue in year 11. Using the assumptions and figures detailed in section 4.3:-

$$\begin{aligned}
 \text{Cost of one days} &= \frac{(295.3 \times 10^3)}{365} - \frac{(113.8 \times 10^3)}{365} \\
 \text{downtime (£K)} & \\
 &= \text{£497 K.}
 \end{aligned}$$

5.3.3 Economic comparison of the solution options.

The economic cost of any problem ultimately depends on the way in which the problem is dealt with. The question 'what does vortex induced vibration cost the oil industry' can, therefore, only be answered by analyzing the different methods which are used by offshore engineers to deal with the problem. These methods are referred to here as 'solution options', even though some of them make no attempt to solve the problem in a technical sense.

In Chapter 3 (Section 3.1) it was shown that, faced with the threat of vortex induced vibration, the offshore engineer had to choose between three basic solution options. The cost of each of these solutions is assessed below so that an economic comparison can be made.

(Note that the first option presented in chapter 3 has been expanded into two options below to allow a more thorough analysis).

Solution 1 : Operate despite the presence of the vibration.

With this option there is a high risk of riser or wellhead failure which would force the operator to either:-

- (a) Replace the failed component and continue until the well was complete.
- (b) Abandon the well. This would occur if it became impossible to "stab" back into the incomplete well.

If "a" occurred the amount of downtime might be anything from about 30 days to several months, depending on the type and number of failures that occurred. The opportunity cost of the 'lost' production would therefore be in excess of £15 M.

In addition to this enormous cost there is the risk of forced well abandonment (b above) which would, at the very least, result in the loss of investment in the well (up to £2.5 M) plus the opportunity cost of 'lost' production (up to £11 M). At worst, this course of action could result in complete failure to exploit the oil field which, for a typical North Sea field, would cost about £500 M.

Solution 2 : Operate despite the presence of the vibration with an improved design.

Both the cost and the success of structural design improvements to solve vortex induced vibration problems will be highly site and operation specific. In many instances, such as Exxon's attempts to drill in the hostile waters off Brazil, it would not work (Lamb 1989).

There is one reported case, however, where this option was successfully used. After Britoil had suffered catastrophic failure of a wellhead during drilling off the Shetland Isles in 1981 they re-designed the wellhead to strengthen it and to reduce stress concentrations. The design changes solved the vortex induced vibration problem and the well was completed successfully (Lamb, 1989).

Britoil were not willing to supply information on the cost of the new wellhead, but the estimated cost, based on 1989 values is shown in the following table :-

Item	Description	Cost (£K)
Labour	Design. (360 man-days @ £210/day.)	75.6
	Manufacture. (360 man-days @ £140/day.)	50.4
Materials	Assorted.	10.0
Total estimated cost of new wellhead.		136.0

Solution 3 : Halt operation during unfavourable seas.

The number of days downtime incurred as a result of choosing this option will be highly site and rig specific but consider, as a real example, Britoil's drilling operation off the N.W Coast of the Shetlands in 1981 (Lamb, 1989).

Dangerous currents were present for about 70% of the time and, therefore:-

$$\text{Total time taken} = \frac{\text{Normal time taken}}{(1 - 0.70)} = \frac{23}{0.3} = 77 \text{ days.}$$

$$\text{Downtime} = 77 - 23 = 54 \text{ days.}$$

The total cost of this downtime consists of three main items, as detailed in the table below:-

Item	Cost (days)	Cost/day (£M)	Total cost (£M)
Opportunity Cost	54	0.497	26.9
Rig hire	54	0.036	1.9
Labour	54	0.002	0.1
Total cost of downtime			28.9

The total cost is dominated by the opportunity cost i.e. the "lost" production.

Solution 4 : Control the vibration by fitting a suppression device.

In the vast majority of cases where vortex induced vibration has been a significant threat to offshore drilling, suppression devices have been used (Chapter 3).

Strakes, shrouds and fairings have all been used offshore. Strakes and shrouds may be suitable for use in shallow waters but in deep waters and in all cases where currents are high (2 m/s or above) fairings are the most favoured device.

Although fairings are the most effective solution they are not generally popular because of their cost, size and the requirement for free rotation about the drilling riser in order to allow for changes in the direction of the ocean current.

Because of a general lack of knowledge surrounding the use of fairings and, in particular, their performance under full scale ocean conditions, offshore operators are forced to carry out a large amount of preparatory work before using them. This may include both fundamental design work and a comprehensive series of model and full scale tests.

In order to assess the typical costs involved, consider Exxon's use of fairings on a drilling riser off the coast of Brazil in 1982 (Lamb, 1989). The table below contains all the major phases which were involved in the fairing of the drilling riser together with an estimate of their cost. (Exxon were unwilling to release a detailed cost breakdown).

Table 5.1 Estimated cost of using fairings.

Operation	Cost (£K)
Fundamental design and development using scale models.	20.0
Full scale design and manufacture.	330.0
Full scale testing to verify performance under ocean conditions.	80.0
Modification to the drilling rig.	200.0
Instrumentation to monitor riser behaviour.	50.0
Opportunity cost of delayed production due to extra time needed to fit the fairings (3 days).	1492.0
Extra drilling rig hire (3 days).	108.0
Extra labour to fit fairings (3 days).	20.0
Total cost of fairings.	2300.0

(B.P. Exploration and Production estimated that, in 1984, the total cost of using fairings was approximately £1000 per foot of riser. Assuming an average rate of inflation of 7% over the past 5 years, this figure becomes £1.4 K per foot. Since the Exxon riser in the above example was 1400 feet long, the cost of fitting fairings to it, using B.P's estimate, is £2.0 M which gives credence to the estimate shown in table 5.1 above.)

6 DISCUSSION.

This analysis has shown that offshore oil production is highly profitable, even in times of a modest oil price. A typical North Sea field can yield profits of £500 M which represents a return on investment of 34%. This level of return is accompanied by high risk, however, particularly in hostile ocean environments where hydrodynamic loading can cause a variety of operating difficulties; vortex induced vibration is just one example.

The economic effects of vortex induced vibration on the exploitation of offshore oil reserves are dominated by the opportunity cost of lost, or more correctly, delayed production, which amounts to about £500 K per day.

Faced with a vortex induced vibration problem, the offshore engineer has a number of options which are summarised in the table below in ascending order of cost:-

Table 6.1 Summary of the solution options and their costs.

Solution option		Cost of solution		Comment
No.	Description	(£M)	(% C.E.)	
2	Operate despite viv with improved design	0.1	< 0.01	Impractical except in marginal case
4	Fit suppression device.	2.0	0.13	Cost depends on depth and current.
3	Halt operations in high current	30.0	2.0	Cost is highly site specific.
1	Operate despite viv.	15.0 to 500.0	1.0 to 34.0	Cost is highly site specific.

Note: C.E. = Total capital employed, which amounts to £1493 M.

Option 2 is clearly very attractive and should be used where vortex induced vibration presents a marginal problem. In cases where vortex induced vibration is more serious the fitting of suppression devices, although expensive, is by far the most economic solution (table 6.1).

One of the aims of this thesis was to establish the effect of partial suppression device coverage on suppression performance (chapter 6). It is useful, therefore, at this point to assess the economic advantage of being able to reduce coverage.

Consider the example sited above where Exxon used 100 % fairing coverage on a drilling riser in 1400 feet of water. Assuming the coverage could be reduced to 50 % whilst maintaining adequate suppression performance the estimated costs of the various phases in table 5.1 above are modified as follows :

Operation	Cost (£K)
Fundamental design and development using scale models.	20.0
Full scale design and manufacture.	165.0
Full scale testing to verify performance under ocean conditions.	80.0
Modification to the drilling rig.	200.0
Instrumentation to monitor riser behaviour.	50.0
Opportunity cost of delayed production due to extra time needed to fit the fairings (1.5 days).	746.0
Extra drilling rig hire (1.5 days).	54.0
Extra labour to fit fairings (1.5 days).	10.0
Total cost of fairings.	1325.0

In this example, the 50 % reduction in device coverage results in a cost saving of 43 % (£1M).

7 CONCLUSION.

A typical North Sea oil field has the potential to yield about £500 M profit which represents a Return on Capital Employed of 34%.

Vortex induced vibration can cause operational problems which significantly increase the risks involved in offshore oil field exploitation. The increased costs associated with vortex induced vibration are usually dominated by the opportunity cost of delayed oil production.

Marginal vortex induced vibration problems may sometimes be solved economically by modifying the design used in the drilling riser system. Where the potential vortex induced vibration problem is more serious by far the most economic solution is to fit suppression devices to the marine riser.

The use of suppression devices can be expensive. A typical well costs £2.5 M but this figure could double if fairings are fitted to the drilling riser.

Substantial savings could be made if it were possible to reduce the amount of suppression device coverage whilst maintaining adequate control of the vortex induced vibration.

CHAPTER 5

MATHEMATICAL MODELLING OF SUPPRESSION DEVICE PERFORMANCE

1 INTRODUCTION AND OBJECTIVE.

Hydrodynamically, the performance of a suppression device is judged using two main criteria:-

- The degree of amplitude suppression it provides, as quantified by its particular Amplitude Suppression Ratio.
- The hydrodynamic drag force upon it, for a given free stream velocity.

At present, there are no established mathematical or empirical models which predict the performance of suppression devices.

Investigations into device performance have been, without exception, experimental (Chapter 3), and it has not been possible to generalise the results to include the effects of variables such as Stability Parameter, device coverage and current shear.

As a result, the oil industry is forced to carry out model testing whenever the effects of different designs, or different operational conditions, need to be established.

The main reason for the lack of a model is the complex nature of the fluid-structure interaction which occurs during vortex induced vibration. Our understanding of the vortex induced vibration of a bare cylinder is far from complete, and is not sufficient to allow the dynamic response of a bare cylinder to be predicted, except for certain special cases which include:-

- Uniform incident flow and single mode vibration.
- Sheared incident flow with cylinder behaving as an infinite string.

Given this partial level of understanding pertaining to the behaviour of a bare cylinder, it is not surprising that the more complex case of a suppressed cylinder has not yet been successfully modelled.

There is, of course, good reason to develop such a model as it would lead to both academic and practical benefits :-

- The process of model development and the predictions that the completed model would provide would lead to an increased understanding of the physical processes involved in the suppression of vortex induced vibration. This would be of direct academic interest and could also lead to improvements in the design of practical suppression devices.
- The model would enable engineers to place less reliance on physical modelling which can be expensive in time and money.

The objective of the work described in this chapter was to develop a mathematical model which can predict the performance of various suppression devices, in terms of their Amplitude Suppression Ratio (ASR) and drag coefficient.

By implication, the model must be capable of predicting the maximum amplitude of vortex induced vibration of a bare cylinder and a cylinder which is fitted with a particular type of suppression device. The model must also be capable of predicting the drag force on a vibrating bare cylinder and a cylinder which is fitted with a particular type of suppression device.

Ideally, the model should include the effects of :-

- Stability Parameter.
- Sheared incident current.
- Partial suppression device coverage.
- Multi-modal vibration.

Although there are no models currently available that are directly concerned with the performance of suppression devices, several models of the vortex induced vibration of a bare cylinder have been developed. A review of these models was carried out as a prerequisite to the development of a suppression device performance model, and a summary of the review is given below.

2 SUMMARY OF EXISTING MODELS.

The two main hydrodynamic performance criteria for suppression devices, amplitude response and drag coefficient, are also of prime importance in the study of a bare cylinder's vortex induced vibration. As a consequence, progress has been made with regard to mathematical modelling in both these areas, although the majority of the work has been concerned with amplitude response.

2.1 Amplitude Response.

The principal elements of a mathematical model include, at the very least, a model of the structure's dynamics and a model of the fluid excitation process. Of the two, the structural model is by far the simplest and it is the manner in which the excitation force is modelled, particularly with regard to the effects of fluid structure interaction, which provides the most distinguishing feature with which to differentiate between the various models available.

Using the fluid excitation model as the basis for classification produces four types of mathematical model, each having a different level of abstraction from the real physical phenomenon.

2.1.1 Flow field models.

Vortex induced vibration is characterised by the non-linear interaction which occurs between the fluid mechanics and the structural motion. A rigorous mathematical model must, therefore, offer a solution to the fundamental equations of both fluid and structural mechanics, and these equations must be solved simultaneously because they are coupled.

The equation of structural motion depends on the composition of the structure and on its mounting arrangement, but for the simple case of a linearly damped, pin-pin mounted cylinder under axial tension the equation is (Vandiver, 1985) :-

$$M \cdot \frac{d^2 y}{dt^2} + R \cdot \frac{dy}{dt} + K \cdot y = F(t) \quad (1)$$

Where:

- M = Effective structural mass.
- R = Effective linear damping coefficient.
- K = Effective structural stiffness.
- F(t) = External force applied to the structure.
- y = Displacement at the antinode.

The fluid motion is described by the Navier-Stokes equations (three equations in all; one for each cartesian co-ordinate, x,y,z) together with the continuity equation (Prandtl & Tietjens, 1957).

For laminar flow, there are only four unknowns which are required for a complete solution, but despite there being four distinct equations, as described above, no unique solution exists because of non-linearities in the partial differential terms (Humphries, 1987).

The situation is considerably more complicated when the flow is turbulent and, in order to proceed it is therefore necessary to simplify the full Navier-Stokes equations in some way. Fromm & Harlow (1968) have developed a simplified model which is capable of describing the flow around a two dimensional stationary bluff body.

A second approach, pioneered by Chaplin (1973) and Graham (1978), is to artificially introduce discrete vortices into the wake of the cylinder, thereby avoiding rigorous modelling of the boundary layer and flow separation. The overall flow field around the cylinder is described by a superposition of the cylinder's potential flow with that associated with the discrete vortices.

Early models completely ignored the boundary layer and had fixed separation points; only large scale vorticity was modelled using single point discrete vortices.

Improvements in the method have been made possible by the increase in the processing power of computers, and it is now possible to model the boundary layer, variable separation points and free shear layers using large arrays of small discrete vortices. Good results have been obtained for the case of a stationary cylinder in two dimensional sub-critical flow. The method has been recently developed to include structural motion but is not yet capable of incorporating three dimensional effects.

For a more comprehensive review of the method and its development see Humphries (1987).

2.1.2 Wake oscillator models.

This technique introduces a major abstraction into the modelling, since no attempt is made to model the flow field around the cylinder. The equations of fluid motion are discarded and the fluid is effectively replaced by a mathematical forcing function (the wake oscillator) which attempts to reproduce its effects without any concern for its physical manifestation.

Equation 1 is normally used to describe the structural motion. The input force (F) is determined by the wake oscillator which is designed to reproduce the self limiting nature of the vortex induced vibration and has a frequency which follows the Strouhal relationship.

The concept of the wake oscillator was first proposed by Bishop & Hassan (1964), but has since been developed by numerous other researchers, notably Hartlen & Currie (1970) and Iwan & Blevins (1974).

Wake oscillator models have been successful in describing many of the features which are known, through experiment, to characterise vortex induced vibration. For example, the self limiting resonant response which occurs when the natural frequency of vortex shedding is close to the structural natural frequency, and lock-on itself, which is widely

considered to be the most interesting and perplexing characteristic of vortex induced vibration.

Wake oscillator models have been specifically designed to reproduce the non-linear interaction which occurs between the structure and the ambient fluid during vortex induced vibration and, in so doing, have furthered the understanding of the phenomenon, but they are also capable of the relatively simple task of predicting the peak amplitude of vibration.

The results from all wake oscillator model variants can be expressed in the form:-

$$\frac{Y}{D} = f(K_s, St, \psi(z)) \quad (2)$$

Where:

Y/D is the peak cross-flow Reduced Amplitude.

K_s is the Stability Parameter.

St is the Strouhal number.

$\psi(z)$ is the mode shape.

(K_s and St are defined in chapter 2.)

For example, Iwan showed that the Iwan & Blevins wake oscillator model could be used to predict peak Reduced Amplitudes using (Blevins, 1977):-

$$\frac{Y}{D} = \frac{0.07 \gamma}{(K_s + 1.9) \cdot St^2} \left(0.30 + \frac{0.72}{(K_s + 1.9) St} \right)^{1/2} \quad (3)$$

Where the mode shape correction factor (γ) is defined as:

$$\gamma = \psi_{\max}(z) \left(\frac{\int \psi^2(z) \cdot dz}{\int \psi^4(z) \cdot dz} \right)^{1/2} \quad (4)$$

Despite the undoubted usefulness of this type of model there are limitations. In particular, conventional wake oscillator models are inherently two dimensional and assume full vortex shedding correlation and a single mode response. The method also relies on experimentally derived coefficients for its calibration.

2.1.3 Random excitation models.

Perhaps the most serious limitation with the wake oscillator model is its inability to model three dimensional effects. In response to this shortcoming, Blevins & Burton (1975) decided to represent the fluctuating fluid excitation force as a random vibrating variable (see also Blevins, 1977).

The lift force is actually represented by an oscillator with a narrow band power spectrum and the bandwidth is chosen to match that of the vortex shedding.

The magnitude of the excitation lift force is considered to be a continuous function of amplitude and its axial correlation is represented by a characteristic length which increases uniformly with amplitude.

The model can be used to predict peak resonant amplitudes during vortex induced vibration; the form of the relation ship (equation 5) is identical to that of the wake oscillator, except for the additional dependence on the cylinder's Slenderness Ratio (L/D) which is the result of the inclusion of correlation length in the model.

$$\frac{Y}{D} = f(K_s, St, \psi(z), \frac{L}{D}) \quad (5)$$

In cases where the correlation length is less than the length of the cylinder the cylinder's Slenderness Ratio will affect the percentage of the cylinder which is exposed to the correlated lift force and, therefore, will affect the amplitude of vibration.

Once full correlation is reached, which is assumed to occur at a Reduced Amplitude of 0.5, the dependence on the Slenderness Ratio disappears.

The random excitation model tends to predict peak amplitudes which are about 30% lower than the wake oscillator for low levels of damping (i.e. self limiting amplitudes). In cases where the damping is high and the Reduced Amplitudes are below 0.5, the difference increases because of the difference in the way in which the correlation length is treated by the two models.

Whitney & Nikkel (1983) have used a random excitation model, similar to the Blevins & Burton method described above, to predict the dynamic response of marine risers in sheared ocean current.

A method of determining the length and position of the axial regions of correlated fluid excitation force was proposed which was based on the riser's local Reduced Velocity for a single vibration mode (Reduced Velocity is defined in chapter 2). It was assumed that, as was the case for uniform flow, excitation would occur in regions where the local Reduced Velocity lay in the range $4.0 < V_r < 10.0$.

The method is repeated in order to analyze as many modes of vibration as is thought necessary. The overall cylinder mode shape is calculated by summing the individual mode shapes using a sum of squares method.

A different type of random excitation model has been developed by Vandiver & Chung (1988) for predicting the response of long cables in sheared flow. The cable is modelled as a string with constant mass per unit length and axial tension; the equation of motion is, therefore, a derivative of the wave equation.

The model is based on the assumption that, in a sheared flow, the fluid excitation can be represented by a spatially distributed random variable which is a function of the (assumed) vortex shedding frequency and correlation length.

Further, it is assumed that lock-on does not occur because of the presence of the shear flow and the very close spacing of the cables natural frequencies which, together, lead to a broad band response (i.e. strumming).

Good agreement was obtained between model predictions and the experimentally measured dynamic response of a slender cable (17.7 m long with a diameter of 28.6 mm) which was exposed to a variety of highly turbulent sheared flows (the r.m.s. turbulence level varied between 10% and 20% of the mean flow).

There is no doubt that random excitation models have been successful in enabling axial variations in the fluid force to be modelled, however, as with wake oscillator models, experimental results are essential in determining the value of some of the model input parameters.

2.1.4 Energy balance method.

King (1974) and Vickery & Watkins (1964) have both used an energy balance approach to develop an analytical expression for the peak amplitude of vortex induced vibration which takes the form:-

$$\frac{Y}{D} = f(C_{le}, V_x, K_s, \psi(z)) \quad (6)$$

C_{le} is defined as the effective lift coefficient:-

$$C_{1e} = \frac{\int C_1(z) \cdot \psi(z) \cdot dz}{\int \psi(z) \cdot dz} \quad (7)$$

Brooks (1987), of the Chevron oil corporation, developed an energy balance method in order to predict the maximum amplitude of vibration of a marine riser fitted with buoyancy modules.

(Buoyancy modules are foam collars, usually having a diameter twice that of the bare riser, which are attached to the riser at discrete intervals in order to support the weight of the riser, thereby lessening the amount of axial tension which has to be applied at the surface platform.)

Brooks developed a relationship between Reduced Amplitude (Y/D) and the effective hydrodynamic lift coefficient (C_{1e}) using an energy balance which involved three terms:-

$$\begin{array}{lll} \text{Energy input} & & \text{Energy dissipated} \\ \text{from vortex} & = & \text{by structural} \\ \text{shedding} & & \text{damping} \end{array} + \begin{array}{l} \text{Energy dissipated} \\ \text{by hydrodynamic} \\ \text{damping} \end{array}$$

The relationship can be represented as a curve with C_{1e} as the input variable, and Y/D the output variable, neither of which are known.

Blevins (1977) has published a different relationship between these two variables which is based on the work of King (1974); the curve is shown in Fig. 2.11. This empirically derived relationship represents the effective lift coefficient which is present during vortex induced vibration for a range of peak Reduced Amplitudes.

The intersection of the two curves represents the only stable operating point for each particular case; at this point the energy input to the cylinder is exactly balanced by that absorbed in the structural and hydrodynamic damping.

This method of predicting the amplitude response has, according to Brooks, been used by Chevron to successfully determine the optimum position of buoyancy modules on two marine risers off the West coast of the Shetland Isles.

2.1.5 Empirical methods.

Because of the complexity of vortex induced vibration and the uncertainties and imperfections which are contained in the models described above, great reliance is still placed on empirical models when it comes to the solution of practical vortex induced vibration problems. Some of the empirically based prediction techniques are described briefly below.

For uniform, sub-critical flow and single mode vibration there is a well known empirical relationship between the maximum amplitude of vortex induced vibration and the structures Stability Parameter (King, 1974, see also Chapter 2 of this thesis where the relationship is shown in Fig. 2.15).

Although this relationship is not a mathematical model, there is a theoretical argument, which is based on an energy balance during resonant vibrations, for the dependence of peak resonant amplitudes on the Stability Parameter (Chapter 2, section 2.3.2).

The prediction of amplitude response in a sheared flow is less well established. Vandiver (1985) has developed a model to predict the effects of sheared flow on vortex induced vibration which is based on the assumption that distinct regions of hydrodynamic excitation and damping exist along the length of the cylinder. The model assumes that the excitation lengths are known in advance, but offers no suggestion as to how they might be determined.

Vandiver equates the effects of the shear with an increase in effective Stability Parameter and derives an expression which allows the equivalent Stability Parameter to be calculated, provided certain assumptions are made.

The maximum amplitude of vortex induced vibration in the sheared flow can then be determined from the established empirical relationship between Stability Parameter and Reduced Amplitude (Fig. 2.15).

Lamb & Walker (1987) tested the Vandiver method by carrying out a series of large scale experiments with uniform and sheared flow. For stepped sheared flow good agreement was found between theory and experiment, although for linear sheared flow the Vandiver model overestimated the level of vibration, particularly for low rates of shear.

Humphries (1988) also showed the Vandiver model to over-predict the single mode response of a large scale model cylinder which was subjected to linear sheared incident flow.

In both the above cases the Whitney & Nikkel method was used to estimate the position of the excitation and damping regions along the length of the cylinder and it was concluded that this was responsible for the over-prediction.

2.2 Drag Force.

2.2.1 Uniform flow.

As discussed in Chapter 2, Skop, Griffin & Ramberg (1977) have developed a method which can be used to predict the drag force on a vibrating cylinder. They showed that data from a variety of sources could be reduced to an empirical relationship :-

$$\frac{C_d}{C_{ds}} = 1.0 + 1.16 (W_r - 1.0)^{0.65} \quad (8)$$

Where:

- C_d = Drag coefficient of vibrating cylinder.
 C_{ds} = Drag coefficient of equivalent stationary cylinder.
 W_r = Wake response parameter, defined below.

$$W_r = \left(1 + 2 \frac{Y}{D}\right) \cdot \left(\frac{f}{f_s}\right) \quad (9)$$

Where:

- Y/D = Reduced Amplitude
 f = Frequency of cylinder vibration
 f_s = Frequency of vortex shedding from an equivalent stationary cylinder

As more data became available, equation 8 was updated:-

$$\frac{C_d}{C_{ds}} = (0.124 + 0.933 W_r) \quad (10)$$

At lock on, the cylinder vibrates at its natural frequency (i.e $f=f_n$) and $f/f_s = 1/(St \times V_r)$. Equation 10 is, therefore, often quoted as:-

$$\frac{C_d}{C_{ds}} = 0.124 + 0.933 \left(\frac{1 + 2 \frac{Y}{D}}{St \cdot V_r} \right) \quad (11)$$

Griffin's method for predicting the drag amplification due to vortex induced vibration is well established and commonly used, although it is known to over-predict the degree of amplification for large vibration amplitudes (Humphries & Lamb, 1986, Humphries, 1987).

As reported by Humphries, some confusion exists as to the method of applying Griffin's technique. In the original paper by Skop, Griffin & Ramberg (1977) it is suggested that the local Reduced Amplitude is used to calculate the response parameter, however, in a later paper by Griffin (1985) the response parameter is based on the global maximum Reduced Amplitude.

Humphries (1987) tested both methods and recommended that, for ease of application, the global Reduced Amplitude should be used. He also recommends the following Strouhal number values:-

For sub-critical Reynolds number,	St = 0.2.
For critical Reynolds number,	St = 0.1.
For super-critical Reynolds number,	St = 0.3.

2.2.2 Sheared flow.

Humphries (1987) has used Griffin's method to predict drag amplification in sheared flow and found considerable scatter in his experimental plot of drag amplification versus wake response parameter.

Humphries found that equation 11 generally overpredicted the drag amplification and could be used to define the upper bound.

The lower bound is described by:-

$$\frac{C_d}{C_{ds}} = 0.05 + 0.40 \left(\frac{1 + 2 \frac{Y}{D}}{St \cdot V_r} \right) \quad (12)$$

In both equations:-

Y/D is the maximum Reduced Amplitude.
 St as for uniform flow (see 2.2.1 above).
 V_r is based on the average velocity.

3 DEVELOPMENT OF AN AMPLITUDE RESPONSE MODEL.

3.1 Introduction.

It is probable that developments of the pseudo-rigorous flow field models, described in section 2.1.1 above, will continue, and this may lead to a model which is capable of incorporating three dimensional effects and cylinder motion.

Such a model would offer the ultimate degree of sophistication and would allow the greatest degree of freedom in the definition of the fluid-structure system.

The current state of development of the flow field model is, however, still a long way off that required for modelling the type of system with which this project is concerned

and, consequently, this type of model was not considered suitable for the purposes of this thesis.

Similarly, the wake oscillator models were also considered to be unsuitable because of their inherent two dimensionality.

The Vandiver & Chung approach, although undoubtedly attractive for highly sheared flows and very slender cylinders, was not considered suitable because of its inability to deal with lock-on.

The random excitation method of Blevins & Burton was considered suitable for development, given its ability to cope with axial variation in the input parameters, but was rejected in favour of an energy balance method because of the greater flexibility and ease of implementation which it offered.

The objective of this work was centred around the prediction of the maximum steady state amplitude of vibration, and this is most efficiently accomplished by considering an energy balance, such as used by Brooks. The method is generally applicable to a wide variety of physical situations and, in particular, is ideal for the inclusion of axial variations in the input parameters such as those produced by sheared incident current and the application of certain types of suppression device.

3.2 Outline of the Model.

The amplitude response model (ARM) consists of the following principal elements:-

a) Structural motion

The cylinder is assumed to vibrate harmonically in one of its normal modes.

b) Energy formulae

In general, there are three principal forces acting on the structure:-

- Hydrodynamic excitation caused by the vortex shedding in regions of lock-on.
- Hydrodynamic damping caused by the relative motion between the structure and the fluid in regions outside the lock-on range.
- Structural damping caused by internal friction.

Formulae are derived for the work done by each of these forces during one complete cycle of the cylinder's vibration.

c) Energy balance solution

The law of conservation of energy is used to form an energy balance which results in an expression of the form:-

$$\frac{Y}{D} = f(K_s, V_r, \psi(z), C_l(z)) \quad (13)$$

Where:

$C_l(z)$ is the local lift coefficient.

Experimental results are used to derive an expression for $C_l(z)$ and, since K_s , V_r and $\psi(z)$ are known or can be estimated, equation 13 can be used to determine the Reduced Amplitude. A computer program has been written to implement the method. An iterative technique is used because of the dependence of the lift coefficient on the amplitude of vibration.

A full description of the ARM is presented in the following three sections of the thesis.

3.3 Bare Cylinder in a Uniform Flow.

Although Brooks has used an energy balance method for predicting the peak amplitude response of specific riser configurations, no general form of the method has yet been developed (to the author's knowledge), either for bare or suppressed cylinders.

The first stage in model development was, therefore, concerned with the relatively simple case of a bare riser undergoing vortex induced vibration in a single mode and in a uniform velocity profile. The complicating effects of current shear and suppression device coverage are dealt with later in sections 3.4 to 3.6.

3.3.1 Structural motion.

Consider a flexible cylinder, located by end constraints, which is exposed to a uniform free stream velocity (U) and which, as a result, undergoes peak amplitude vortex induced vibration in a plane orthogonal to the flow (i.e. in the 'yz' plane, Fig. 5.1).

The cylinder is assumed to oscillate in one of its normal modes, such that the local maximum amplitude ($Y_l(z)$) is related to the global maximum amplitude (Y) by a mode shape parameter ($\psi(z)$):-

$$Y_1(z) = Y \cdot \psi(z) \quad (14)$$

The cylinder is modelled as a one dimensional body which oscillates with Simple Harmonic Motion. Since the in-line motion, which inevitably accompanies cross-flow vortex induced vibration, is relatively small it can be ignored, and the equation of motion for the cylinder is:-

$$y(z, t) = Y_1(z) \cdot \sin(\omega \cdot t) \quad (15)$$

3.3.2 Energy formulae.

As mentioned above, there are, in general, three forces which transfer energy to or from the cylinder as it vibrates; hydrodynamic excitation, hydrodynamic damping and structural damping.

Hydrodynamic excitation is assumed to act over all regions of the cylinder which are subject to locked-on vortex induced vibration. The criterion for establishing the presence of lock-on is based on the local Reduced Velocity.

Under certain ambient conditions, notably those where the incident flow is sheared, parts of the vibrating cylinder may not be locked-on to the local vortex shedding frequency and in these regions hydrodynamic damping will occur.

Structural damping will, of course, be present over the entire cylinder, except at the nodes where there is no relative internal motion.

(a) HYDRODYNAMIC EXCITATION.

As described in Chapter 2, a cylinder which is undergoing vortex induced vibration experiences a periodic force as a result of the vortex shedding process. The cross-flow component of this force provides the excitation energy to sustain the vibration.

During peak amplitude vibration a state of resonant lock-on exists where there is a match between the structures natural frequency and the frequency of the periodic exciting force.

Under these conditions, the hydrodynamic exciting force and the resulting motion of the cylinder vary harmonically, at the same frequency.

For sustained vortex induced vibration to occur the excitation force must lead the

displacement (described by equation 15) by some phase angle (ϕ), so the equation representing the excitation force takes the form (Bearman, 1984):-

$$f(z, t) = F_1(z) \cdot \sin(\omega \cdot t + \phi) \quad (16)$$

Where:

$f(z, t)$ is the local, instantaneous excitation force per unit length.

$F_1(z)$ is the local maximum value of $f(z, t)$.

The maximum vibration amplitude occurs when there is a resonance between the input force and the resulting cylinder displacement. At resonance, the force and displacement terms are 90 degrees out of phase and equation 16 becomes :-

$$f(z, t) = F_1(z) \cdot \cos(\omega \cdot t) \quad (17)$$

Consider a small element of the cylinder, of length dz , as it moves through a small displacement, $dy(z, t)$. The work done on the element ($dw(z, t)$) by the fluid force is:-

$$dw(z, t) = f(z, t) \cdot dz \cdot dy(z, t) \quad (18)$$

The total work done on the element over one complete cycle ($w(z)$) is obtained by integration w.r.t. y :-

$$w(z) = \int f(z, t) \cdot dz \cdot dy(z, t) \quad (19)$$

Over one complete cycle of the vibration, time varies from 0 to $(2\pi/\omega)$ and, therefore, equation 19 may be written as:-

$$w(z) = \int_0^{2\pi/\omega} f(z, t) \cdot dz \cdot \frac{dy(z, t)}{dt} \cdot dt \quad (20)$$

Substituting for $f(z, t)$ using equation 17, and for $dy(z, t)/dt$ using equation 15, after differentiation w.r.t. t gives:-

$$w(z) = \int_0^{2\pi/\omega} F_1(z) \cdot dz \cdot Y_1(z) \cdot \omega \cdot \cos^2(\omega \cdot t) \cdot dt \quad (21)$$

For steady state peak amplitude vibration, which is the subject of this analysis, $F_1(z)$, $Y_1(z)$ and ω are not functions of time and equation 21 can be integrated to give:-

$$w(z) = F_1(z) \cdot dz \cdot Y_1(z) \cdot \pi \quad (22)$$

Equation 22 is an expression for the local work done by the fluid on an element of the cylinder over one complete cycle of vibration. The work done on the whole cylinder over one complete cycle (W) is obtained by integrating equation 22 w.r.t. z over the entire length of the cylinder :-

$$W = \int_0^L F_1(z) \cdot Y_1(z) \cdot \pi \cdot dz \quad (23)$$

The term $F_1(z)$ can be expanded by introducing a lift coefficient, $C_l(z)$ which is defined in the usual manner:-

$$F_1(z) = C_l(z) \cdot D \cdot \left(\frac{\rho \cdot U^2}{2} \right) \quad (24)$$

Substituting for $F_1(z)$ using equation 24, and for $Y_1(z)$ using equation 14, equation 23 becomes:-

$$W = \pi \cdot \int_0^L C_l(z) \cdot D \cdot \left(\frac{\rho \cdot U^2}{2} \right) \cdot Y \cdot \psi(z) \cdot dz \quad (25)$$

Assuming the cylinder's diameter and the fluid density are constant:-

$$W = \pi \cdot D \cdot \frac{\rho}{2} \cdot Y \cdot \int_0^L C_l(z) \cdot U^2 \cdot \psi(z) \cdot dz \quad (26)$$

Since work and energy are synonymous, W represents the energy transferred from the

fluid to the cylinder during one vibration cycle; this is referred to as EHE (Energy from Hydrodynamic Excitation).

(b) HYDRODYNAMIC DAMPING.

Hydrodynamic damping is defined here as the component of the total hydrodynamic force acting on the cylinder which is exactly out of phase with the cylinder's velocity.

It is important to note that the hydrodynamic excitation force described in (a) above is assumed to act over all regions of the cylinder which are deemed to be exposed to locked-on vortex induced vibration, and that the excitation force is defined in terms of the net fluctuating force on the cylinder (i.e. net of any local hydrodynamic damping).

It follows from these definitions and assumptions that, in all regions of lock-on, the hydrodynamic damping is zero.

For the case which is considered here of a bare cylinder in a uniform flow which is undergoing vortex induced vibration the entire cylinder is subject to the hydrodynamic excitation force and, therefore, there is no hydrodynamic damping.

The total energy transferred from the cylinder per cycle to overcome hydrodynamic damping (EHD) is, therefore, zero in this case.

(c) STRUCTURAL DAMPING.

Structural damping in a vibrating cylinder is usually dominated by hysteretic damping. If a cylinder is allowed to vibrate freely following an initial displacement (i.e. the cylinder is plucked) its vibration decays exponentially and, therefore, resembles that of a viscously damped structure (King, 1974).

Each element of the vibrating cylinder, of length dz , can be regarded as a body undergoing simple harmonic motion in its own right. Each element has the following characteristics:-

Mass	= $m(z).dz$.
Damping ratio	= ζ .
Natural frequency	= ω .

$m(z)$ is the mass per unit length, including the hydrodynamic added mass.

Since each element vibrates in unison, exhibiting the same decaying exponential response following a 'pluck', each element must have an identical damping ratio.

Let the viscous damping force per unit length acting on each element be $f(z,t)$. The total

damping force acting on each element is, therefore, $f(z,t).dz$, and this is related to the coefficient of viscous damping by the following expression:-

$$f(z, t) . dz = c . \frac{dy(z, t)}{dt} \quad (27)$$

The coefficient of viscous damping (c) for each element and the damping ratio (ζ) are related by the following expression:-

$$c = 2 . \zeta . \omega . m(z) . dz \quad (28)$$

Equation 27 can, therefore, be written:-

$$f(z, t) . dz = 2 . \zeta . \omega . m(z) . dz . \frac{dy(z)}{dt} \quad (29)$$

The work done on each element by this damping force as it moves through a small displacement ($dy(z,t)$) is:-

$$dw(z, t) = f(z, t) . dz . dy(z, t) \quad (30)$$

The total work done on the element over one complete cycle ($w(z)$) is obtained by integration w.r.t.y:-

$$w(z) = \int f(z, t) . dz . dy(z, t) \quad (31)$$

Over one complete cycle of the vibration time varies from 0 to $(2\pi/\omega)$ and, therefore, equation 31 may be written as:-

$$w(z) = \int_0^{2\pi/\omega} f(z, t) . dz . \frac{dy(z, t)}{dt} . dt \quad (32)$$

Substituting for $f(z,t).dz$ using equation 29 gives:-

$$w(z) = 2 \cdot \zeta \cdot \omega \cdot m(z) \cdot dz \cdot \int_0^{2\pi/\omega} \left(\frac{dy(z, t)}{dt} \right)^2 \cdot dt \quad (33)$$

It has already been shown that the local displacement amplitude is given by:-

$$y(z, t) = Y_1(z) \cdot \sin(\omega \cdot t) \quad (34)$$

Substituting for $dy(z, t)/dt$ in equation 33 by differentiating equation 34 gives:-

$$w(z) = 2 \cdot \zeta \cdot \omega^3 \cdot m(z) \cdot dz \cdot Y_1^2(z) \cdot \int_0^{2\pi/\omega} \cos^2(\omega \cdot t) \cdot dt \quad (35)$$

The value of the integral is simply (π/ω) , and, therefore:-

$$w = 2 \cdot \pi \cdot \zeta \cdot \omega^2 \cdot m(z) \cdot dz \cdot Y_1^2(z) \quad (36)$$

Equation 36 is an expression for the work done by the structural damping force on an element of the cylinder over one complete cycle of vibration. The work done on the whole cylinder over one complete cycle (W) is obtained by integrating equation 36 w.r.t. z over the entire length of the cylinder :-

$$W = 2 \cdot \pi \cdot \zeta \cdot \omega^2 \cdot \int_0^L m(z) \cdot Y_1^2(z) \cdot dz \quad (37)$$

Expanding the term $Y_1(z)$ using equation 14 and replacing ζ with $(\delta_s/2\pi)$:-

$$W = \delta_s \cdot \omega^2 \cdot Y^2 \cdot \int_0^L m(z) \cdot \psi^2(z) \cdot dz \quad (38)$$

This expression for the energy lost to structural damping over one complete cycle of vibration (ESD) is synonymous with that quoted by King (1974).

3.3.3 Energy balance solution.

Consider the case of a cylinder which is undergoing steady state vortex induced vibration. During each cycle of the vibration there is a perpetual state of imbalance between the input and output energies involved, and the cylinder responds with a sinusoidally varying acceleration, velocity and motion.

Over each complete cycle, however, there is an exact balance between the opposing energies:-

$$EHE = EHD + ESD \quad (39)$$

Which, for the case of a bare cylinder in a uniform flow simplifies to:-

$$EHE = ESD \quad (40)$$

A computer program has been written which evaluates EHE and ESD using the expressions given in equations 26 and 38. An iterative technique is used to find the value of Y which satisfies the energy balance (i.e. equation 40).

The input variables that are required before a solution can be obtained are listed in the table below.

Table 5.1 Input parameters for the ARM computer program.

Symbol	Description	Unit
$C_l(z)$	Local lift coefficient.	-
D	Cylinder diameter.	m
U	Free stream velocity.	m/s
$m(z)$	Mass per unit length of cylinder, including added mass.	kg/m
δ_s	Logarithmic decrement of structural damping.	-
$\psi(z)$	Mode shape.	-
ω	Frequency of locked on vibration.	r/s

In the majority of cases, for both experimental models and full scale structures, all of the above parameters can be easily determined or estimated, with the exception of the local lift coefficient which is known to vary significantly with the amplitude of vibration

(i.e. one of the input variables is a function of the output, Y).

In order to proceed, the relationship between the local lift coefficient and the local amplitude of vibration must be established and, for this information, one has to turn to the results of experimental studies which have sought to measure fluid excitation forces on oscillating cylinders.

There are several examples of such studies, as shown in the table below, which was adapted from Griffin (1985):-

Table 5.2 Investigations into the lift coefficient of a cylinder undergoing vortex induced vibration.

Reference	Fluid	Cylinder type or mode shape	Vibration type
Farquharson & McHugh (1956)	Air	Sinusoidal	Free
Vickery & Watkins (1964)	Water	Pivoted cantilever	Free
Hartlen et al (1968)	Air	Cantilever	Free
Griffin & Koopman (1977)	Air	Rigid	Free
Sarpkaya (1978)	Water	Rigid	Forced
King (1977)	Water	Cantilever	Free
Simmons & Cleary (1979)	Air	Rigid	Forced

Griffin (1985) has plotted the results of all the separate studies summarized in table 5.2; the graph is reproduced here as Fig. 5.2 (The abscissa, $Y_{EFF}(\max)$, is explained below). Although certain trends are apparent, there are quite large discrepancies between the results of the 7 different researchers.

Of course, since the lift coefficient is known to depend on the amplitude of vibration, one would expect the mode shape to affect the data points. Fig. 5.2 contains results for rigid cylinders, pivoted cantilevers, conventional (i.e. encastre) cantilevers and a pin mounted cylinder with a sinusoidal mode shape and, therefore, some of the discrepancies can be assigned to mode shape effects.

Another source of discrepancy is the difference in the experimental conditions. For example, one would not necessarily expect a cylinder which is undergoing natural vortex induced vibration to behave in exactly the same way as one forced to vibrate by mechanical means, although the available evidence suggests that the flow patterns around freely vibrating cylinders are the same as those around cylinders which are forced to

vibrate, provided the amplitude of vibration and Reduced Velocity is the same in each case (Bearman 1984).

Griffin (1985) has attempted to correct for the differences in mode shape by plotting the lift coefficient against a modified Reduced Amplitude, based on the mode shape correction factor developed by Iwan from his study of the wake oscillator mathematical model. This method of presentation has gone some way toward reducing the spread in the data points.

(The Modified Reduced Amplitude is denoted by $Y_{\text{EFF}}(\text{max})$ and is simply a function of Reduced Amplitude and mode shape. For a rigid cylinder, the mode shape correction factor is unity, and the Modified Reduced Amplitude and Reduced Amplitude are, therefore, equivalent (Griffin, 1985).)

Blevins (1977) assumed that the local lift coefficient ($C_l(z)$) could be described by a quadratic function of local amplitude:-

$$C_l(z) = a \cdot Y_l^2(z) + b \cdot Y_l(z) + c \quad (41)$$

He then used some of the data points shown in Fig. 5.2 to determine the coefficients a, b and c. The peak amplitudes predicted by this function are given in table 5.3 below.

It is not possible to infer the local lift coefficient from the results of Vickery and Watkins, Hartlen or King without conditioning the result by first assuming the form of the relationship (e.g. Blevins quadratic function) because of the mode shape of the cylinders used in their studies. Indeed, this is true for any mode shape such that $\psi(z) \neq 1$ because there are an infinite number of solutions which, when integrated to include every element of the cylinder, would result in the correct relationship.

The approach adopted here, like that of Blevins, is based on the premise that the maximum value of the local lift coefficient ($C_l(z)$) is determined uniquely by the local Reduced Amplitude ($Y_l/D(z)$). (In general, the lift coefficient will also vary with Reduced Velocity, but the maximum value, which occurs in the range $6.5 < V_r < 8.0$ (Chapter 2, 2.3.3) depends on the amplitude of vibration.)

The method used here does not, however, rely on there being a preconceived type of relationship, such as the quadratic function used by Blevins. Instead, the results from the tests on the rigid cylinders are used directly to construct the relationship between $C_l(z)$ and $Y_l(z)$. This method automatically eliminates the effect of mode shape because $\psi(z) = 1$ for a rigid cylinder.

Implicit in the method is the assumption that the rigid cylinders used to generate the data points in Fig. 5.2 were sufficiently long to adequately model the increase in correlation length which occurs along the axis of a slender cylinder as the amplitude of vibration

increases.

Three of the six researchers listed in table 5.2 used rigid cylinders (Griffin & Koopman, Sarpkaya, and Simmons & Cleary) and the data points from these tests are plotted in Fig. 5.3.

Because each data point requires a whole series of tests to be carried out, the number of data points available from each reference was limited, so much so that it was difficult to infer the required relationship between C_l and Y/D . Fortunately, there were other sources of information which could be used to supplement these experimental data points:-

- Bearman (1978) found that the fluid excitation force on a vibrating rigid cylinder (as measured in terms of the local fluid pressure which fluctuated in phase with the cylinder's velocity) began to decrease when the Reduced Amplitude exceeded 0.44, and became zero for $Y/D = 1.3$. This data point (1.30, 0) has been added to Fig. 5.3.
- Walshe (1962) carried out tests on a rigid cylinder (Slenderness Ratio 7.8) to investigate the lift coefficient at small Reduced Amplitudes and found that, as Y/D became close to zero (< 0.01), the lift coefficient was in the range 0.17 - 0.32. The data point (0, 0.32) was, therefore, added to Fig. 5.3 to conservatively represent Walshe's findings.

As shown in equation 43 below, the relationship between Y/D and C_l is complicated by a dependence on the Reduced Velocity. Unfortunately, the data points shown in Fig. 5.3 were not all obtained at the same value of Reduced Velocity. Before using the data points to construct a continuous relationship between Y/D and C_l , it was, therefore, necessary to standardise them by correcting for differences in the Reduced Velocity at which each point was obtained.

Using equation 43, below, it is easy to show that, for a rigid cylinder exposed to a uniform flow which has a particular value of Stability Parameter, the peak Reduced Amplitude is proportional to the product ($V_r^2 \times C_l$). In order to standardise the data points in Fig. 5.3 it was, therefore, necessary to multiply each one by the factor $(V_{rb}/V_{ra})^2$, where:-

V_{rb} = The common base Reduced Velocity used to standardize all the data points.

V_{ra} = The actual Reduced Velocity at which the data point was measured.

When vortex induced vibration occurs in water, peak amplitudes usually coincide with a Reduced Velocity in the region of 6.5 to 8.0 (Chapter 2, 2.3.3), and a value of 6.5 was, therefore, selected for V_{rb} . The 'new' data points formed as a result of applying the Reduced Velocity correction factor are shown in Fig. 5.4.

The relationship between Y/D and C_l was formed by linearly interpolating between the data points shown in Fig. 5.4 so as to form an envelope which encompassed all the points. The resulting curve (points A to C, Fig. 5.4) is, therefore, conservative.

The general shape of the curve A-C can be explained as follows. From A to B there is a progressive increase in the degree of vortex shedding correlation, as well as an increase in the strength of the vorticity, until full correlation is achieved at point B.

Thereafter, as the amplitude of vibration increases, the cylinder is subjected to an increasing amount of damping whilst the strength of the shed vortices cannot increase above a certain limit and, consequently, the effective local lift is reduced.

For Reduced Amplitudes greater than 1.3 it was assumed that C_l became increasingly negative following an extrapolation of the line BC in Fig. 5.4. The justification for this extrapolation was purely heuristic, as it was known that, if the lift coefficient did not become increasingly negative, the model would inevitably predict limiting Reduced Amplitudes of, at most, 1.3, regardless of mode shape. This would have contradicted experimental evidence which has shown that Reduced Amplitudes of up to 2.0 are possible for cantilevers.

3.3.4 Model predictions.

(a) EFFECTIVE LIFT COEFFICIENT.

For any mode shape ($\psi(z)$) the effective lift coefficient is defined as (King, 1974):-

$$C_{le} = \frac{\int_0^L C_l(z) \cdot \psi(z) \cdot dz}{\int_0^L \psi(z) \cdot dz} \quad (42)$$

This definition arose because of the lack of knowledge pertaining to the local lift coefficient and, in particular, its dependence on mode shape. It can be used to represent the behaviour of the whole cylinder with out regard for local effects.

Clearly, C_{le} depends on both mode shape and Reduced Amplitude. King (1974) calculated values of C_{le} from his experimental work on model cantilevers, and the curve is shown in Fig. 5.5.

So as to allow a comparison to be made between the predictions of the ARM and King's results the ARM was adapted to calculate values of C_{le} over a range of Reduced Amplitude for the cantilever mode shape. As can be seen in Fig. 5.5, the agreement between the two curves is quite good.

The ARM was also used to generate C_{le} vs Y/D curves for rigid cylinder and sinusoidal mode shapes, and these have been included in Fig. 5.5. Of course, the curve for the rigid mode shape is identical to that used to represent the local lift coefficient since $\psi(z) = 1$.

For cylinder's having a very low level of structural damping the limiting amplitude of vibration is determined solely by the mode shape, and is that level of vibration at which the effective lift coefficient reaches zero.

The limiting Reduced Amplitudes predicted by the ARM are compared with those of Blevins (as discussed above) and the Wake Oscillator model (Blevins, 1977) in table 5.3 below.

Table 5.3 Limiting Reduced Amplitudes.

Mode shape	Predicted maximum amplitudes		
	ARM	Blevins	Wake oscillator
Rigid	1.30	1.0	1.37
Sinusoidal	1.65	1.2	1.58
Cantilever	2.00	1.4	1.79

The merits of these predictions are discussed in section b below.

(b) THE EFFECT OF STABILITY PARAMETER ON REDUCED AMPLITUDE.

One of the most important tests of any mathematical model's practical relevance is its ability to account for the effect of Stability Parameter on Reduced Amplitude.

By using equations 26, 38 and 40 it is possible to show that, for a bare cylinder in a uniform flow:-

$$Y/D = \frac{V_r^2}{4\pi K_s} \cdot \frac{\int_0^L C_l(z) \cdot \psi(z) \cdot dz}{\int_0^L \psi^2(z) \cdot dz}$$

(43)

Where:

$$K_s = \frac{2m_e \cdot \delta_s}{\rho \cdot D^2} \quad (44)$$

$$m_e = \frac{\int_0^L m(z) \cdot \psi^2(z) \cdot dz}{\int_0^L \psi^2(z) \cdot dz} \quad (45)$$

The relationship between Y/D and K_s is, therefore, uniquely determined by the Reduced Velocity and mode shape.

The data points used to construct the $C_l(z)$ vs $Y_l(z)$ curve were all referred to the same base Reduced Velocity of 6.5, and this value was, therefore, used as input to the ARM computer model. (Note that the actual input required by the program is that detailed in table 5.1, and that arbitrary values of U , ω and D were selected so as to give a Reduced Velocity of 6.5.)

The ARM program was run several times to obtain predicted peak Reduced Amplitude of vibration over a range of Stability Parameter ($0 < K_s < 10$) for rigid cylinder, sinusoidal and cantilever mode shapes and the results are shown in Fig. 5.6a, b and c respectively.

Predicted Reduced Amplitude for the cantilever mode shape show good agreement with the measured data points taken from King (1974), (Fig. 5.6a). In general, the ARM prediction is conservative, but this is to be expected because of the conservative approach which was used in the development of the model.

Comparatively little experimental data is available for sinusoidal mode shapes, but the few data points which are plotted in Fig. 5.6b do match reasonably well with the predicted curve. Note that the point marked with a cross was obtained with a very slender cylinder which probably behaved rather like a cable. During strumming, cables are known to have a maximum Reduced Amplitude about $1D$.

Reasonable agreement was also found for the rigid mode shape, although the ARM program generally overpredicted the Reduced Amplitude; this is particularly noticeable at high values of Stability Parameter (Fig. 5.6c).

It is no coincidence that the ARM's overprediction is most pronounced at high Stability Parameter. The reason is concerned with the assumption, within the ARM, that the peak Reduced Amplitude occurs when the Reduced Velocity lies in the range 6.5 to 8.0; this is a fair assumption for large amplitude vortex induced vibrations which coincide with low values of Stability Parameter, but it is known that, as the Stability Parameter

increases, the peak Reduced Amplitude tends to coincide with lower Reduced Velocity (see, for example, Figs 5.8b, 5.9b and 5.10b which were adapted from Humphries (1987)).

If the ARM were to be run again, with lower values of Reduced Velocity for the higher Stability Parameter cases, the predicted Reduced Amplitudes would be significantly reduced, since Reduced Amplitude is proportional to the square of the Reduced Velocity (equation 41 above).

3.4 Bare Cylinder in Sheared Flow.

3.4.1 Background.

The majority of investigations into the effects of sheared flow on vortex induced vibration have been concerned with very slender cables where the lock-on phenomenon can, under certain circumstances, be ignored (section 2.1.3). Such studies are, therefore, not directly relevant to this thesis.

The most recent comprehensive study of the effects of sheared flow under lock-in conditions was carried out by Humphries (1987) who used experimental results to test Vandiver's 'Equivalent Stability Parameter' prediction method. Humphries concluded that there were too many fundamental uncertainties pertaining to the method to enable it to be used in practice and, instead, proposed an empirical method of predicting the peak Reduced Amplitude which was based on the shear parameter, β :-

$$\beta = \frac{D}{U_{ref}} \cdot \frac{dU}{dz} \quad (46)$$

The prediction curve is shown in Fig. 2.16; there is a considerable amount of scatter in the data points and, at low values of shear parameter, the method overpredicts by as much as 50%.

The method assumes that the effects of shear are independent of Stability Parameter and mode shape. Tests by Humphries (1987) support the assumption of Stability Parameter independence, but there is no evidence to show that mode shape can be ignored when predicting the effects of current shear on amplitude response.

3.4.2 A new method for predicting the effect of sheared flow on Reduced Amplitude.

Clearly, there is a need for a prediction method which can offer more accurate results than that suggested by Humphries and which, preferable, is based more closely on the physical processes involved, thereby automatically accounting for the indirect effects of

mode shape and Stability Parameter (if any).

To this end, an empirical method has been developed which is based on the following assumptions:-

- Each local element of the cylinder can be categorized into one of two types, depending on the local Reduced Velocity. If $4.5 < V_r < 10$ then lock-on is assumed to occur. Where the local Reduced Velocity lies outside this range the cylinder is assumed to be subject to local hydrodynamic damping.
- In regions of lock-on the effects of shear can be accounted for at a fundamental level by transforming the relationship between local Reduced Amplitude and lift coefficient. The transformation is based entirely on the local Reduced Velocity. The importance of local Reduced Velocity in accounting for the effects of shear is widely accepted, and is used by Vandiver in the method just discussed. The new method is described in section a below.
- In regions of hydrodynamic damping the energy lost per cycle is based on the cross-flow component of the total drag force on the local element which includes components due to the free stream velocity and the absolute velocity of the element. The damping model is described more fully in section b below.

(a) THE EFFECT OF SHEAR IN LOCKED-ON REGIONS.

The Reduced Amplitude of a bare cylinder in a uniform flow is governed by the Stability Parameter and the Reduced Velocity. It is generally accepted that, regardless of the value of Stability Parameter, cross-flow vortex induced vibration occurs within the range $4.5 < V_r < 10$ with peak amplitudes in the range $6.5 < V_r < 8.0$ (chapter 2).

The amplitude response envelope can, therefore, be represented by the curve shown in Fig. 5.7. Actual values of Reduced Amplitude are predicted by multiplying the peak value, as determined from a knowledge of the Stability Parameter, by the Reduced Velocity multiplier (RVM) shown in Fig. 5.7.

This prediction method for uniform flow was used, in conjunction with the relationship between local Reduced Amplitude and lift coefficient (Fig. 5.4) and equation 43 to calculate a new relationship between local Reduced Amplitude and lift coefficient; one determined by the local Reduced Velocity.

For $\psi(z)=1$, equation 43 gives:-

$$\frac{Y}{D} = \frac{C_1 \cdot V_r^2}{4\pi K_s} \quad (47)$$

(C_1 replaces $C_1(z)$ because, for a rigid cylinder in a uniform flow, the lift coefficient loses its dependence on z)

By definition, the maximum Reduced Amplitude occurs when $V_r = 6.5$ and, in this case, the relationship between the lift coefficient and the Reduced Amplitude is given by the following equation:-

$$C_1 = \frac{1.3 - (Y/D)}{1.78} \quad (48)$$

(Equation 48 describes the negatively sloping part of the curve shown in Fig. 5.4.)

Equation 47 can, therefore, be written as:-

$$(Y/D)_{\max} = \frac{2.46}{K_s + 1.89} \quad (49)$$

Rearranging equation 47 to make C_1 the subject and then replacing K_s using equation 49 gives:-

$$C_1 = \frac{\left(30.86 \frac{Y/D}{(Y/D)_{\max}}\right) - \left(23.75 \frac{Y}{D}\right)}{V_r^2} \quad (50)$$

But, $(Y/D)/(Y/D)_{\max}$ is the Reduced Velocity multiplier, as defined in Fig. 5.7 and, therefore, equation 50 can be used, in conjunction with the information in Fig. 5.7 to model the effect of local Reduced Velocity on the local lift coefficient.

The ARM program was modified to include this Reduced Velocity effect and, as a check on the method, the program was run several times to predict the Reduced Amplitude for a rigid cylinder with a Stability Parameter of 0.001, 0.1 and 1.0. The predicted results, as expected, have the same basic form as the curve in Fig. 5.7, with the peak values of Reduced Amplitude determined by the Stability Parameter, in agreement with Fig. 5.6c.

Changing the mode shape to sinusoidal or cantilever alters the peak value of the Reduced Amplitude in accordance with Fig.s 5.6b and 5.6a respectively, but the basic form of the response is unaffected.

In essence, this method of treating shear is akin to representing the cylinder as a collection of small (rigid) cylinders, each of which behaves as if it were exposed to its own, local, uniform Reduced Velocity. The lift force and energy input per unit length due to hydrodynamic excitation for each element is exactly the same as for an equivalent rigid cylinder in a uniform flow (i.e one with identical Reduced Amplitude and Reduced Velocity).

(b) THE EFFECT OF SHEAR IN NON LOCKED-ON REGIONS.

In quiescent fluid, a vibrating cylinder is subject to viscous or linear damping, provided the amplitude of vibration does not exceed about 1D (King 1974), however, if the ambient fluid is moving, the damping becomes quadratic and the total instantaneous drag force per unit length on the cylinder (F) can be calculated by assuming (Vandiver and Chung, 1987):-

$$F = C_{ds} \cdot D \cdot \left(\frac{\rho \cdot U_r^2}{2} \right) \quad (51)$$

Where:

U_r is the instantaneous relative velocity between the cylinder and the free stream.
 C_{ds} is the drag coefficient of the cylinder when it is stationary.

In the majority of practical situations where vortex induced vibration occurs in a sheared flow, the non locked-on regions are exposed to flowing ambient fluid and it was, therefore, decided that the quadratic damping model should be adopted.

The cross-flow component of F is, from trigonometry:-

$$f(z, t) = C_{ds} \cdot D \cdot \frac{\rho}{2} \cdot \left(U^2 + \left(\frac{dy(z, t)}{dt} \right)^2 \right)^{1/2} \cdot \frac{dy(z, t)}{dt} \quad (52)$$

Expanding $(dy(z,t)/dt)$ using equation 15 after differentiation w.r.t.t and using the same procedure as outlined in sections 3.3.2a and c for developing expressions for energy transfer, it can be shown that the energy lost to hydrodynamic damping per cycle, from each element of length dz is:-

$$w(z) = \frac{\rho}{2} \cdot C_{ds} \cdot D \cdot (Y \cdot \omega \cdot \psi(z))^2 \cdot dz \cdot \int_0^{2\pi/\omega} (U^2 + [Y \cdot \omega \cdot \psi(z) \cdot \cos(\omega \cdot t)]^2)^{1/2} \cdot \cos^2(\omega \cdot t) \cdot dt \quad (53)$$

The integral can be evaluated by making the approximation (Brooks 1987):-

$$\int_0^{2\pi/\omega} (U^2 + [Y \cdot \omega \cdot \psi(z) \cdot \cos(\omega \cdot t)]^2)^{1/2} \cdot \cos^2(\omega \cdot t) \cdot dt \approx \frac{8}{3} \left[\frac{1}{2} \left(\frac{5U}{3\omega} \right)^2 + (Y \cdot \psi(z))^2 \right]^{1/2} \quad (54)$$

The total energy per cycle lost to hydrodynamic damping is obtained by integrating equation 53 w.r.t.z, over the entire length of the cylinder, but only including those regions which are not locked-on (i.e where the local Reduced Velocity does not lie in the range $4.5 < V_r < 10$):-

$$W = \frac{4}{3} \cdot \rho \cdot C_{ds} \cdot D \cdot (Y \cdot \omega)^2 \times \int_0^L \frac{8}{3} \left[\frac{1}{2} \left(\frac{5U}{3\omega} \right)^2 + (Y \cdot \psi(z))^2 \right]^{1/2} \cdot \psi^2(z) \cdot dz \quad (55)$$

The ARM program was modified to evaluate the energy transferred per cycle using equation 55. Integration was carried out using a numerical technique.

At this point in the development of the ARM it was possible to test the program's predictions by direct comparison with experimental work on a series of cantilevered cylinders carried out by Humphries (1987).

Humphries tests were designed to determine the effect of a 3:1 linear shear flow profile ($\beta = 0.02$) on the Reduced Amplitude of the cantilevers over a range of Stability Parameter. His results, some of which are reproduced here as Figs 5.8b, 5.9b and 5.10b, demonstrated that the shear profile reduced the peak response (when compared to the uniform flow case) by a factor of about 0.82 and that this factor appeared to be independent of Stability Parameter.

The ARM was run using the input parameters (table 5.1) appropriate to three of the cylinders used by Humphries and used to predict the response to both uniform and 3:1

linearly sheared flow profiles, over a range of Reduced Velocity. The results are shown in Figs 5.8a, 5.9a and 5.10a and a summary of the predicted and measured peak amplitudes is given in table 5.5, below.

Table 5.5 The effect of linear shear on peak amplitude response.

Stability Parameter	<u>Peak amplitude ($\beta = 0.02$)</u> Peak amplitude ($\beta = 0$)	
	ARM prediction	Measured
0.22	0.84	0.83
0.68	0.84	0.84
2.30	0.85	0.80

In general, the ARM predictions are validated by the experimental results. For the cylinders having a Stability Parameter of 0.68 and 2.3 the ARM accurately predicted the absolute response in the uniform and sheared flow cases. The response of the cylinder with the lowest Stability Parameter (0.22) was underpredicted, although the relative effect of the shear was well modelled. For this cylinder, Humphries measured a peak Reduced Amplitude of 2.4 which is well above the accepted maximum value for cantilevers of 2.0. The underprediction was, therefore, to be expected.

The shift in peak amplitudes toward lower Reduced Velocity which occurred in Humphries results was predicted by the ARM for all three cylinders.

Note also that the ARM predicted that the effect of the linear shear on peak amplitudes would be independent of Stability Parameter, as stated by Humphries (see table 5.5 above).

3.5 Suppressed Cylinder in Uniform Flow.

3.5.1 Model development.

This section of the report is concerned with the extension of the model to include the effects of fitting various types of suppression device to the bare cylinder. At this stage the method remains restricted to uniform incident currents.

It is not appropriate to model suppression devices as elements which merely introduce a degree of hydrodynamic damping into the energy balance because, even when they cover the entire cylinder, most suppression device will still allow a certain amount of vortex induced vibration to occur. It follows that, under certain circumstances, suppression devices must generate a net lift force which transfers energy to the cylinder.

There is no information available on the nature of this lift force for any type of suppression device and, in order to proceed, it is necessary to construct a lift coefficient model based on assumptions and peripheral information.

The following assumptions were made:-

- The lift coefficient has a positive value when $Y/D = 0$. This is necessary for the initiation of vortex induced vibration.
- The lift coefficient increases as the Reduced Amplitude is increased from zero due to an increase in correlation.
- When the maximum correlation length is reached the lift coefficient decreases and becomes zero at a particular value of Reduced Amplitude. Further increase in Reduced Amplitude produces a negative lift coefficient. This assumption makes the lift coefficient self limiting which is essential for finite levels of vibration as the Stability Parameter tends to zero.

These three assumptions result in a lift coefficient - Reduced Amplitude relationship which has the same basic form as that for a bare cylinder (Fig. 5.4). Although the form of the curve is the same, it is obvious that the actual value of the lift coefficient must be less than the bare cylinder value for any degree of suppression to be obtained. From the available experimental evidence it is known that suppression devices generally perform over a range of Stability Parameter and, therefore, the reduction in lift coefficient must occur over a wide range of Reduced Amplitude.

In the absence of any data to the contrary, it seemed reasonable to assume the simplest possible model of lift coefficient reduction; a negative offset applied to the bare cylinder curve shown in Fig. 5.4 which is independent of Reduced Amplitude. The size of the offset (known as the suppression device offset, or SDO) is determined only by the type of suppression device and accounts for differences in the level of performance achieved by different designs.

3.5.2 Model predictions.

It was not possible to use the model to predict the ASR of any suppression device initially because of the lack of fundamental lift coefficient data; there was no way of knowing what value of SDO to use.

It was, therefore, necessary to carry out a series of experiments in order to determine values of the SDO for a number of different types of suppression device. (This method of model calibration has been used as an essential part of many mathematical models of vortex induced vibration. For example, the wake oscillator model relies on experimentally derived coefficients for its calibration.)

The experimental programme, which is described in Chapter 6, included tests on a whole range of suppression device coverage, from 0 to 100% and an investigation into the effect of suppression device position, relative to the vibration nodes and antinodes, on suppression performance. The 100% coverage results were used to calibrate the model and the partial coverage results were then compared with model predictions as a means of validating the ARM.

Model predictions for the effect of partial suppression device coverage are compared with experimentally measured results in Chapter 7.

3.6 Suppressed Cylinder in Sheared Flow.

3.6.1 Model development.

The ARM was adapted to model the case of a suppressed cylinder in a sheared flow by combining the individual effects of shear and suppression device coverage in a simple, linear fashion. Thus, any complex second order interactions between shear and suppression device coverage which may be present were ignored.

The effect of a combination of shear and suppression device coverage was, therefore, modelled at a fundamental level by manipulating the relationship between the local lift coefficient and the local Reduced Amplitude.

The actual method used was as follows:-

- a) Identify regions of lock-on based on local Reduced Velocity. This process was exactly the same as for the bare cylinder in a sheared flow (section 3.4.2).
- b) In regions of lock-on, calculate the local lift coefficient assuming a bare cylinder in the sheared flow. Next, reduce the local lift coefficient by the SDO appropriate for the suppression device which is in use.
- c) In regions of non lock-on calculate the energy absorbed by hydrodynamic damping using equation 55 above. The value of C_d used in equation 55 depends on the particular type of suppression device being used.

3.6.2 Model Predictions.

As for the simpler case of a bare cylinder in a sheared flow, discussed above, it was not possible to obtain predictions from the ARM until the model had been calibrated by experiment so as to determine the appropriate value of SDO for each type of suppression device.

The comparison between model predictions and experimental results is, therefore, presented later in Chapter 7.

4 DEVELOPMENT OF A DRAG FORCE MODEL.

4.1 Bare Cylinder in Uniform Flow.

Griffins method (section 2.2) is well established and, although prone to overprediction for cylinders with high amplitudes of vibration, is generally satisfactory, therefore no model development was deemed necessary.

One of the reasons for overprediction is probably due to the use of the global maximum Reduced Amplitude to predict the wake response parameter. Griffin's method is based on sound physical arguments about the size and strength of the wake behind a vibrating cylinder. These physical effects must exist at a local level and must depend on local parameters i.e. local Reduced Amplitude and Reduced Velocity. Griffin's method was, therefore, used here as originally proposed by Skop, Griffin & Ramberg (1977) with the wake response parameter based on local values of Reduced Amplitude and Reduced Velocity.

Equation 11 (section 2.2.1) was used to calculate the local drag and the total drag on the cylinder was then obtained by integration. A computer program, known as DFM (Drag Force Model), was written to implement the method.

A comparison is made between predicted and measured drag coefficients for the bare cylinder in a uniform flow which was tested as part of this thesis in Chapter 7.

Chapter 7 also contains comparisons between the DFM predictions and measured drag data for the more complex cases involving sheared flow and suppression device coverage, which are discussed below.

4.2 Bare Cylinder in Sheared Flow.

The use of local parameters to calculate the wake response parameter for the uniform case makes Griffin's method easy to apply to sheared flows.

Indeed, since DFM already calculated the wake response parameter based on local values of Reduced Velocity and Reduced Amplitude, no modifications were required to cope with sheared incident flows.

This method has been used by Humphries where it overpredicted the drag whilst successfully modelling the form of the C_d - Re relationship for vibrating cantilevers.

4.3 Suppressed Cylinder in Uniform Flow.

In section 3.4.2b the cross-flow component of the total drag on a bare riser vibrating in a flowing fluid was derived; the drag force was proportional to the relative velocity between the cylinder and the free stream (equation 52).

Using the same basic approach, the in-line component of the total drag force per unit length can be derived:-

$$f(z, t) = C_{ds} \cdot D \cdot \frac{\rho}{2} \left(U^2 + \left(\frac{dy(z, t)}{dt} \right)^2 \right)^{1/2} \cdot U \quad (56)$$

This expression was used by DFM to estimate the drag on suppression devices.

Note that this approach, of basing the drag force on the relative velocity, could have been used for the bare cylinder. This was, in fact, tried but was found to under-estimate the drag amplification.

For a bare cylinder, the drag amplification which accompanies vortex induce vibration is due to two effects:-

- a) The increase in relative velocity due to the vibration.
- b) An increase in the strength of the shed vortices (Chapter 2).

Basing the drag amplification on the relative velocity is, therefore, bound to lead to an under-estimation because it ignores the increase in vortex shedding strength. Griffin's method was used because it is able to account for both effects.

For a suppression device, however, which is designed to reduce the strength of shed vortices, it is reasonable to assume that the drag amplification will be dominated by effect 'a' above, and this is the justification for using equation 56 in DFM.

4.4 Suppressed Cylinder in Sheared Flow.

The suppressed cylinder was treated exactly as in 4.2.2 above. No modification to DFM was required because it already based the total drag force for the cylinder on local values of Reduced Amplitude and velocity. thus, automatically accounting for axial variations in velocity due to the sheared flow profile.

4.5 Calculation of Drag Coefficients.

Once the total drag force on the cylinder had been estimated by the DFM, a drag coefficient was calculated. This calculations was based on equation 4, 7 or 8 (chapter 6), depending on the flow profile.

5 CONCLUSION.

Computer based mathematical models have been developed which enable the behaviour of a cylinder undergoing vortex induced vibration to be determined under a wide variety of operating conditions.

The two pieces of information which are vital to the engineer faced with the problem of vortex induced vibration, viz. the amplitude of vibration and the drag force, can be estimated by the two programs.

The following situations can be modelled by the programs:-

- Bare cylinder in a uniform or sheared incident flow.
- Cylinder with full or partial suppression device coverage in a uniform or sheared flow.

Some preliminary validation of the programs has been undertaken, as described above, but a more comprehensive validation is clearly required. The experimental test programme, described in Chapter 6, was designed to provide the measured data necessary for this validation, and the validation itself is described in Chapter 7.

CHAPTER 6

AN INVESTIGATION INTO THE PERFORMANCE OF THREE TRADITIONAL SUPPRESSION DEVICES

1 INTRODUCTION AND OBJECTIVES.

In chapter 3 it was concluded that much of the previous research into the performance of suppression devices had been carried out in a uncoordinated, piecemeal fashion. As a consequence, it was difficult to make direct comparisons between the results of different researchers and between the performance of different suppression devices.

Chapter 3 also showed that there is a lack of information relating to the suppression of vortex induced vibration in marine risers, which typically operate under the following conditions:-

- Low Stability Parameter.
- Exposed to sheared as well as uniform currents, often in the upper sub-critical or critical flow regimes.
- Single and multi-modal vibration.

It was shown in chapter 4 that vortex induced vibration can have a serious detrimental effect on the exploitation of offshore oil reserves. Suppression devices were shown to be the most economic solution to these problems, but they are expensive to use offshore and can double the cost of drilling a well.

Where suppression devices are used offshore it is common to fit them over virtually the whole length of the marine riser. This practice is believed to be conservative and, in chapter 4, it was demonstrated that significant cost reductions could be made if it were feasible to reduce the amount of coverage whilst maintaining adequate suppression performance.

A computer based mathematical model was developed in chapter 5 in order to predict the effect of partial suppression device coverage on Reduced Amplitude. Such a model, if validated, would be of great benefit in optimising the use of suppression devices. It could be used to predict the smallest coverage which is needed to give the required degree of amplitude suppression, as well as to investigate the effects (economic and technical) of using alternative types of suppression device.

It is against this background that the experimental test programme discussed in this chapter was designed. The main objectives of the test programme were to increase the general understanding of the suppression of vortex induced vibration in marine risers and to test the validity of the mathematical prediction models which were developed in chapter 5.

In order to achieve these main objectives a number of secondary objectives were defined:-

- 1 To directly compare the performance of a representative suppression device from each of the three major classes: topographic, shroud and wake device (Chapter 3).

In order to simulate offshore conditions, tests were required using a slender circular cylinder with a low Stability Parameter.

- 2 To determine the effect of suppression device coverage on performance.
- 3 To determine the effect of current shear on the performance of the suppression devices.
- 4 To determine the effect of Reynolds number on the performance of suppression devices. The regimes of principal interest were the upper sub-critical and critical regimes, as these are most frequently encountered with marine risers.
- 5 To investigate the performance of suppression devices where the vortex induced vibration is multi-modal.

2 EXPERIMENTAL APPARATUS.

2.1 The Deep Water Flume.

The Deep Water Flume (DWF) was designed and constructed in 1983 specifically to enable the vortex induced vibration of long slender structures to be investigated. A photograph of the DWF during its construction phase is shown in Fig. 6.2.1.

The DWF consists of a large steel tank with internal plate steel fabrications (Fig. 6.2.2) in which water is made to circulate as shown in Fig. 6.2.3. The circulating flow is generated by two 275 kW Worthington Simpson pump sets, each of which discharges through a narrow, vertically positioned slit which acts as a jet pump. Jet pumps were required in order to convert the high head - low flow characteristic of the Worthington Simpson centrifugal pumps into the high flow - low head which was required in the circulation loop.

The test section is 6 m deep, 1.5 m wide and 8 m long. A photograph of the test section is shown in Fig. 6.2.4. Velocity through the test section can be controlled in the range 0.5 m/s to 2.1 m/s (uniform profile) using gate valves which are positioned immediately downstream of the centrifugal pumps.

Humphries (1988) measured the turbulence intensity in the test section of the DWF for the uniform velocity profile and found it to be approximately 5% over the velocity range

0.5 to 2.1 m/s.

Three sets of perforated metal screens can be positioned upstream of the test section so as to generate non-uniform flow profiles (Fig. 6.2.3). Set A straightens the flow and reduces any large scale turbulence. Set B is used to generate sheared current profiles which vary linearly over the vertical extent of the test section (known as linear shear profiles or LSPs). Set C is used in conjunction with Set B in order to generate steep LSPs.

In order to produce stepped shear profiles (SSPs), blockage screens are placed in position C only. Details of the LSPs and SSPs which were used during testing are given in Section 4.1.

The DWF has its own cabin which houses the equipment which is necessary for data collection, processing and storage. The facility requires the presence of an engineer and a technician for its operation. Most mechanical fitting operations are carried out with the aid of a team of scuba divers, thus, the need to drain and refill the DWF (a costly and time consuming process) is minimised.

2.2 The Models.

In order to fulfil the objectives of the test programme, two 5.6 m long models were designed and built for use in the DWF.

Both models had a circular cylindrical cross-section, making them geometrically similar to conductor pipes and deep water marine risers with buoyancy modules attached. The effects of full scale complications, such as the presence of choke and kill lines was not investigated during this study.

The use of a simple cylindrical model also allowed the experimental results to be directly compared with previous work, much of which was carried out on circular cylinders. It also facilitated in the development of the mathematical models (chapter 5) which relied on previously obtained experimental data for their calibration; this was only available for circular cylinders.

The majority of the test work was carried out using Model 1; a highly flexible cylinder with an outer diameter of 76 mm. As explained below, this model was designed to vibrate in one or more of its first three normal modes, depending on the free stream velocity setting. Model 1 was used to study the performance of suppression devices during single and multi-modal vibration with Reynolds numbers in the upper sub-critical regime.

In order to study vortex induced vibration in the critical flow regime a second model was required. Model 2 had a diameter twice that of Model 1 so as to increase the Reynolds number. The model was, of course, less slender and less flexible than Model

1 and testing was, therefore, restricted to the first normal mode of vibration. Details of both models are given below.

2.2.1 Model 1.

(a) SPECIFICATION.

The cylindrical model was designed to the following specification:-

- The model must have a low Stability Parameter, preferably with a value of about 0.2, to simulate the worst case offshore conditions (chapter 4, section 1).
- The outer diameter of the model must be large enough to enable upper sub-critical Reynolds numbers to be obtained, but not too large so as to give rise to blockage effects in the test section.
- The natural frequencies of the model should be such that both single and multi-modal vibration can be observed.
- There must be sufficient room inside the model to accommodate four pairs of accelerometers and their leads.
- The model must allow the fitting of a variety of suppression devices. Each suppression device must be designed so that it can be fitted and removed quickly and easily by underwater divers, thus avoiding draining the DWF which would cause long periods of down-time.
- The model must endure a considerable amount of cyclical and steady hydrodynamic loading for the duration of the test programme.

(b) MODEL DESIGN.

The model consisted of a small diameter central steel core which was covered with a set of 24 concentric plastic sleeves (Fig. 6.2.5). In service, a tensile load was applied to the model to give it the required natural frequency and to minimise steady deformation due to the hydrodynamic drag loading.

This tensile load was born almost entirely by the steel core; the plastic sleeves were present only to give the structure its required cylindrical shape. The space between the core and sleeves was used to accommodate the accelerometers and their leads and was allowed to fill with water when the model was installed in the DWF in order to give the required mass per unit length. Photographs of the assembled model can be seen in Figs. 6.2.6 and 6.2.7.

The central core was 5.63 m long, had a diameter of 8 mm and was made of half hard 316 grade stainless steel with good fatigue resistance and a U.T.S. of 1000 MN/m².

Each end of the core was threaded and connected via tapered spacers to an end flange. Great care was taken to minimise stress raisers by careful machining of the core thread and by avoiding any sharp changes in cross-section.

The outer diameter of the model was 76 mm giving an Aspect ratio of 74, sufficient to ensure freedom from end effects. The Reynolds number range during testing was therefore:-

$$Re \text{ (min)} = \frac{U(\text{min}) \times D}{\nu} = \frac{0.5 \times 0.076}{1.3 \times 10^{-6}} = 2.9 \times 10^4$$

$$Re \text{ (max)} = \frac{U(\text{max}) \times D}{\nu} = \frac{2.1 \times 0.076}{1.3 \times 10^{-6}} = 1.2 \times 10^5$$

For a smooth cylinder in the DWF, which has a free stream turbulence level of 5%, critical flow begins at a Reynolds number of about 7×10^4 and is fully established when the Reynolds number reaches 1.2×10^5 .

The gap/diameter ratio was $1.5/0.076 = 18.7$ which was ample to ensure freedom from blockage effects. Every and Humphries (1987) have shown that provided the G/D ratio exceeds 3.0, blockage has no influence on the hydroelastic behaviour of the cylinder.

The sleeves which formed the outer diameter of the model were made from ABS plastic. Each of the 24 sleeves was attached to the central core using two aluminium cruciforms and each sleeve was separated from its neighbour by a gap of 5 mm to allow the model to flex. Segmenting the sleeves in this way ensured that the outer sleeves did not contribute significantly to the bending stiffness of the model. Fibre glass tape was used to cover the gaps between the sleeves so as to present a uniform cylindrical surface to the flow.

In order to locate the suppression devices, particularly the rotating fairings, each sleeve was fitted with a small plastic rim close to its lower edge (Fig. 6.2.5).

Four pairs of accelerometers were fixed to the central core using aluminium clamps. The accelerometers were arranged as orthogonal pairs so as to detect cross-flow and in-line vibration. The axial position of the accelerometers was chosen to coincide with an antinode in each of the first seven natural modes of vibration as shown in the following table:-

Table 6.1 Accelerometer positions for Model 1.

Accelerometer pair	Axial distance from model top	Sensitive to modes
1	L/6	3
2	L/4	2, 6
3	L/2	1, 3, 5, 7,..
4	5L/8	4

The number and positioning of the accelerometers also provided ample backup for measuring the first four modes, should any one of the accelerometers fail.

(c) NATURAL FREQUENCY OF VIBRATION.

The model was designed so that the natural frequencies of the first three modes were controlled by the axial tension, with a negligible contribution from the bending stiffness. (Had the bending stiffness been allowed to play a significant part, the modal frequencies would have been more widely spaced which would not have allowed multi-modal vibration to be observed. It would also have reduced the degree of control over the natural frequencies of the model.)

Because of the insignificant bending stiffness, the model's natural frequencies could be accurately predicted by the simple taut string equation:-

$$f_n = \frac{1}{2L} \sqrt{\frac{T}{m_e}} \quad (1)$$

Where:

- L = length of model cylinder = 5.63 m.
T = axial tension applied to the model (variable).
m_e = effective mass per unit length = 9.5 kg/m (estimate).

The axial tension was set to allow the excitation of the first three normal bending modes in the cross-flow direction. This was judged to be a sufficiently high number of modes to allow multi-modal response to be observed.

Note that a higher number of modes could have been obtained by reducing the axial tension, but this was ruled out because of the increased fatigue loading on the model and

because the reduced tension would have allowed the model to bend excessively under the action of hydrodynamic drag. This would further reduce the life of the model and could also have distorted the dynamic behaviour of the model in an unrealistic manner.

Vortex induced vibration occurs with significant amplitudes over the range $4.5 < V_r < 10$ with a peak response in the region of $V_r = 7$ (King, 1977). The tension was selected in an attempt to attain the whole of the response range for first mode and as much of the range as possible for the second and third modes.

The first mode frequency was, therefore, selected to correspond with a Reduced Velocity of 4.0 at the minimum flume velocity of 0.5 m/s :-

$$f_n = \frac{0.5}{4.0 \times 0.076} = 1.64 \text{ Hz.}$$

The tension required to give this natural frequency is, from equation 1, 3240 N. Using this tension gave the following natural frequencies and ranges of Reduced Velocity for the first three modes of vibration :-

Table 6.2 Expected ranges of Reduced Velocity.

Mode	Natural frequency	Reduced Velocity	
		Minimum	Maximum
1	1.64	4.0	16.8
2	3.30	2.0	8.4
3	4.82	1.4	5.7

Table 6.2 shows that using a tension of 3240 N should result in significant vibration in all three modes. The whole of the Reduced Velocity range of 4 to 10 is covered for first mode, most of this range is covered for second mode and lock-on should be well established in third mode at the higher test velocities.

(d) STABILITY PARAMETER.

The Stability Parameter is defined (chapter 2, section 3.2):-

$$K_s = \frac{2m_e \cdot \delta_s}{\rho \cdot D^2} \quad (2)$$

For Model 1:-

$$\begin{aligned} m_e &= 9.5 \text{ kg/m (estimate).} \\ \rho &= 1000 \text{ kg/m}^3. \\ D &= 0.076 \text{ m.} \end{aligned}$$

Based on experience, δ_s was estimated to be 0.1, giving a Stability Parameter of 0.33. The model would, therefore, have a Stability Parameter which was representative of marine drilling risers and which, according to the empirical prediction curve (Fig. 2.15) would result in a peak cross-flow Reduced Amplitude of about 1.7 when the model was exposed to a uniform current.

(e) FATIGUE LIFE.

Fatigue calculations were performed because of concern that the central steel core of the model would suffer fatigue failure before the test programme had been completed.

The fatigue life of a tensioned structure which is subjected to combined bending and direct stress is difficult to predict because the phenomenon is complex and not fully understood. It was, therefore, desirable that the steel core of the model should not be stressed above its endurance limit, thus giving it an infinite fatigue life.

Calculations were performed which showed that in first and second modes of vibration the stress in the steel core would remain below the endurance limit, even at the maximum predicted amplitude of 1.7 diameters. In third mode, however, amplitudes above 1.0 D would induce stresses above the endurance limit.

Amplitudes were not expected to reach 1.0 D in third mode because the maximum Reduced Velocity would be below that at which peak amplitudes occur. The design was, therefore, judged to be satisfactory.

(f) MODEL FIXING AND TENSIONING ARRANGEMENT.

The model was pin-pin mounted in the DWF. Each end of the model was supported by load cells which enabled the hydrodynamically induced lift and drag forces to be measured. In addition, the top end of the model was connected to a hydraulic tensioning unit which maintained a constant axial tension in the model during testing. The tensioning unit and the load cell measurement system are described below.

The hydraulic tensioning unit consisted of a pump, accumulator and cylinder, and was designed to enable a constant tension to be applied to the model despite the significant axial vibration of the top of the model which would undoubtedly occur during testing (Fig. 6.2.8).

A simple piston type hand pump was used to generate the required hydraulic pressure in the system. A 10 litre accumulator was placed in parallel with the pump. The accumulator contained a flexible bladder with hydraulic oil on one side and Nitrogen gas on the other. During operation the steady state system pressure was dictated by the Nitrogen pre-charge pressure. Any small fluctuation in the volume of hydraulic oil in the accumulator (caused by oscillation of the hydraulic cylinder) had a negligible effect of the hydraulic pressure because of the compressible nature of the Nitrogen gas. Thus, the accumulator maintained a nominally constant hydraulic pressure, and hence model tension, despite significant axial motion of the top of the model.

An Enerpac hydraulic cylinder was used which had a small effective hydraulic area on the high pressure side. This meant that the volume of oil transferred to, or from, the accumulator for a given cylinder displacement was small. Thus, pressure and tension fluctuations were kept to a minimum.

The Enerpac cylinder, like the vast majority of hydraulic cylinders, was not capable of operating reliably when exposed to side loads. Since it was required to resist side loads of up to 500 N during service, modifications were carried out. Two phosphor-bronze bearings were made and fitted internally which allowed side loads to be applied. These bearings also helped to reduce internal friction in the cylinder which would otherwise have caused variations in tension.

The bottom end of the model was attached to the base of the DWF via a 0.5 m long cylinder with a universal joint at either end (Fig 6.2.9). Two orthogonal rods containing load cells were used to locate the 0.5 m long cylinder and to indirectly measure the hydrodynamic forces on the model.

The bottom support structure was contained in a pit below the base of the DWF. This enabled the entire vertical extent of the test section to be occupied by the model cylinders. The pit was covered during testing to make the floor of the flume hydrodynamically clean.

The model's top support was similar, in principle, to the bottom support, the only differences in the design were due to the presence of vertical oscillation in all components of the top support. To accommodate this motion, the orthogonal load cell rods were fitted with small universal joints at each end. The uppermost flange of the top support was connected, via a universal joint, to the hydraulic tensioning cylinder (Fig. 6.2.10).

2.2.2 Model 2.

(a) SPECIFICATION.

- The model must have a low Stability Parameter, preferably no greater than 0.2.
- The outer diameter of the model must be large enough to enable critical Reynolds numbers to be obtained, but not too large so as to give rise to blockage effects in the test section.
- The first mode natural frequency of the model should allow testing to be carried out over as much of the Reduced Velocity range 4 to 10 as possible, consistent with maintaining critical flow around the model.
- There must be sufficient room inside the model to accommodate one pair of accelerometers and their leads.
- The model must allow the fitting of the commercial helical strake made by Albrown Universal Components. Details of this strake are given in section 2.4.2 below.

(b) MODEL DESIGN.

The model consisted of a single PVC pipe (wall thickness 8 mm) with steel fittings at either end. The outer diameter was 0.145 m and the overall length was 5.63 m (Fig. 6.2.11).

A single pair of accelerometers were located on the inside of model at its mid point so as to detect first mode vibration. During testing the model was allowed to fill with water.

Fig. 6.2.12 shows Model 2 complete with the commercial strake. In this photograph the strake is attached with a pitch of 4.5 D and 75% central coverage. The ring clamp and turn-buckle arrangement used to tension the strake can be clearly seen in the photograph.

With an outer diameter of 0.145 the model had an aspect ratio of 39, sufficient to ensure that end effects would not seriously affect the experimental results. The blockage ratio (gap/diameter) was 9.3, well above the critical value of 3 at which the blockage starts to influence the vortex induced vibration of a cylinder which is exposed to a confined stream flow (chapter 2).

Tests were carried out (described in section 7 below) which showed that fully critical flow was established around Model 2 when the velocity in the DWF reached about 1.4 m/s. (Of course, the flow remained critical for all achievable test velocities above this

value.) All tests were, therefore, carried out with flow velocities in the range 1.4 to 2.1 m/s.

(c) NATURAL FREQUENCY OF VIBRATION.

Although the model had appreciable stiffness, it was designed so that some axial tension was needed in order to give the desired natural frequency. This would allow some scope for fine tuning the frequency of the model once installed in the DWF.

The formula for calculating the first mode natural frequency of a member which has bending stiffness (EI) and which is subjected to an axial tension (T) is (Blevins, 1979):-

$$f_n = \frac{\pi}{2L^2} \cdot \left(1 + \frac{T \cdot L^2}{E \cdot I \cdot \pi^2}\right)^{1/2} \cdot \left(\frac{E \cdot I}{m_e}\right)^{1/2} \quad (3)$$

For Model 2:-

$$\begin{aligned} L &= 5.63 \text{ m.} \\ E &= 2.70 \times 10^9. \\ I &= 7.25 \times 10^{-6}. \\ m_e &= 32.0 \text{ kg/m}^3. \end{aligned}$$

Given that the most important aspect of the model's vibrational behaviour is its peak amplitude, the ideal natural frequency is that which produces the maximum amplitude of vibration at a velocity midway between the minimum and maximum test velocities, i.e. $U = 1.75$ m/s. Assuming the peak amplitude occurs at a Reduced Velocity of 7.0, the ideal natural frequency is:-

$$f_n = \frac{U}{V_r \times D} = \frac{1.75}{7.0 \times 0.145} = 1.72 \text{ Hz.}$$

Using equation 3, the tension required to produce a natural frequency of 1.72 Hz is 5900 N. A check was made to ensure that the PVC pipe was capable of withstanding this axial loading and the end fittings were designed to avoid failure due to the shearing forces induced by the tension.

(d) STABILITY PARAMETER.

For Model 2:-

$$\begin{aligned} m_e &= 32.0 \text{ kg/m (estimate).} \\ \rho &= 1000 \text{ kg/m}^3. \\ D &= 0.145 \text{ m.} \end{aligned}$$

Based on experience, δ_s was estimated to be 0.1, giving a Stability Parameter of 0.30. The model would, therefore, have a Stability Parameter which was representative of marine drilling risers and which, according to the empirical prediction curve (Fig. 2.15), would result in a peak cross-flow Reduced Amplitude of about 1.7 when the model was exposed to a uniform current.

(e) FATIGUE LIFE.

The PVC pipe would undoubtedly be subjected to high cyclical stresses during testing, however, no calculations were made to estimate the fatigue life because of the unpredictable nature of the PVC material and because, even if failure were to occur, the cost of repair would be quite low.

(f) MODEL FIXING AND TENSIONING ARRANGEMENT.

This was identical to that used for Model 1 and is described under section 2.2.1 (f) above.

2.3 Instrumentation.

2.3.1 Measurement of velocity.

Velocity was measured at three different depths in the test section of the DWF using three A.OTT propeller meters (Fig. 6.2.13). Each meter contained a reed switch which was normally open, but closed once per propeller revolution. The output from the meter was, therefore, a series of pulses which had a fixed mark but variable space which depended on the flow velocity. The A.OTT meters had a quoted accuracy of $\pm 2\%$ over the velocity range 0.035 to 4.0 m/s.

The pulsed signal was fed into a signal conditioning unit which integrated it, thus producing a voltage which was directly proportional to the flow velocity. The signal was then fed directly to an analogue to digital converter (ADC) before reaching the PDP computer where it was processed and recorded (Fig. 6.2.14).

2.3.2 Measurement of vibration.

(a) MODEL 1.

Four pairs of accelerometers were used to measure the cross-flow and in-line vibration of the model. Wilcoxon Research model 827J single axis accelerometers were used because of their suitability for the harsh environment in the DWF. They were accurate to $\pm 5\%$ in the frequency range 1 to 4000 Hz and had an amplitude range of 0 to 75 g.

At the time of designing the instrumentation these accelerometers were the only ones available which would operate without failure for a reasonable length of time under the severe in-service conditions. Other more accurate types of accelerometer had been tried but were found to require regular servicing or replacement. This degree of unreliability in the model was unacceptable because of the high cost of repair which involved removal of the model from the DWF.

A signal conditioning unit provided the constant current supply to the accelerometers which was necessary for their operation. The unit also acted as a buffer for the output voltage from the accelerometers (the output was 100 mV/g).

Signals from all eight accelerometers were passed from the conditioning unit to a Fylde amplifier, Kemo low pass (anti-aliasing) filter, ADC and finally to the PDP computer (Fig 6.2.14). These signal conditioning/sampling units are described below. On leaving the Fylde amplifier the accelerometer signals were teed off to a switch box which allowed a two channel spectrum analyzer to sample the signals in parallel with the PDP computer.

(b) MODEL 2.

A single pair of accelerometers were used to measure the cross-flow and in-line vibration of the model. Sensotec RDP single axis accelerometers were used in place of the Wilcoxon Research type because of their increased accuracy ($\pm 1\%$) in the frequency range of interest (0 to 20 Hz).

The signal processing chain for the accelerometers was identical to that used for model 1, described above.

2.3.3 Measurement of hydrodynamic forces.

Lift and drag forces on both models were measured using four Sensotec type RM waterproof load cells with a load range of ± 2230 N. The load cells were provided with internal temperature compensation and had a quoted accuracy of $\pm 0.2\%$ full scale.

The load cell used to measure the axial tension in the model was identical to the lift and drag cells except for its working range which was $\pm 11,148$ N.

A signal conditioning unit provided the accurate reference voltage needed by the load cells (10.0 V) and also contained the electronic components necessary to buffer the output voltage from each cell.

Signals from all five load cells were passed from the conditioning unit to the Fylde amplifier, Kemo low pass filter and finally to the ADC and PDP computer (Fig 6.2.14).

2.3.4 Signal processing equipment.

The load cell and accelerometer signals were amplified such that the output signals were in the range ± 5 V for compatibility with the ADC. Fylde FE-351-UA differential d.c. amplifiers were used which were capable of amplification by factors of between 2.0 and 10,000.

The amplifiers had a very low noise performance with high linearity and stability (0.01% per degree Celsius and 0.1% over 12 months) and the frequency response was flat to within 2% from 0 to 10 kHz.

The sensitivity of each of the data channels on leaving the amplifiers is given in table 6.3:-

Table 6.3 Data channel sensitivity.

Channel	Measurement	Sensitivity
1 to 8	Acceleration	2 g/V
9	Tension	2230 N/V
10 to 13	Drag & lift force	446 N/V
14 to 16	Velocity	1 (m/s)/V

Kemo VBF/20 anti-aliasing low pass filters were used on all channels for which the frequency content of the signal was of interest (i.e. accelerometers and load cells, Fig 6.2.14). Four pole Butterworth filters were used with a roll off of 24 db/octave. The corner frequency was set to 20 Hz.

2.3.5 Data capture and digitisation.

The equipment and procedures for data capture and digitisation were the same for both models.

A dedicated PDP 11/73 mini-computer with built in ADC was used to digitise the signals on all data channels and store them on magnetic tape for subsequent analysis.

The ADC used was a DEC ADV11C which was a 12 bit, 16 channel device. The device was set to operate in the range ± 5 V and, therefore, had a resolution of 2.44 mV (1 in 2^{12}).

A specially developed sampling program (JIPSAM) was written by BHRA instrumentation personnel which enabled all 16 channels of data to be sampled in real time. Data was sampled at a rate of 60 Hz for 4.6 minutes at each particular test condition. The frequency resolution was, therefore, 0.004 Hz and the maximum frequency which could be reliably measured was 30 Hz (Pollard, 1979).

The Kemo low pass filter (described above) was used to ensure that spurious frequencies (aliases) were not recorded by the mini-computer. The maximum frequency of interest was that associated with the model's third mode in-line response which was estimated to be 9.8 Hz. This signal would alias with signals of 60 ± 9.8 Hz, i.e. 50.2 and 69.8 Hz, however, the anti-aliasing filter attenuated the signals at these frequencies by 35 db and 46 db respectively, thus rendering any errors due to aliasing insignificant.

An Ono-Sokki CF 920 spectrum analyzer was used in parallel with the mini-computer in order to process and store data from the accelerometers (Fig 6.2.14). This parallel system was used to provide a backup for the valuable vibration data.

In fact, the spectrum analyzer provided a far more efficient method of obtaining the model's amplitude of vibration than the alternative of post-processing the acceleration signals stored on magnetic tape and was, therefore, used as the primary means. The data on magnetic tape was only used where a more detailed analysis was required, or where the information given by the spectrum analyzer was in doubt.

A Hanning window was used when sampling with the spectrum analyzer. Sampling was carried out for 40 s at a rate of 100 kHz. Sixty four averages were made of each power spectrum to improve the accuracy of measurement. The frequency resolution was, therefore, 0.025 Hz and the maximum frequency which could be reliably measured was 50 kHz (Pollard, 1979).

2.3.6 Video.

A colour underwater video camera was mounted on a traversing mechanism 1 m downstream of the model. The camera could be moved over the full vertical extent of the test section which allowed the vibration response of the model to be comprehensively observed. Video recordings provided a valuable insight into the model's behaviour, particularly during complicated multi-modal vibrations. It was also possible to measure the peak to peak amplitude response on the video screen with a surprisingly high degree of accuracy (estimated at 5%).

Although no evidence was found, it was possible that the presence of the video camera in the model's wake affected the fluid loading on the model and, consequently, the camera assembly was removed whilst data was being recorded.

2.4 Suppression Devices.

As discussed in chapter 3, a wide variety of suppression devices are available and the best type to use will, of course, depend on the particular application under consideration. In each case many different factors need to be considered before a device can be chosen. Such factors may include:-

- Degree of vibration in the unsuppressed structure.
- Amount of suppression required.
- Maximum acceptable drag coefficient.
- Speed and direction of current (and variability of direction).
- Constraints on device size.
- Fitting and maintenance requirement/constraints.
- Cost of using device.

To reflect the wide variety of suppression devices which are available and the fact that, in general, no one device can be regarded as a clear favourite for offshore use, it was decided to investigate one representative device from each of the three major classes which were defined in chapter 3. A helical strake, a perforated shroud and a fairing were chosen because of their proven effectiveness and popularity in marine engineering applications.

2.4.1 Suppression devices for Model 1.

All three devices were designed in segmented form to cover the 24 outer plastic sleeves of the cylindrical model (Fig 6.2.5). They were made in two halves so that they could be fitted and removed by divers whilst the model remained in the test section of the DWF.

(a) THE STRAKE.

The strake used in this project was of rectangular section and was designed to the following specification (Fig. 6.2.15):-

- Three start.
- 15 D Pitch.
- 0.1 D strake height.

The design was based on that recommended by Woodgate and Maybrey (1959) although the pitch was increased from 5 D to 15 D for the following reasons:-

- Increasing the pitch reduces the amount of strake material required and makes the manufacture and fitting of the strake easier, at both model and full scale.
- Strake performance does not appear to be very sensitive to the pitch used and preliminary experiments with Model 1 demonstrated that the 15 D pitch gave satisfactory results.

Each strake was made from a rectangular strip of ABS plastic and glued to a split ABS cylinder which was designed to fit tightly over the 76 mm diameter outer sleeves of the model. A special jig was made which allowed the strake helix to be cold formed prior to, and during, gluing. (Note that this would not have been possible if the strake pitch had been much smaller than 15 D.)

When fitted to the model the strakes were clamped tightly using wire ties which prevented relative movement between the strakes and the model.

(b) THE SHROUD.

The design of the shroud used in this project was based closely on the optimum specification given in chapter 3 (Fig. 6.2.16) :-

- 0.1 D radial gap (G).
- OAR of homogeneous perforated region = 36%.
- Overall OAR = 24%. (Small areas had to be left un-perforated for locating purposes).
- 0.13 D hole size.
- Round holes (for ease of manufacture).

Each half shroud was made from a flat sheet of ABS plastic. Holes were drilled using a template and the sheet was then hot formed to the required cylindrical shape. Plastic spacers were then glued on the inside to produce the required 0.1 D gap. When fitted to the model the shrouds were rigidly clamped using cable ties.

(c) THE FAIRING.

A fairing is capable of completely eliminating flow separation and vortex shedding, but to do this it requires a chord length to thickness (c/t) ratio in excess of 10.

In practice, offshore operational considerations usually dictate the use of short fairings and c/t ratios of 2 to 3 are most popular in offshore applications.

The fairing used in this project had a c/t ratio of 2.5 and a radius of curvature (R) of 320 mm (Fig. 6.2.17). Each fairing was made in two halves by hot forming ABS plastic sheets to obtain the desired curvature. These curved sheets were then strengthened by the addition of three ABS ribs. Once assembled around the model, the two fairing halves were bolted together.

The fairings were free to rotate about the model cylinder and were located axially by small thrust rings which were glued to the bottom of the 24 model segments.

A photograph of all three devices, as fitted to Model 1 in the test section of the DWF, is shown in Fig. 6.2.18.

2.4.2 Suppression device for Model 2.

Due to budget limitations it was only possible to test a single suppression device on this large scale model. A sample of commercially available strake, as supplied to offshore operators, was used for all the tests.

The strake was supplied by the Albrown Universal Components company and was made of a tough rubber based compound moulded to give a "T" cross section (Fig. 6.2.19). Through the centre of the "T" section there was a steel rope which enabled the strake to be tensioned once it had been wrapped around the cylinder. This tension forced the strake tight against the cylinder, thus providing a positive means of location.

The strake had a height of 0.1 D and, being flexible, could be fitted to Model 2 with any desired pitch. In fact, two different pitches were tested; a pitch of 4.5 D was used initially to comply with the manufacturer's recommendation and, later, a pitch of 15 D was used to allow a direct comparison to be made with the the strake used on Model 1.

Fig. 6.2.12 shows the strake fitted to Model 2 with the 4.5 D pitch; note the turnbuckles which were used to tension the strake once the desired pitch had been set.

3 DATA REDUCTION.

The JIPSAM sampling program (2.3.5 above) contained subroutines which calculated the maximum, minimum, mean and standard deviation of the signals from all data channels over each 4.6 minute test period.

3.1 Velocity.

The mean velocities measured by the three transducers were used to calculate a reference velocity, the definition of which depended on the nature of the flow profile.

For the uniform and linear shear profiles, the reference velocity was simply the mean velocity over the entire vertical extent of the test section. For the stepped shear profiles, the velocity in the upper region was used (i.e. the high velocity region).

3.2 Vibration Response.

The vibration response of Models 1 and 2 were analyzed predominantly in the frequency domain using the Ono-Sokki digital signal analyzer. This machine automatically transformed the incoming time signals from the accelerometers into frequency spectra using a Fast Fourier Transform process.

After testing was complete, the spectra, which were stored on floppy discs, were processed and analyzed. The dominant frequencies were noted and acceleration spectra were integrated twice to form displacement spectra and, hence, used to establish the zero to peak amplitude of vibration at each discrete frequency. This information was used to plot graphs of Reduced Amplitude versus Reduced Velocity and Reference Velocity.

Vibration response was also studied with the help of a software analysis package which ran on BHRA's VAX 11/750 main frame computer. The package allowed graphs of acceleration, velocity and displacement to be plotted in the time domain.

3.3 Hydrodynamic Forces.

The mean hydrodynamic forces on both models were obtained directly from the results of the JIPSAM sampling program. For the uniform flow profile the steady drag coefficient for each model was defined in the usual manner (chapter 2, equation 7):-

$$C_d = \frac{2F_T}{L.D.\rho.U^2} \quad (4)$$

Where:-

- L = length of model.
 D = diameter of model.
 U = mean velocity.
 F_T = total drag force on the model.

$$F_T = F_{db} + F_{dt}$$

Where:

- F_{db} = mean drag force measured by the in-line load cell at the bottom of the model.
 F_{dt} = mean drag force measured by the in-line load cell at the top of the model.

For the linear shear profiles the situation was slightly more complicated. Consider the local drag force (dF) on a cylinder which is exposed to a linear shear flow:-

$$dF = \frac{\rho \cdot D}{2} \cdot C_d(z) \cdot U^2(z) \cdot dz \quad (5)$$

Where:

- $C_d(z)$ = local drag coefficient.
 $U(z)$ = local free stream velocity.

The total drag force on the cylinder is:-

$$F_T = \frac{\rho \cdot D}{2} \int_0^L C_d(z) \cdot U(z) \cdot dz \quad (6)$$

In order to obtain a single coefficient which represents the drag on the cylinder it is necessary to know, or assume, the relationship between C_d and velocity.

Assuming C_d is independent of velocity, and, hence, independent of z (valid for sub-critical flows only) it is easy to show that:-

$$C_d = \frac{2F_T}{\rho \cdot D \cdot L \cdot U_{ref}^2 \left[\frac{4(\alpha^2 + \alpha + 1)}{3(1 + \alpha)^2} \right]} \quad (7)$$

Where:

U_{ref} = Reference Velocity, defined as average velocity over depth of test section.

and $\alpha = U_T/U_B$ (U_T and U_B are the velocities at the Top and Bottom of the test section.)

All the drag coefficients presented in this thesis for linear shear profiles were calculated using equation 7.

For stepped shear profiles the following equation was used:-

$$C_d = \frac{2F_T}{\rho \cdot D (Z_h \cdot U_h^2 + Z_l \cdot U_l^2)} \quad (8)$$

Where:

Z_h = length of upper excitation region.
 Z_l = length of lower excitation region.
 U_h = velocity in upper excitation region.
 U_l = velocity in lower excitation region.

Equation 8 was obtained directly from the standard definition of drag coefficient (chapter 2, equation 7), again with the assumption that C_d is independent of velocity.

4 COMMISSIONING.

4.1 D.W.F. Flow Profiles.

The sheared profiles described below were selected in an attempt to model full scale profiles which are likely to be encountered by marine risers in service. Stepped shear profiles are common in deep water, where two or more water masses with differing velocities co-exist. Linearly sheared flows are particularly common where there is a well developed boundary layer adjacent to the sea bed (see, for example, Gardner & Cole, 1982).

All the sheared current profiles used in this project were generated by fitting blockage screens at positions B and/or C in the DWF (Fig. 6.2.3) as described below. The screen configuration needed to produce each individual flow profile was found largely by trial and error.

Flow profiles were measured by traversing a propeller meter vertically over the full 5.6 m depth of the test section. Velocity measurements were recorded at 0.5 m increments. The position of measurement coincided exactly with that which the models would adopt during testing (Fig. 6.2.3).

With only the flow straightening screens in position, a uniform velocity profile was achieved, as shown in Fig. 6.4.1.

Three rows of screens, with different open area ratios, placed in position B (Fig. 6.2.3) were used to produce a linear shear profile where:-

$$\frac{U_T}{U_B} = \alpha \approx 1.5 \quad (9)$$

(The subscripts refer to the Top and Bottom of the test section.) This profile was known as the LS15 profile and is shown in Fig. 6.4.2. By placing additional screens, again having various open area ratios, in position C (Fig. 6.2.3) a steeper profile was achieved where:-

$$\frac{U_T}{U_B} = \alpha \approx 3.0 \quad (10)$$

This profile was known as the LS30 and is illustrated in Fig. 6.4.3.

A series of stepped shear profiles were generated by placing screens with a fixed open area ratio in position C only. Three 0.5 m high screens positioned at the bottom of the test section produced a high velocity region which spanned the top 55% of the test section (Fig. 6.4.4). This profile was known as the SS55.

Increasing the number of screens progressively decreased the vertical extent of the high velocity region through 40%, to 24% (Figs. 6.4.5 and 6).

4.2 The Models.

4.2.1 Air Tests.

Air tests are tests carried out on the model when it is surrounded by air rather than water. Air tests were carried out on Models 1 and 2 to determine the level of structural damping present, and to check on the accuracy of the predicted natural frequencies.

(a) MODEL 1.

After being weighed, Model 1 was installed in the empty DWF. The model was then tensioned and plucked to promote first mode vibration. The frequency of vibration was measured and a time history of the decaying vibration was recorded. This test was repeated for a range of tensions which spanned the working tension of 3200 N.

The measured natural frequency was compared with that predicted from the simple taut string equation (equ. 1) and the logarithmic decrement of damping in air was measured for each test and used to estimate the model's Stability Parameter. The results are shown in the following table:-

Table 6.4 Results from pluck tests in air: Model 1.

Tension (N)	Natural frequency (Hz)		δ_s	K_s
	Predicted	Measured		
2700	2.77	2.95	0.0247	0.086
3200	3.02	3.25	0.0264	0.092
3700	3.25	3.55	0.0236	0.083

The agreement between the predicted and measured natural frequency was reasonable, validating the use of the simple string equation. The slight under-prediction was probably due to the mass distribution in the model, which was biased towards the ends.

Damping in the model was less than expected by a factor of four with the consequence that the expected peak amplitude of vibration increased from about 1.6 to 1.9 diameters.

(b) MODEL 2.

The procedure described above was repeated for Model 2, although this time only the working tension of 5900 N was used and the predicted natural frequency was calculated from equation 3. The results are shown in the table below:-

Table 6.5 Results from pluck test in air: Model 2.

Tension (N)	Natural frequency (Hz)		δ_s	K_s
	Predicted	Measured		
5900	2.47	2.30	0.048	0.15

Again, the value of damping was underestimated at the design stage and the predicted peak Reduced Amplitude (based on Fig. 2.15) was revised upwards to about 1.8.

4.2.2 Water tests.

With the DWF full of water and the tension set to the working value, a pluck test was performed in order to determine the model's natural frequency in water. This test was performed for both models and the results are shown in table 6.6:-

Table 6.6 Results from pluck tests in water.

Model	Tension (N)	Natural frequency (Hz)	
		Predicted	Measured
1	3200	1.58	1.55
2	5900	1.72	1.69

Equations 1 and 3 were used to predict the natural frequencies of models 1 and 2 respectively. The value of effective mass per unit length used in each case was based on an assumed added mass coefficient of unity. The agreement between the predicted and measured natural frequencies validates this assumption.

4.3 The Instrumentation.

4.3.1 Hydrodynamic Forces.

In order to check the manufacturer's calibration, the load cells were individually bench tested and calibrated in-situ in the DWF. Bench testing simply involved applying known loads to the load cells and comparing the output signals with those predicted by the manufacturer's calibration. There was good agreement between the measured and predicted signals for all four load cells.

In order to check the entire load measurement system, the load cells were calibrated in-situ by applying a transverse load to the model and recording the load cell signals

using the JIPSAM sampling program. In order to calibrate all four load cells, the force was applied, first in the cross-flow, and then in the in-line direction.

The theoretical reaction at the location of each load cell to the applied transverse load was calculated from a knowledge of the various moment arms involved. This allowed a graph of measured versus applied load to be plotted for all four load cells (Fig. 6.4.7).

4.3.2 Vibration response.

There was no accurate method available to calibrate the accelerometers and so the manufacturers calibration figures were used, however, the output from the accelerometers was checked using the underwater video camera. With the model vibrating steadily in first mode at it's peak amplitude (1.53 D), the amplitude was measured using the accelerometers and by measuring the screen image from the video recording. Agreement between the two methods was within 5%.

In order to check the signal processing equipment and the method of data capture and reduction, the signal from the accelerometers was replaced by that from a signal generator. The amplitude and frequency of several artificially generated signals were then measured using the Ono-Sokki signal analyzer, a similar Hewlett Packard machine and the VAX software. The agreement between these three measurements was within 2% for both frequency and amplitude response.

4.3.3 Velocity.

The signal processing equipment was calibrated by replacing the signal from the propeller meters with that from a pulse generator of known frequency. Values of the 'velocity' signal recorded by the JIPSAM program were then compared with the expected values which were calculated from the propeller meters calibration constants (supplied by the manufacturer). The agreement between these two figures was within 2%.

5 ERROR ANALYSIS.

The values of error quoted below are the maximum expected for the quantity under consideration and include the errors due to all the components in the measurement chain (i.e. transducer, signal processing and data capture).

5.1 Measured Parameters.

5.1.1 Drag force.

Measurement of the drag force involved four elements (Fig. 6.2.14), each of which introduced some error, either variable or constant, as shown in the table below:-

Table 6.7 Possible errors in the measurement of drag force.

Element	Error		Information Source
	Variable (+/- %)	Constant (+/- N)	
Load cell	0	2.2	Manufacturer
Amplifier	0.1	0	Manufacturer
Filter	1.0	0	Supplier
ADC	0	1.1	12 bit device

In order to combine these individual error terms to give an overall figure for the accuracy of measurement, it is necessary to turn all the terms into percentages. This requires a value of force to be specified.

Once the individual percentage errors are known, the overall maximum expected error can be determined using the following equation (Brinkworth, 1973):-

$$MEOE = [(E_1)^2 + (E_2)^2 + \dots + (E_n)^2]^{1/2}$$

(11)

Where:

- MEOE = Maximum expected overall error (%).
- E_n = Error introduced by the n^{th} element (%).

The greatest percentage error occurs where the drag force is a minimum. This will occur for Model 1 when the velocity is at its minimum value of 0.5 m/s.

Assuming a drag coefficient of 1.2, the mean drag force on Model 1 would be 64 N (equation 7, chapter 2). Due to the moment arms present in the force measurement system this force would produce a reaction of 51 N in the top load cell and 36 N in the bottom load cell.

The percentage errors can now be calculated and the MEOE is, from equation 11:-

$$\begin{aligned} &[(4.3)^2 + (0.1)^2 + (1.0)^2 + (2.2)^2 \\ &+ (6.1)^2 + (0.1)^2 + (1.0)^2 + (3.1)^2]^{1/2} \end{aligned}$$

(12)

MEOE = +/- 8.5%.

(Note that the errors attributable to both load cells are included in the above equation.)

The drag force on the model during peak amplitude first mode vibration is of particular interest and it is, therefore, useful to consider the error under this condition. Peak amplitude is likely to occur at a velocity of 0.9 m/s and the drag coefficient can be assumed to be of the order of 2.5. The total drag force on the model will, therefore, be 433 N and the actual forces measured by the top and bottom load cells will be 344 N and 246 N respectively.

Following the procedure explained above, the MEOE can be calculated to be +/- 1.7%.

For model 2, with its larger diameter and higher test velocities, the MEOE will, of course, be reduced still further.

5.1.2 Vibration amplitude.

Most of the analysis was done using the spectrum analyzer and this method involved three elements which introduced errors into the measurement (Fig. 6.2.14). Details are given in table 6.8:-

Table 6.8 Possible errors in the measurement of vibration amplitude.

Element	Error		Information Source
	Variable (+/- %)	Constant (+/- m)	
Accelerometer	5.0	0	Manufacturer
Amplifier	0.1	0	Manufacturer
Spectrum analyzer	0.1	0	Estimate

Using equation 11, the MEOE was calculated to be +/- 5.0%.

5.1.3 Velocity.

The elements and errors involved in the measurement of velocity were as follows:-

Element	Error		Information Source
	Variable (+/- %)	Constant (+/-m/s)	
Propeller meter	2.0	0	Calibration cert.
Freq. to voltage	1.0	0	Calibration facility
Filter	1.0	0	Supplier
ADC	0	0.001	12 bit device

The MEOE in the measurement at each position in the test section of the DWF was +/- 2.5%. For the majority of tests, the velocity was obtained by averaging the measurements of all three propeller meters which reduced the MEOE to +/- 1.4%.

5.2 Derived Parameters.

5.2.1 Reduced Amplitude.

The diameter of the model could be measured highly accurately and, therefore, the error in Reduced Amplitude was simply that in the measurement of amplitude; +/- 5%.

5.2.2 Drag coefficient.

The drag coefficient was calculated using an equation of the form:-

$$C_d = \frac{2 \cdot F_d}{A \cdot \rho \cdot U^2} \quad (13)$$

The MEOE in the measurement of velocity has already been shown to be +/- 2.5%. For Model 1 (worst case) the MEOE in the drag force varies from a maximum of +/- 8.5% at the lowest velocity to about +/- 1.7 % during peak amplitude vibrations.

The MEOE in C_d at the lowest test velocity (0.5 m/s) is, therefore:-

$$[(8.5)^2 + 2(2.5)^2]^{1/2} = \pm 9.2\% \quad (14)$$

But at a velocity of 0.9 m/s., which was expected to correspond with peak amplitude vibration in first mode for Model 1, the MEOE reduces to:-

$$[(1.7)^2 + 2(2.5)^2]^{1/2} = \pm 3.9\% \quad (15)$$

Because of the greater drag loading on Model 2, the MEOE in the drag coefficient for this model would be expected to be better than $\pm 3.9\%$.

6 EXPERIMENTAL METHOD.

6.1 Test Procedure.

6.1.1 Model 1.

The axial tension normally applied to the model was released and the tension signal was zeroed at the Fylde amplifier. The JIPSAM program was used to sample and record the signal.

Working tension was then applied to the model using the hydraulic unit. The remaining 15 signal channels were all zeroed at the Fylde amplifier and these were sampled and recorded by running the JIPSAM program for a second time.

The sampled values were then used to create a zero file which would be read by the JIPSAM program during subsequent testing. The JIPSAM program also required access to a file containing the calibration coefficients of all 16 channels. This file was created manually from a knowledge of the sensitivity of each channel in terms of the number of units per bit.

Once the zeroing procedure had been completed the DWF pumps were started and a flow of 0.5 m/s was set through the test section. After a delay of several minutes to allow steady state conditions to be established the JIPSAM program was run and data from all 16 channels was sampled for 4.6 minutes.

On completion of sampling, the velocity was increased in increments of between 0.1 and 0.15 m/s, depending on the model's response. During the three minute wait for steady state conditions to be re-established the output from the previous JIPSAM run was analyzed. The output was a printed table containing the maximum, minimum, mean and standard deviation of all 16 channels. A brief scan of these values enabled any problems with the model or instrumentation to be identified and remedied at the earliest opportunity.

After approximately two hours of testing the maximum velocity of 2.1 m/s was reached and testing was halted. The zeros on all 16 channels were re-checked and recorded.

The above procedure was then repeated starting with a velocity of 2.1 m/s and then decreasing the velocity in increments of between 0.1 and 0.15 m/s until a velocity of 0.5 m/s was reached.

Finally, the underwater camera was lowered into the test section downstream of the model and used to record the vibration of the model at selected velocity settings. During video recording the Ono-Sokki was used to measure the amplitude and frequency of vibration.

6.1.2 Model 2.

The test procedure was the same as that described above for Model 1 with the following exceptions:-

- Only 11 of the 16 data channels were used because of the smaller number of accelerometers in this model.
- The minimum test velocity was increased to about 1.40 m/s to ensure fully critical flow around the model.

6.2 The Test Programme.

6.2.1 Model 1.

A total of 48 tests were carried out (counting incrementing and decrementing velocity tests as a single test) which took six months and 240 man-days to complete. The test programme is summarised in table 6.9:-

Table 6.9 Test programme for Model 1.

Test	Flow profile	Supp. device	Device Coverage (%)
1	Uniform	None	0
2	"	"	"
3	LS15	"	"
4	LS30	"	"
5	SS55	"	"
6	SS40	"	"
7	SS24	"	"
8	Uniform	Shroud	20
9	"	"	25
10	"	"	33
11	"	"	50
12	"	"	75
13	"	"	100
14	Uniform	Fairing	20
15	"	"	25
16	"	"	33
17	"	"	50
18	"	"	100
19	LS15	"	50
20	"	"	33
21	LS30	"	50
22	"	"	33
23	SS55	"	50
24	"	"	33
25	SS40	"	33

26	"	"	25
27	SS24	"	25
28	"	"	20
29	Uniform	Strake	20
30	"	"	25
31	"	"	33
32	"	"	50
33	"	"	75
34	"	"	100
35	LS15	"	50
36	"	"	33
37	LS30	"	50
38	"	"	33
39	SS55	"	50
40	"	"	33
41	SS40	"	50
42	"	"	33
43	SS24	"	20
44	Uniform	"	25
45	"	"	50
46	"	"	75
47	"	"	50
48	"	"	50

Tests 1 to 7 on the bare model were required as a bench-mark by which to judge the performance of the suppression devices in uniform, linear and stepped shear flows. Note that all the flow profiles used in this test programme are illustrated in Fig. 6.4.1 to 6.

Tests 8 to 43 were designed to investigate the performance of the three different suppression devices for different amounts of regular device coverage in uniform and sheared flow. Regular is used here to mean coverage which is placed at regular intervals along the length of the model so as not to be biased toward the model's nodes or

antinodes. This type of coverage is illustrated in Fig. 6.6.1a.

Because of time and cost limitations, tests with the linear and stepped shear velocity profiles were performed on only two of the three suppression devices. The fairing and strake were selected for these tests because of their superior vibration suppression performance over the shroud during the tests involving the uniform velocity profiles.

Tests 44 to 48 were designed to investigate the effect of different amounts of irregular device coverage in a uniform flow. This type of coverage is illustrated in Fig. 6.6.1b and was necessary to investigate the importance of mode shape on device performance. The strake was selected for use during these tests so as to enable a direct comparison with the large scale strake tests which were carried out with critical flow using Model 2.

Note that the first mode of vibration was the only mode of interest for these tests and, consequently, they were carried out over a restricted velocity range (0.5 to 1.3 m/s).

6.2.2 Model 2.

A total of 10 tests were completed as detailed in table 6.10 below. Once set up, each test took approximately 1/2 day to complete but, because a team of sub-aqua divers was required to make changes to the model, the total time for testing each configuration was considerably longer. In fact the test programme took a total of six weeks and 54 man-days to complete.

Table 6.10 Test programme for Model 2.

Test	Strake pitch	Coverage (%)
1	-	0
2	-	0
3	-	0
4	-	0
5	4.5 D	100
6	4.5 D	50
7	4.5 D	25
8	15 D	100
9	15 D	50
10	15 D	25

The Model 2 test programme was designed to establish the behaviour of a bare and suppressed cylinder in the critical flow regime. It was, therefore, necessary to determine the flow velocity at which critical flow became fully established. The easiest way of doing this was to determine the shape of the drag characteristic (i.e. C_d vs U curve) for a stationary version of Model 2. The point at which the drag levelled off at a relatively low value would mark the onset of fully critical flow.

This test had to be done on a stationary cylinder because even small levels of vibration would seriously distort the drag characteristic, making it difficult to determine the onset of the critical regime. The actual level of vibration at which this distortion would become serious is the so called critical Reduced Amplitude which was discussed in chapter 2. For cross-flow vibration the critical value is about 0.05; for in-line vibration it is between 0.01 and 0.02.

Tests 1 and 2 were carried out for the purpose of determining the drag characteristic, using a rigid version of Model 2. It was not possible to adequately restrain the PVC version of Model 2 (described in section 2.2.2 above) so a geometrically identical model was constructed in a stiffer material (aluminium). This model was braced at its mid point to restrain cross-flow vibration, but the bracing did not interfere with the measurement of the (in-line) drag force.

During tests 1 and 2, accelerometers were used to monitor cross-flow and in-line vibration to ensure that the critical levels were not reached.

All the other tests (3 to 10) were carried out using the flexible PVC version of Model 2. Tests 3 and 4 were the bare cylinder reference tests; tests 3 to 8 were designed to determine the strake performance over a range of coverage in a critical flow.

All tests were carried out with a uniform flow profile and, in all tests involving the strake, the strake was applied to the central region of the model (this type of coverage was also used during the Model 1 test programme and is illustrated in Fig. 6.6.1b).

Note that strake pitches of 4.5 D and 15 D were examined. The 4.5 D pitch was recommended by the manufacturer as the ideal, although this was not based on experimental evidence. The larger pitch was selected to match that used during the Model 1 tests. If it proved effective, such a pitch would offer advantages over the recommended pitch since it would make the strake quicker and easier to fit and less strake would be required to cover a given length of cylinder.

7 MODEL 1 RESULTS.

Throughout the test programme the vortex induced vibration of the model was dominated by the cross-flow component although, in general, there was also a coupled in-line vibration at twice the cross-flow frequency. This coupled mode of vibration is one of the hall marks of vortex induced vibration and causes the cylinder to describe a figure of eight motion in the plane which lies perpendicular to it's longitudinal axis (chapter 2, 3.3).

Because the model's behaviour was dominated by the cross-flow vibration, the in-line component is not discussed in detail in this thesis.

7.1 The Bare Cylinder.

7.1.1 Uniform velocity profile (Tests 1 and 2).

Fig. 6.7.1 shows the model's Reduced Amplitude (Y/D) in modes one, two and three over the complete range of test velocity. Each graph contains results from two identical tests, and includes data points for both incrementing and decrementing velocity. In general, the results show reasonable repeatability, and there is no evidence of significant hysteresis.

The peak Reduced Amplitude in first mode was 1.53. Fig. 6.7.1 shows that, at velocities above 1.4 m/s, there was considerable scatter in the data for first mode. In this region the response was dominated by second mode, and the first mode vibration is believed to result from the imperfect excitation of this mode. In any kind of vibrating system with closely spaced modes it is quite unusual to excite a pure single mode. The exciting force will inevitably have some bandwidth and will, therefore, excite a range of frequencies. This is particularly true of vortex induced vibration where the spectrum of the excitation force, although centred on the so called Strouhal shedding frequency, is known to have a significant bandwidth.

In second mode, the peak Reduced Amplitude was 1.21, significantly less than the value for first mode and possible reasons for this are discussed in chapter 7.

At the highest test velocity (2.08 m/s) the vibration was dominated by third mode, although the peak Reduced Amplitude was probably not reached.

The extent to which the vibration was multi-modal can be seen in Fig. 6.7.2 where all three modes are plotted on the same graph. The first mode peak occurred as an almost pure single mode response, but at velocities above 1.2 m/s the model vibrated multi-modally.

The mean drag coefficient for the model, together with that for an equivalent stationary cylinder, is shown in Fig. 6.7.3. As expected, the vibration of the model caused a large

increase in drag, but the reduction of drag coefficient in the critical Reynolds number regime was still apparent.

In order to standardise the data, it is common practice to present Reduced Amplitude as a plot against Reduced Velocity and this has been done for each mode of vibration in Fig. 6.7.4. As was the case in the Reduced Amplitude versus reference velocity plots, each graph shows the results of two separate tests, although this time only the outer envelope of the data points is shown for clarity.

Vortex induced vibration is expected to occur between Reduced Velocities of four and ten with a peak response at a Reduced Velocity of about seven (chapter 2). In general, therefore, the model's response was a classical one, although there are two points which deserve mentioning :-

- The first mode response was unusually large at the upper end of the Reduced Velocity range; the reason for this is not known.
- The response in second mode dropped off at a quite a low Reduced Velocity (8 instead of the expected value of 10). This was probably due to the presence of third mode vibration which became dominant at corresponding velocities.

7.1.2 Linear shear velocity profiles (Tests 3 and 4).

Plots of Reduced Amplitude versus reference velocity for the LS15 and LS30 profiles are shown in Fig. 6.7.5; plots versus Reduced Velocity are shown in Fig. 6.7.7. In Fig. 6.7.7 the corresponding results for the uniform velocity profile are also shown to allow a direct comparison.

The main effect of the velocity shear was to reduce the peak amplitudes of vibration as summarised in the table below :-

Table 6.11 Reduction in peak amplitude due to linear shear.

Linear shear profile	Reduction in peak amplitude (%)	
	Mode 1	Mode 2
LS15	7	48
LS30	26	72 (1)

(1) This figure is non-conservative as the peak may not have been reached.

The large reduction in the peak amplitude of mode 2 can be explained by considering the position of the modal nodes and antinodes relative to the exciting and damping

regions of the sheared velocity profile.

In second mode, whilst the upper of the two antinodes is excited by the high velocity region of the sheared flow, the lower antinode is damped because the velocity is too low to excite this mode of vibration. This provides a high degree of damping.

In contrast, when the first mode antinode is excited the damping regions either side of the excitation region coincide with nodes and are, therefore, far less effective at reducing the amplitude of vibration.

Shear tended to increased the multi-modal nature of model's vibration by extending the range of velocity (and Reduced Velocity) at which significant vibrations occurred (Figs. 6.7.5 and 7).

The effects of the linear shear on drag coefficient can be seen in Fig. 6.7.6. The drag force is reduced when compared to values for the uniform velocity profile which is to be expected because of the reduced level of vibration. In addition, because of the high velocities which were present in the upper part of the linear shear profiles, there was an earlier transition to critical flow. This trend was probably accentuated by the increased level of turbulence which is inherent in a sheared flow. The early transition can be seen in Fig. 6.7.6, particularly for the LS30 profile.

7.1.3 Stepped shear velocity profiles (Tests 5 to 7).

Plots of Reduced Amplitude versus reference velocity for the 55%, 40% and 24% stepped shear profiles are shown in Fig. 6.7.8; plots versus Reduced Velocity are shown in Fig. 6.7.9 which also includes the corresponding results for the uniform velocity profile to enable the effects of the stepped shear to be easily seen.

As for the linear shear profiles, the main effect of the stepped shear was to reduce the peak amplitude of vibration as summarised in the table below :-

Table 6.12 Reduction in peak amplitude due to stepped shear.

Stepped shear profile	Reduction in peak amplitude (%)	
	Mode 1	Mode 2
SS55	20	45
SS40	31	40
SS24	47	66

With the exception of the second mode peak for the SS40 profile, decreasing the excitation length progressively decreased the peak amplitude response, as one would expect. The other major effect of the stepped shear was to significantly increase the upper bound of the Reduced Velocity response range for first mode, and this made the response in the higher modes increasingly multi-modal (Fig. 6.7.8).

As with the linear shear, the second mode response was reduced more than the first. Again, this was probably due to the position of the nodes and antinodes relative to the regions of excitation and damping, as explained above.

There was a tendency for the first mode response to contain a double peak which corresponded to excitation from the upper and lower velocity regions of the stepped shear profile.

Generally, the drag coefficient was decreased by the shear because of the associated decrease in vibration amplitudes and the early transition to critical flow brought about by the increase in free stream turbulence (Fig. 6.7.10).

The stepped shear profiles had a transition region between the lower and upper velocity regions and, for analysis purposes, the low velocity region was defined so as to include the transition region. There was, therefore, a tendency to overestimate the drag coefficient; this is particularly evident at low velocities for the SS55 and SS40 profiles, where the drag coefficient exceeds that for the uniform velocity case (Fig. 6.7.10)

7.2 The Shrouded Cylinder.

All the results from tests where suppression devices were fitted are presented as curves which form an outer envelope around the data points. The data points are not shown so that families of curves can be plotted on a single graph without loss of clarity. The omission of the data points does not result in any significant loss of information because the data exhibited very little scatter and hysteresis (even less than that for the bare cylinder tests).

For all three suppression devices, the ASR's quoted below are for modes one and two. It was not possible to calculate values for mode three because the peak amplitude for the bare model was not reached.

7.2.1 Uniform velocity profile (Tests 8 to 13).

The effect of regular shroud coverage on Reduced Amplitude for modes one, two and three is shown in Fig. 6.7.11. As expected, increasing the coverage progressively decreased the model's peak Reduced Amplitude, although coverages of 50% or less produced only a small amount of suppression. The Reduced Velocity response range was affected little by the presence of the shroud, except for 100% coverage where the upper

bound of the first mode Reduced Velocity response range was significantly reduced.

The amplitude suppression performance is summarised in the table below:-

Table 6.13 ASRs for the shroud in a uniform profile.

Mode	Amount of coverage (%)					
	20	25	33	50	75	100
1	10	9	15	26	43	60
2	9	5	13	20	36	54

The drag coefficient for the shrouded model (Fig. 6.7.12) was influenced by two distinct phenomenon :-

- The reduced levels of vibration afforded by the shroud which tended to reduce the drag coefficient.
- The elimination of critical flow.

The presence of the shroud caused the flow incident upon the cylinder to be highly turbulent (regardless of the Reynolds number) thus eliminating the chance of laminar separation. Thus, drag coefficient tended to become independent of Reynolds number and showed an increased level in regions where flow around the bare model had been critical.

The link between vibration amplitude and drag coefficient is illustrated in Fig. 6.7.12 for the 100% coverage case; three peaks can be seen which correspond to the peak amplitudes in modes one, two and three. The curve also shows that for complete shroud coverage the drag coefficient is Reynolds number independent.

7.3 The Faired Cylinder.

7.3.1 Uniform velocity profiles (Tests 14 to 18).

The effect of regular fairing coverage on Reduced Amplitude for modes one, two and three is shown in Fig. 6.7.13. Increasing the coverage progressively decreased the model's peak Reduced Amplitude and, unlike the shroud, the fairing gave a significant amount of suppression with coverages of less than 50%.

The fairing lowered the Reduced Velocity at which the peak amplitude occurred and also narrowed the Reduced Velocity response range by reducing the upper bound; these trends were evident for both first and second modes (Fig. 6.7.13). A summary of all the

ASR's is given in the table below:-

Table 6.14 ASRs for the fairing in a uniform profile.

Mode	Amount of coverage (%)				
	20	25	33	50	100
1	22	33	54	87	98
2	20	34	-	84	98

The drag coefficient for the faired model (Figs. 6.7.14) was, like the shroud, influenced by two distinct phenomenon :-

- The reduced levels of vibration afforded by the fairing which tended to reduce the drag coefficient.
- The elimination of critical flow from those parts of the cylinder which were covered with the fairing. The reasons for this are discussed below.

Because of the low c/t ratio of the fairing, it tended to adopt an angle of about 15% to the incident flow and the flow around the fairing was, therefore, asymmetric. On one side of the fairing the flow remained attached and, consequently, the flow pattern on that side was Reynolds number independent. Flow separation did occur on the other side, however, and it is likely that the position of the separation point was Reynolds number dependent. Despite this, because of the shallow curvature of the fairing, any movement of the separation point would have had only a small effect on the wake width and, hence, on the drag coefficient.

Thus, the flow pattern around both sides of the fairing was largely Reynolds number independent. It follows that the drag coefficient would also be expected to share this independence as was, indeed, observed (see the 100% coverage curve in Fig. 6.7.14).

7.3.2 Linear shear profiles (Tests 19 to 22).

The effect of fairing coverage on Reduced Amplitude for modes one and two in the LS15 and LS30 profiles is shown in Fig. 6.7.15. The performance of the fairing in the sheared velocity profiles compared favourably with that in the uniform profile; the shear significantly increased the Amplitude Suppression ratio, particularly for the LS30 profile, where a coverage of only 33% suppressed the peak vibration in first mode by 87% and in second mode by 98% (table 6.15 below).

Note that the ASR for all shear flow profiles, linear and stepped, is defined with reference to the bare cylinder response in the particular shear flow under consideration.

Table 6.15 ASRs for the fairing in linear shear profiles.

Flow prof.	Mode	Coverage (%)	
		33	50
LS15	1	52	92
LS15	2	68	89
LS30	1	87	97
LS30	2	98	98

The drag characteristic for the faired model in the linear shear profiles is shown in Fig. 6.7.16. Differences between these drag curves and those for the uniform flow case (Fig. 6.7.6) can largely be explained by the differences in the vibration response.

For the LS30 profile there was a noticeable reduction of the drag coefficient as the Reynolds number was increased (Fig. 6.7.16). This was the result of critical flow around the upper part of the model (those parts not covered by the strake) where the velocity reached values considerably higher than it did in the uniform profile test.

7.3.3 Stepped shear profiles (Tests 23 to 28).

The effect of fairing coverage on Reduced Amplitude for modes one and two in all three stepped shear profiles is shown in Figs. 6.7.17 and 18. In first and second modes the SS55 profile had little effect of the ASR (cf uniform flow values), but as the excitation length was further decreased the ASR's were improved (table 6.16 below).

All three profiles had the effect of reducing the upper bound of the Reduced Velocity Response Range; this is particularly noticeable for first mode (Fig. 6.7.17)

Table 6.16 ASRs for the fairing in stepped shear profiles.

Flow prof.	Mode	Amount of coverage (%)			
		20	25	33	50
SS55	1	-	-	53	93
SS55	2	-	-	65	89
SS40	1	-	36	89	-
SS40	2	-	70	85	-
SS24	1	92	95	-	-
SS24	2	79	85	-	-

The drag characteristics for the stepped shear profiles are shown in Fig. 6.7.19. As was observed for the linear shear profiles, the general form of these characteristics is strongly related to the model's amplitude response.

7.4 The Straked Cylinder.

7.4.1 Uniform velocity profile (29 to 34).

The effect of regular strake coverage on Reduced Amplitude for modes one, two and three is shown in Fig. 6.7.20. As expected, increasing the coverage progressively decreased the model's peak Reduced Amplitude and, although the strake performed better than the shroud, large amounts of coverage were required to give a significant amount of suppression (table 6.17). Because of this no tests were carried out for coverages below 33%.

The Reduced Velocity response range for first mode was adversely affected by the presence of the strake, which tended to increase the upper bound of the range.

Table 6.17 ASRs for the strake in a uniform profile.

Mode	Amount of coverage (%)		
	33	50	100
1	15	41	73
2	32	44	89

The drag coefficient for the straked model (Figs. 6.7.21) was influenced by two distinct phenomenon, much as the shrouded and faired models had been:-

- The reduced levels of vibration afforded by the strake which tended to reduce the drag coefficient.
- The elimination of critical flow around those parts of the model which were covered by the strake.

Critical flow around a cylindrical structure is the result of a rearward movement of the flow separation points as the Reynolds number increases (chapter 2). This phenomenon cannot occur where a strake is fitted, however, because the flow is forced to separate at the sharp helical edge of the strake. The flow separation points are, therefore, fixed making the flow pattern and drag coefficient Reynolds number independent.

The elimination of critical flow led to a slight increase in drag coefficient, compared to the bare cylinder case, at velocities where the flow around the bare model had been

critical.

In the sub-critical regime, at least 50% strake coverage was required to significantly reduce the drag coefficient.

7.4.2 Linear shear profiles (Tests 35 to 38).

The effect of strake coverage on Reduced Amplitude for modes one and two in the LS15 and LS30 profiles is shown in Fig. 6.7.22. The performance of the strake in the sheared velocity profiles compared favourably with that in the uniform profile; there were two main improvements :-

- The Amplitude Suppression ratio was increased, particularly for first mode. Values of ASR are given in table 6.18 below.
- The tendency for the strake to increase the upper bound of the Reduced Velocity response range in the uniform profile was completely eliminated and, in first mode, the upper bound was significantly reduced.

Table 6.18 ASRs for the strake in linear shear profiles.

Flow prof.	Mode	Coverage (%)	
		33	50
LS15	1	29	50
LS15	2	33	44
LS30	1	36	49
LS30	2	(1)	(1)

Note (1) The peak response was probably not reached for these tests.

The main effect of the linear shear on the model's drag coefficient appeared to be related directly to the vibration amplitude of the model (Fig. 6.7.23), as was the case for the other two suppression devices. For both shear profiles, the vibration was dominated by first mode; the peak amplitude was reduced (cf uniform flow case) but the Reduced Velocity response range was broadened. These two characteristics are reflected in the shape and level of the drag coefficient curve; the curve has a single broad peak and the level of the peak is low when compared to the curve for the uniform flow case.

7.4.3 Stepped shear profiles (Tests 39 to 43).

The effect of strake coverage on Reduced Amplitude for modes one and two in all three stepped shear profiles is shown in Figs. 6.7.24 and 25. In first mode, the Amplitude Suppression ratio was improved by the sheared profiles and, with the exception of the SS40 profile, the Reduced Velocity response range was also reduced.

The effect of strake coverage on the second mode response was less clear and difficult to understand. Although the Amplitude Suppression ratio was improved by the SS55 and SS24 profiles, the SS40 profile worsened the Amplitude Suppression ratio, giving virtually no vibration suppression. A summary of the ASR's is given in the table below.

Table 6.19 ASRs for the strake in stepped shear profiles.

Flow prof.	Mode	Coverage (%)	
		33	50
SS55	1	39	45
SS55	2	32	52
SS40	1	29	40
SS40	2	(1)	10
SS24	1	93	98
SS24	2	57	97

(1) The peak amplitude appeared not to have been reached for this test.

There was a noticeable shift in the Reduced Velocity response range for some of the tests, with peak amplitudes occurring at a Reduced Velocity of about 8, rather than 6 for the uniform profile.

The variation of the model's drag coefficient in the stepped shear profiles can be seen in Fig. 6.7.26. When compared to the drag curves for the uniform case (Fig. 6.7.10), most of the differences can be explained by the level of vibration present in the two cases. For example, the drag curves for the SS24% flow profile shows no significant peaks because there was no significant vibration.

7.4.2 Irregular strake coverage (Tests 44 to 48).

Note that, for all the tests described in this section, only the first mode of vibration was measured. Diagrams illustrating the types of irregular strake coverage are shown in Fig. 6.6.1b.

Fig. 6.7.27 shows the effect of central strake coverage on Reduced Amplitude over a range of Reduced Velocity. A high degree of suppression was achieved (table 6.20 below), even with only 25% coverage, and all three results compare very favourably with those where the strake coverage was regular (table 6.17).

Table 6.20 ASRs for the strake with central coverage.

Mode	Amount of coverage (%)			
	25	50	75	100
1	32	60	59	73

The trend towards improved suppression is, perhaps, not surprising, as one would intuitively expect a better performance from a device placed close to the antinode of vibration (i.e. central coverage in the case of first mode vibration).

The results for the 50% and 75% coverages were remarkably similar, indeed, there was no advantage in increasing the amount of coverage beyond 50%. Notice also that both these coverages produced, for the first time during the test programme, significant hysteresis. Possible reasons for this are discussed in chapter 7.

The corresponding drag curves for the central coverage tests are shown in Fig. 6.7.28. Most of the differences between the three curves can be explained by considering two factors:-

- The amplitude of vibration (large amplitudes tending to produce large values of drag coefficient).
- The high drag of the strake itself, even when there is no vibration. (A strake of this type has a drag coefficient of about 1.4 when stationary, which is considerably higher than that of a bare cylinder, particularly if the flow around the cylinder is critical (chapter 3).

The hysteresis which occurred in the vibration of the model for coverages of 50% and 75% clearly had an effect on the hydrodynamic drag, as one might expect. Notice that the drag for the 75% coverage was generally higher than for the 50% case. Both coverages produced similar levels of suppression, but the extra length of strakes present in the 75% case increased the overall drag on the model.

The remaining irregular coverage tests involved a fixed amount of strake coverage (50%) which was placed in three different positions along the length of the riser (tests 45, 47 and 48. See Fig. 6.6.1b for a diagram of the coverage positions).

The amplitude response for these tests is shown in Fig. 6.7.29. Clearly, the position of the coverage had a profound effect on the suppression performance. With the strakes

placed at the ends of the model, i.e. next to the modal nodes, there was virtually no suppression, however, as the strakes were moved closer to the antinode, the ASR increased through 0.38 to 0.60 (table 6.21 below).

Table 6.21 ASRs for the strake with 50% central coverage.

Mode	Position of the 50% coverage		
	Nodes (Test 48)	Quartiles (Test 47)	Antinode (Test 45)
1	3	38	60

(Note that the result for the antinodal coverage has already been presented and discussed above, as part of the central coverage tests.)

Fig. 6.7.30 shows how the drag was affected by the position of the 50% strake coverage. At the higher Reduced Velocities all three curves begin to merge. Here, there was very little vibration for any of the tests and, since the amount of strake coverage is the same for all tests, one would expect the drag coefficient to be the same.

At lower Reduced Velocity's, however, the three curves are markedly different, clearly showing the effects of the differing levels of vibration amplitude.

8 MODEL 2 RESULTS.

8.1 The Bare Cylinder.

8.1.1 Rigid cylinder tests (Tests 1 and 2).

The drag characteristic for the rigid cylinder is shown in Fig. 6.8.1. The curve contains the results of test 1 and 2, showing that the experiment was repeatable. At the lower end of the test velocity the drag coefficient had a value of about 1.3, indicating sub-critical flow (note that these results have not been corrected for flume blockage and are, therefore, approximately 10% above the real value).

As the velocity increased, there was a marked reduction in drag; this is the hall mark of the onset of critical flow. Critical flow became fully established at a velocity of about 1.4 m/s. The remaining tests were, therefore, carried out over the velocity range 1.4 to 2.1 m/s, so as to be totally in the critical regime.

8.1.2 Flexible cylinder tests (Tests 3 and 4).

The results of both tests are shown in Fig. 6.8.2. The most striking feature of this curve is its low peak value which only reached 0.87, significantly less than that which was predicted from the model's Stability Parameter. This over-prediction is believed to be caused mainly by the presence of the critical flow, although the sinusoidal mode shape is also believed to be a factor. (Note that the established relationship between peak Reduced Amplitude and Stability Parameter is strictly valid only for cantilever mode shapes and sub-critical flow.) These points are discussed further in chapter 7.

In the critical flow regime, the fluid separates from points nearer to the cylinder's base and the wake width is significantly reduced. Under these conditions it is reasonable to expect that the vortices which are shed would be of reduced size (compared to their sub-critical counterparts). In addition, because the vortices are shed closer to the base of the cylinder, the fluctuating lift force which is associated with them would be reduced. Both these effects would tend to reduce the amplitude of vortex induced vibration.

At a Reduced Velocity of about 7, the model stopped vibrating in first mode and began to vibrate cross-flow in second mode. This is a remarkable event because the Reduced Velocity at this point, based on the second mode natural frequency of 5.5 Hz, was only 2.2, much lower than the established value for the onset of vortex induced vibration of about 4.5.

An explanation for this is that, because of the narrow wake which accompanies the critical flow, the model was behaving as if it had a smaller diameter. If this were the case the effective Reduced Velocity would be raised, and would fall closer to the established value of about 4.5.

This argument can also be used to explain why the Model's first mode amplitude response peaked over a relatively low band of Reduced Velocity (5.0 to 6.5, rather than the established 6.5 to 8.0 (chapter 2).

Fig. 6.8.3 shows the drag coefficient for the bare cylinder. Clearly, the drag was increased by the vortex induced vibration; the maximum amplification (a factor of about 1.7) occurred at a Reduced Velocity of 5.5, however, the drag coefficient remained very low when compared with typical values for a cylinder undergoing vortex induced vibration in the sub-critical flow regime (C_d 's of over 2.0 are common). This was to be expected, given the reduced size of the wake which occurs when the flow is critical.

8.2 The Straked Cylinder (Tests 5 to 7).

8.2.1 4.5 D pitch.

The effect on amplitude levels of fitting three different amounts of the industrial strake to the centre section of the cylinder can be seen in Fig. 6.8.4. The suppression ratios achieved were very similar to those of the 15 D pitch strake in a sub-critical flow (table 6.20), as shown in the table below:-

Table 6.22 ASRs for the strake with central coverage.

Mode	Amount of coverage (%)		
	25	50	100
1	40	63	72

The drag curves for these tests are shown in Fig. 6.8.5. Notice that, although the effects of the model's vibration on the drag curves are visible, they are overwhelmed by another phenomenon; the elimination of the critical flow regime over regions of the model which were covered by the strake.

Eliminating the critical regime increased the wake width and, with it, the drag. As the amount of strake coverage was increased from 25% to 100%, so a greater proportion of the model became affected in this way and, consequently, the drag became greater.

For the 100% coverage case where there was very little vibration ($V_r = 5.25$) the drag coefficient had a value of 1.5. If allowance is made for flume blockage this value becomes 1.34, which is in good agreement with previously measured values for stationary straked cylinders (chapter 3).

8.2.1 15 D pitch (Tests 8 to 10).

The Reduced Amplitude for these tests can be seen in Fig. 6.8.6. A remarkably high level of suppression was achieved, even with only 25% coverage, indeed, there appeared to be little advantage in increasing the amount of coverage above this level; a summary of the ASRs is given below:-

Table 6.23 ASRs for the strake with central coverage.

Mode	Amount of coverage (%)		
	25	50	100
1	71	87	87

These results are significantly better than those achieved by the 4.5 D pitch strake in critical flow, and by the 15 D strake in sub-critical flow.

This improvement over the sub-critical suppression performance does seem reasonable if one considers the reduction in the effective diameter of the bare cylinder which accompanies critical flow.

Although the bare cylinder experiences a reduction in effective diameter, the section covered by the strake does not, since the strake fixes the separation points. Under these conditions, the relative effect of the strake on the behaviour of the cylinder would, inevitably, increase, leading to an improved ASR.

Not surprisingly, the drag curves for these tests (Fig. 6.8.7) show the same trends as those for the 4.5 D strake, although one may have expected a slight reduction in drag coefficient as a result of the reduced levels of vibration.

CHAPTER 7

DISCUSSION OF RESULTS AND COMPARISON WITH PREDICTIONS

1 INTRODUCTION.

This chapter is organised so as to relate closely with the project objectives which were stated at the beginning of chapter 6.

The discussion that follows concerns the general phenomena and trends that are apparent in the results, rather than those which are specific to one particular test result; the latter phenomenon have, where possible, already been discussed at the place of their presentation in chapter 6.

When designing a marine riser, the offshore engineer is concerned with the maximum levels of vibration amplitude and drag force that the riser is expected to withstand, rather than with particular values corresponding to specific free-stream currents. This conservative approach is necessary because of the unpredictability and variability of the marine environment.

The discussion below, therefore, concentrates on maximum values of vibration amplitude and drag coefficient, and these are the main criterion used to quantify the performance of the suppression devices.

Where appropriate, an assessment of the accuracy with which the mathematical models (the ARM and DFM models, presented in chapter 5) were able to predict the experimental results is also included.

2 COMPARISON OF SUPPRESSION DEVICE PERFORMANCE.

2.1 Amplitude Response.

So as not to encroach on the sections that follow, the discussion here is restricted to 100% device coverage in sub-critical uniform flow. In other words, the discussion concerns only the first mode of vibration of Model 1 when exposed to a uniform flow.

2.1.1 Bare cylinder reference test.

The peak amplitude reached by the bare Model deserves some discussion because it was significantly different from the value predicted by the established empirical method (i.e.

by using the curve presented in Fig. 2.15).

Model 1 had a Stability Parameter of 0.09 and a peak Reduced Amplitude in first mode of 1.53. This value is compared with four different predicted values of the Peak Reduced Amplitude in the following table:-

Table 7.1 Predicted peak Reduced Amplitudes for Model 1, first mode.

Empirical	ARM	Blevins	Wake Osc.
1.9	1.57	1.20	1.58

Of the four different methods (discussed in chapter 5), only the predictions of the Wake Oscillator and ARM models were close to the measured value.

It is important to note that the empirical method, unlike all the other methods included in table 7.1, makes no allowance for mode shape, and this is the reason for its large over-prediction.

2.1.2 Cylinder fitted with suppression devices.

A summary of the ASRs achieved by the three suppression devices is given below:-

Table 7.2 Measured ASRs: Model 1, 100% device coverage.

Device	ASR (%)
Shroud	60
Fairing	98
Strake	73

These values agree quite well with the typical values for similar devices quoted in chapter 3:-

Shroud : 50%.
 Fairing : close to 100%.
 Strake : 74% (for a cylinder with a low Stability Parameter).

As explained in chapter 5 (section 3.5.2), the ASR values in table 7.2 were used to calibrate the ARM model, which was then used to predict the effects of partial suppression device coverage on device performance. These predictions are discussed in section 4, below.

2.2 Drag Force.

2.1.1 Bare cylinder reference test.

The maximum drag coefficient (C_d (max)) for the bare model (Model 1) was estimated using the DFM computer program. Since the DFM model was based on equation 11 (chapter 5), it was necessary to define values of Reduced Amplitude, Reduced Velocity and Strouhal number before a prediction could be made.

The following assumptions were made:-

- The maximum value of C_d occurred when the vibration amplitude was a maximum i.e. $Y/D = 1.53$ (first mode). This coincidence of maxima has been established by other researchers (Griffin, 1985 for example) and was confirmed by the majority of the tests conducted during this thesis.
- The Strouhal number was 0.21, the accepted value for sub-critical flows.
- The Reduced Velocity was 6.5. This is the minimum value at which peak amplitudes are expected and should, therefore, give conservative predictions of C_d (see equation 11, chapter 5).

Note that in this example, and in the other cases which follow, it would have been possible to have based the DFM's prediction of C_d entirely on measured values of the input parameters (i.e. the Reduced Velocity and Reduced Amplitude corresponding to the peak value of the measured C_d). This was tried and was found to give accurate predictions, however, in most practical situations this information is not known and the usefulness and validity of the DFM model was, therefore, judged on its ability to predict C_d under the standard conditions described above. These standard input parameters were used for all the DFM predictions which are described below, unless stated otherwise in the text.

For the Model 1 bare cylinder reference test, the DFM predicted a maximum C_d of 2.48, 8% above the measured value of 2.29.

2.1.2 Cylinder fitted with suppression devices.

Although it was not possible to predict the ASR of the suppression devices for the case of 100% coverage, it was possible to run the DFM program and obtain predictions of the maximum expected values of C_d .

As described in chapter 5, the DFM estimated the drag force using an expression based on the relative velocity between the cylinder and the free stream flow (equation 56, chapter 5). The values of C_{ds} used in equation 56 for each device were as follows:-

Table 7.3 Drag coefficients for the suppression devices when fitted to a stationary cylinder.

Device	C_{ds}
Shroud	0.96
Fairing	0.76
Strake	1.56

In table 7.3, the figures for the shroud and strake were calculated by multiplying the values quoted in chapter 3 for these devices by correction factors to account for flume blockage. The figure for the fairing was calculated from the results of these current tests. This was possible because, for certain test velocities, the cylinder vibration was almost completely eliminated by the fairing.

The DFM program was run to predict values of C_d (max) for the shroud, fairing and strake when fitted to Model 1 under conditions of vortex induced vibration. The predicted values are compared to the measured ones in the following table:-

Table 7.4 Maximum drag coefficients:
Model 1, 100% suppression device coverage.

Device	Predicted C_d	Measured C_d
Shroud	1.01	1.72
Fairing	0.76	0.84
Strake	1.59	1.68

Clearly, the DFM was seriously under-predicting the drag on the Model. The implication of this under-prediction, following the discussion given in section 4.3 of chapter 5, is that there is a significant increase in the strength of the vortex shedding from a vibrating cylinder fitted with suppression devices when compared to an equivalent stationary cylinder. This change in vortex strength means that it is inappropriate to assume that the drag coefficient is merely a function of the relative velocity between the cylinder and the free stream flow, as had been the case.

Since Griffin's wake response parameter is able to account for an increase in vortex strength, it was used in place of equation 56 (chapter 5). Equation 11 (chapter 5) was used to calculate the amplification in the local drag coefficient of the suppressed cylinder. Values of C_{ds} were taken from table 7.3.

As for the bare cylinder (section 2.1.1 above), the value of Reduced Amplitude used as input to the DFM program was the maximum in each case. The Strouhal number and

Reduced Velocity were assumed to be 0.21 and 6.5 respectively, the same values as for the bare cylinder.

The DFM's revised predictions are compared with the measured values in the following table:-

**Table 7.5 Maximum drag coefficients:
Model 1, 100% suppression device coverage.**

Device	Predicted C_d	Measured C_d
Shroud	1.31	1.72
Fairing	0.78	0.84
Strake	1.85	1.68

Values of C_d (max) for the fairing and strake were predicted to an accuracy of about 10%. The value for the shroud was under-predicted by 24%, primarily because the peak in the amplitude response occurred at an unusually low Reduced Velocity ($V_r = 5.5$); lower than the assumed value of 6.5. Had the correct value of Reduced Velocity been used, the predicted value would have been 1.56, accurate to within 10%.

N.B The revised version of the DFM program was used for all subsequent predictions of drag coefficient, as described in the following sections.

3 THE EFFECT OF SUPPRESSION DEVICE COVERAGE ON PERFORMANCE.

3.1 Amplitude Response.

3.1.1 Regular coverage.

In order to summarise the data presented in chapter 6, graphs of ASR versus Amount of suppression device coverage (denoted as 'Coverage') have been plotted. Figs. 7.3.1 and 7.3.2 show the ASRs for mode 1 for all three suppression devices used on Model 1; the ASRs for Mode 2 are shown in Figs. 7.3.3 and 7.3.4.

For each suppression device, the ARM program was calibrated using the 100% coverage result and was then used to predict the effect of device coverage on ASR. These computer predictions are shown as dashed lines in the figures.

In general, the agreement between the ARM's predictions and the experimental results was good. For the shroud and strake, whose maximum ASR was well below 100%, the relationship between ASR and coverage was approximately linear. For the better

performing fairing, however, the relationship became non-linear, particularly as the ASR approached the 100% mark.

In second mode the fairing performed as well as it had done in first mode (compare Fig. 7.3.2 with 7.3.4). Differences in performance between modes were observed for the strake and shroud, however, but the ASR-Coverage relationship remained approximately linear. The reason for these differences is not known, but it is suggested that they may have been caused by one or more of the following:-

- Interactions between the various modes of vibration present (possibly in-line as well as cross-flow vibrations).
- Frequency related effects (e.g. hydrodynamic force on the suppression device may be affected by frequency).

3.1.2 Irregular coverage.

The predicted and measured relationship between ASR and central strake coverage for Model 1 is shown in Fig. 7.3.5. As expected, both intuitively and from the theory presented in chapter 5, changing the coverage from regular to central improved the suppression performance and introduced a non-linearity into the relationship; these effects were modelled reasonably well by the ARM program.

Tests were also carried out on Model 1 to investigate the effect of altering the position of a fixed amount (50%) of strake coverage on ASR. The results of these tests are shown in tabular form, together with the ARM's predictions, in Fig. 7.3.6. As the strake was moved toward the vibration antinode the ASR was increased dramatically from 0.03 through 0.38 to 0.60.

There was good agreement between predicted and measured results for all except the nodal coverage case. The reason for this discrepancy is believed to be mode shape distortion. It is suggested that the nodal regions of the model were suppressed more than the central (antinodal) section of the model which would account for the relatively low value of ASR which was measured for this particular coverage.

A mode shape distortion of this kind would be expected to raise the natural frequency of vibration somewhat since the effective length of the cylinder is reduced. A rise in frequency of about 10% was observed where the coverage was nodal which gave some credence to the theory.

Further tests to investigate the effect of central strake coverage were carried out in critical flow using Model 2; these are discussed in section 6 below.

3.2 Drag Force.

3.2.1 Regular coverage.

In order to summarise the results and to allow a direct comparison with predicted values, graphs of maximum drag coefficient versus amount of suppression device coverage were plotted for each suppression device, and these can be seen in Figs. 7.3.7 and 7.3.8.

Results for the bare cylinder and completely covered cylinder (discussed above) have also been included in these graphs.

The predicted values, shown in these figures as dashed lines, were obtained using the DFM program along with the following (standard) input parameters:-

- Y/D = Maximum measured value for each particular coverage.
- V_r = 6.5
- St = 0.21

In general, there was good agreement between predicted and measured results; in almost all cases the agreement was within 10%.

3.2.2 Irregular coverage.

The predicted and measured values of C_d (max) for cases where the strake was applied irregularly can be seen in Figs. 7.3.9 and 7.3.10. Again, predictions were obtained using the DFM program with standard input parameters (section 4.2.1).

In general, the DFM's predictions were reasonably accurate, although the very large drag coefficient which occurred with 25% central coverage was under-predicted by 18%.

The effect on C_d of altering the position of a fixed amount (50%) of strake coverage is shown in Fig. 7.3.10. As the position of the strake became closer to the modal antinode the effective drag coefficient was significantly reduced as a consequence of the reduction in the level of vibration amplitude. This trend was well modelled by the DFM, although the absolute values of C_d were under-predicted by an average of 11%.

4 THE EFFECTS OF CURRENT SHEAR.

4.1 Amplitude Response.

4.1.1 Bare cylinder.

Initial attempts at predicting the amplitude response of the bare cylinder in shear flows were inaccurate; in stepped shear flows the peak amplitude was significantly under-predicted and in linear shear flows the broadening of the RVRP which was clearly observed in the experimental results was not modelled. An example of this latter point is shown in Fig. 7.4.1 for the LS30 profile; compare the experimental result (curve A) with the initial ARM prediction (curve B).

After examining the possible reasons for these inaccuracies it was concluded that the most likely explanation concerned the way in which hydrodynamic damping was modeled by the ARM program. As discussed in chapter 5, it was assumed that the hydrodynamic damping which occurred outside regions of lock-on could be calculated by taking the cross-flow component of the resultant time varying drag force on the cylinder. The damping term was quadratic (i.e. proportional to the square of the relative velocity between the cylinder and the ambient fluid).

Whilst quadratic damping may be appropriate in cases where the free stream flow velocity is relatively high (relative to the peak velocity of the vibrating structure), it seems that it over-estimates the level of drag where the free stream velocity is relatively low, as was the case for the shear flow experiments carried out in this project. (N.B. free stream flow is used here to mean the local free stream flow at points along the cylinders axis which are experiencing hydrodynamic damping, not excitation.)

King (1974) has demonstrated that vibrations of a cylinder in a quiescent fluid are subject to linear, rather than quadratic damping. Where the free stream flow velocity is small (relative to that of the vibrating cylinder) it is likely that the hydrodynamic damping will approximate to the quiescent case and, therefore, the ARM program was modified to include linear, rather than quadratic damping. As described below, this modification greatly improved the accuracy of the predictions.

The effect of changing from quadratic to linear damping on the ARM's prediction of amplitude response in the LS30 flow profile is illustrated in Fig. 7.4.1. Both the general form of the response and the peak amplitude are modelled much better by the linear damping version of the ARM.

Note that all subsequent ARM predictions, described in following sections, were obtained using the linear damping version.

Table 7.6 below compares the predicted and experimentally observed peak Reduced Amplitudes for all the shear flows studied in this project.

Table 7.6 Reduction in peak amplitude due to shear flow.

Shear profile	Reduction in peak Amplitude (%)			
	Mode 1		Mode 2	
	Predicted	Measured	Predicted	Measured
LS15	4	7	6	48
LS30	30	26	29	-
SS55	15	20	16	45
SS40	23	31	22	40
SS24	62	47	47	66

Considering the complexities involved, the ARMs predictions for the reduction in first mode amplitudes due to shear flow were reasonably good, however, in second mode the predicted reductions were, in all cases, significantly less than those actually measured.

Possible reasons for this discrepancy include the multi-modal nature of the second mode response and critical flow effects, neither of which were accounted for in the ARM model.

4.1.2 Cylinder fitted with suppression devices.

The effect of linear and stepped shear flow on the performance of the fairing in first mode can be seen in Figs. 7.4.2 and 7.4.3. The ARM predicted an increase in suppression performance as a result of both types of shear. Such an effect was observed experimentally for the linear shear tests, but for the stepped shear tests the increase in performance was only observed for some of the tests.

The effect of linear and stepped shear flow on the performance of the strake in first mode can be seen in Figs. 7.4.4 and 7.4.5. The increase in suppression performance predicted by the ARM for the linear shear flows was in agreement with the experimental results.

In the case of stepped shear, the ARM predicted a small increase in performance for the SS55 and SS40 profiles and a large increase for the more severe SS24 profile. These effects were, in general, also observed experimentally.

Very similar results were obtained for second mode vibrations in the shear flows (Figs. 7.4.6 to 7.4.9). The ARM predicted an increase in suppression performance for all the sheared flows. There was good agreement between predictions and experimental results for the linear shear tests. For the stepped shear tests the agreement was not as good,

although the trends were modelled.

4.2 Drag Force.

4.2.1 Bare cylinder.

The DFM's predictions for C_d (max) in sheared flows are compared with the experimentally measured values in table 7.7 below. Values for the Uniform flow profile have also been included so that the effect of the shear can be seen.

Table 7.7 Maximum drag coefficients in sheared flow.

Flow profile	C_d (max)	
	DFM prediction	Measured
Uniform	2.48	2.29
LS15	2.42	2.04
LS30	1.63	1.38

For both shear profiles the DFM over-predicted the drag by about 20%, but the fall in drag coefficient with increasing linear shear was well modelled.

Note that the values of Y/D , V_r , and St used as input to the DFM program were as detailed in section 4.2.1 above. For all shear flows the Reduced Velocity was based on the Reference Velocity for the particular flow profile under consideration (chapter 6, section 3.1).

4.2.2 Cylinder fitted with suppression devices.

Graphs of C_d (max) versus amount of suppression device coverage for the fairing and the shroud in sheared flow profiles are shown in Figs. 7.4.10 to 7.4.13.

Comparing Figs. 7.4.10 and 7.3.8, it can be seen that for low amounts of fairing coverage the linear shear decreased the value of C_d (max), but as the coverage was increased towards 50% this effect diminished.

This behaviour can be explained simply by studying the model's amplitude response. For low values of coverage the shear reduced the vibration amplitude and, with it, the drag coefficient. With 50% coverage, however, the fairing had already suppressed the vibration by about 90% and further reduction due to the shear was of little significance.

In stepped shear there was a similar convergence of results as the coverage reached 50%, however, there was no definite trend toward lower drag coefficients as the shear excitation length was decreased. In fact, for the SS55 and SS40 flow profiles, C_d (max) was increased, whereas for the SS24 profile it was decreased (relative to the uniform flow case, Fig. 7.3.8). The reasons for this are quite straightforward and involve the model's vibration amplitude and frequency, as explained below.

For a given amplitude of vibration, one would expect an increase in C_d as the length of the upper, high velocity, excitation region is decreased because of the decrease in the Reduced Velocity which occurs over the lower region (See equation 11, chapter 5). Henceforth, this effect is referred to as the Reduced Velocity effect.

The SS55 and SS40 flow profiles did not appreciably reduce the level of vibration present with low values of fairing fitted and, therefore, one would expect the Reduced Velocity effect to dominate, resulting in an increase in C_d (max), as was indeed observed.

In the SS24 case, the shear significantly decreased the model's amplitude response Which overrode the Reduced Velocity effect and produced a lower value of C_d (max).

Similar trends were observed for the strake, as can be seen in Figs. 7.4.12 and 7.4.13. Values of C_d (max) were reduced by the linear shear profiles and by the most severe stepped shear profile (SS24). The SS55 and SS40 profiles had little effect on C_d values, however, because the Reduced Velocity effect was cancelled out by a reduction in the level of vibration.

5 THE EFFECT OF REYNOLDS NUMBER.

This section deals with the effect of Reynolds number, or more precisely, the effect of critical flow on the vortex induced vibration of a circular cylinder. All the critical flow tests were carried out on Model 2. All tests on Model 2 were performed with the uniform flow profile and the Model's vibration was restricted to the first normal mode.

5.1 Amplitude Response.

5.1.1 Bare cylinder.

Model 2 had a Stability Parameter of 0.15 and a peak Reduced Amplitude of 0.87, lower than the value predicted by any of the four methods discussed in chapter 5 (table 7.8).

Table 7.8 Predicted Reduced Amplitudes for Model 2, first mode.

Empirical	ARM	Blevins	Wake Osc.
1.8	1.51	1.15	1.26

Judging by the predictions given by these methods for the peak Reduced Amplitude of Model 1 (section 2.1.1), particular weight should be given to the ARM and Wake Oscillator models.

The measured peak Reduced Amplitude for Model 2 was about 40% lower than that predicted by the ARM model and about 30% lower than the Wake Oscillator prediction. As discussed in chapter 6, there are good reasons for believing that the lower than expected measured value of the peak Reduced Amplitude was due to the critical flow.

Further evidence for this theory can be found by examining Model 1's vibration in second mode. The peak Reduced Amplitude was 1.21, significantly less than the first mode value of 1.53, and the velocity corresponding with this peak was high enough to give partial critical flow (i.e the separation points had begun to move rearward, but had not reached the point of maximum rearward travel).

It, therefore, seems likely that the reduction in the peak second mode response was due at least in part to critical flow, although there are other possible reasons which include:-

- Non-linear damping. It is possible that the damping during second mode was higher than during first mode due to differences in frequency and bending strain.
- Multi-modal vibrations. The first mode peak occurred at a point where the model was vibrating almost purely in first mode, however, when the second mode peak was reached the vibration also contained some first and third modal components. This may have affected the second mode vibration, causing a reduction in the peak value.

Little previous work has been carried out in this field, however, Humphries (1987) has reported the results of experiments on a large scale cylinder which was thought to be subjected to critical flow. The cylinder underwent large amplitude vortex induced vibration; a result which appears to contradict those of this study. On closer inspection, however, it became apparent that the peak amplitude of vibration occurred in upper sub-critical flow and not in fully critical flow. The large amplitude vibrations were, therefore, to be expected.

5.1.2 Cylinder fitted with suppression devices.

Fig. 7.5.1 shows the suppression performance of the 4.5D pitch strake which was fitted to Model 2. The ARM under-predicted the ASR, but the non-linear form of the curve was successfully modelled.

Changing the pitch of the strake from 4.5D to 15D improved the suppression performance considerably, increasing the degree of non-linearity in the ASR versus. coverage curve (Fig. 7.5.2). This increase in performance must have been due to a fundamental change in the relationship between the effective lift coefficient and the amplitude of vibration and was, therefore, not predicted by the ARM program.

By making a simple modification to the assumed form of the relationship between C_l and Y/D within the ARM program it was, however, possible to improve the accuracy of the predicted curve quite considerably, as shown in Fig. 7.5.2.

In order to assess the effect of the critical flow on the performance of the strake it is necessary to compare the results of Figs. 7.5.2 and 7.3.5. In both cases the strake was geometrically similar and was fitted centrally to the cylinder. The only major difference between these two tests was the Reynolds number; for the Model 1 test (Fig. 7.3.5) the flow was in the upper sub-critical regime, whereas for Model 2 (Fig. 7.5.2) the flow was fully critical.

Clearly, the presence of the critical flow resulted in a significant increase in the strake's performance; the increase in ASR was, in fact, over 100% for low amounts of coverage, decreasing to about 20% for complete coverage.

The increase in strake performance is believed to be due to the decrease in the wake width which occurs in critical flow behind regions of the bare cylinder. Since this wake narrowing does not occur behind the strake, the strake's effect on the cylinder's vibration is bound to increase relative to the sub-critical case.

One might expect this trend to occur for any suppression device which has a wake width which is independent of Reynolds number.

5.2 Drag Force.

The values of Y/D , V_r , and St which were normally used as input to the DFM program were:-

- Y/D = Maximum measured value for each particular coverage.
- V_r = 6.5
- St = 0.21

Due to the effects of critical flow, however, it was inappropriate to use the above values of Reduced Velocity and Strouhal number as input to the DFM program when making predictions of the drag force in critical flow.

The Strouhal number in critical flow is known to increase up to a limit of about 0.45 (Bearman, 1969), although, due to a breakdown in the strength and regularity of the vortex shedding process, the Strouhal number is ill defined in this region (Fig. 2.6).

Humphries (1987) suggests a value of 0.3 as being appropriate for critical flow, and this was the value used here, but only for regions of the cylinder which were not fitted with the strake.

Since the flow around the strake is known to be Reynolds number independent, it was not appropriate to change the Strouhal number from its assumed value of 0.21 over regions of the cylinder which were fitted with the strake.

As pointed out in chapter 6, there was a tendency for the peak Reduced Amplitude to occur at relatively low Reduced Velocities in critical flow and, in an attempt to be conservative in the prediction of C_d (max), it was necessary to change the Reduced Velocity from 6.5 to 5.0.

To summarise, the input parameters used for the DFM program were:-

- Y/D = Maximum measured value for each particular coverage (as before).
- V_r = 5.0
- St = 0.21 for strake covered regions.
- St = 0.30 for bare cylinder regions.

For the bare cylinder, the DFM's predicted value of C_d (max) was 0.97 which compares favourably with the measured value of 0.86. Indeed, this level of accuracy is surprisingly good when it is considered that the wake response parameter was derived for sub-critical flows.

Graphs showing values of C_d (max) for the cylinder fitted with the strake are presented in Figs. 7.5.3 and 7.5.4. Predictions for the 4.5D strake were conservative, though reasonably accurate; those for the 15D strake were less accurate and non-conservative for partial strake coverage.

6 THE INFLUENCE OF MULTI-MODAL VIBRATION.

6.1 Bare Cylinder

This project has shown that large amplitude vortex induced vibrations of bare cylinders are possible where more than one mode of vibration occurs simultaneously.

One might expect that the presence of multi-modal vibrations would reduce the peak amplitude attained by any one mode in the modal group and some evidence to support this was found. In uniform flow, Model 1's second mode response was significantly reduced when compared to the first mode peak amplitude. As discussed above, some of this reduction is believed to be due to the presence of partial critical flow, but, judging by the shape of the second mode response curve (Y/D versus V_r) it is likely that a significant proportion of the reduction was due to the presence of first mode vibration.

As far as the RVRR was concerned, the presence of multiple modes did not affect the Reduced Velocity at which each mode started to become significant. For Model 1, all three modes of vibration started at a Reduced Velocity of about 5. In contrast, the upper bound of the RVRR for second mode was reduced from the usual value of 10 to about 8, apparently due to the onset of significant third mode vibration.

It is, therefore, suggested that where there is a possibility of two or modes occurring simultaneously, the higher mode will tend to dominate. Clearly, further tests would be required to confirm this observation.

6.2 Cylinder fitted with suppression devices.

The main reason for carrying out tests with multi-modal vibration was to investigate its effect on the performance of the suppression devices. From a practical viewpoint, there was concern that, although they may work effectively at suppressing a single mode of vibration, certain configurations of device might not be so effective in other modes, or might even increase vibration in these modes.

This project has shown that, in general, the suppression devices tested on Model 1 worked well in suppressing all three modes of vibration, although quantitative information was obtained only for modes 1 and 2.

The fairing performed equally well in first and second modes. Differences in suppression performance between modes 1 and 2 were observed for the strake and shroud, but these were not large. The strake performed better in second mode than in first, but this trend was reversed for the shroud.

CHAPTER 8

THE DEVELOPMENT OF NOVEL SUPPRESSION DEVICES

1 INTRODUCTION AND OBJECTIVE.

The economic assessment presented in Chapter 4 demonstrated that, where serious vortex induced vibration problems exist, traditional suppression devices are the most economic solution. Even so, the devices are expensive to use and may double the cost of drilling a well.

Of the many types of suppression device to choose from, only fairings, strakes and shrouds have been adopted by the oil industry. Very little effort has been directed at improving their design for offshore use, and even less at developing new designs to overcome the specific problems found offshore.

It was therefore felt that, in addition to the study of traditional suppression devices, this thesis should also include work to establish the feasibility of new designs.

2 DESIGN METHOD.

Since the pioneering work of Price in 1956 (Chapter 3), many people have tried to solve the problem of vortex induced vibration by improving established designs and by inventing new ones. As a result, there are currently over 20 established types of suppression device, although not all have been developed beyond the experimental stage (chapter 3).

Comparatively little work has been directed toward the particular vortex induced vibration problems faced by offshore engineers, and it was, therefore, felt that work in this area offered considerable potential for fruitful development.

In fact, two approaches were adopted in the hope of developing, or inventing, a new suppression device. The first area to be explored involved the use of buoyancy modules, which are commonly fitted to deep water marine risers as a means of supporting the weight of the riser.

Although tension is applied to the top of the riser by the surface structure (usually a semi-submersible), this is not sufficient on its own to support the riser at water depths in excess of about 300 m and, consequently, buoyancy modules are added as a relatively cheap way of overcoming the problem.

Conventional buoyancy modules are about 4.5 m long and have an annular cross-section; they are usually made in two halves which are clamped around the riser and increase its diameter by a factor of two (Fig. 8.1).

The aim of the development work was to produce a device which would provide riser buoyancy and, at the same time, act as a suppression device for vortex induced vibration. Such a device would effectively eliminate the cost of manufacturing and fitting a suppression device, which could amount to a saving of £2.5 M for a single drilling operation (Chapter 4).

Of course, not all marine risers require buoyancy modules, and so work was also directed at developing a device which would be suitable for use on a bare riser. Emphasis was placed on low cost of manufacture and, more importantly, ease and speed of fitting, as this represents a large part of the total cost of using any suppression device (chapter 4).

Fortunately, both the above approaches produced a new design of suppression device and the development work and preliminary experimental results for each is presented in sections 3 and 4 below.

3 THE BUOYANT SUPPRESSION DEVICE.

3.1 Introduction and Objective.

There is a belief, within the oil industry, that vortex induced vibration can be reduced by distributing conventional buoyancy modules randomly along the length of the marine riser (see, for example, Brooks, 1987). It is thought that by doing this the normal axial correlation of vorticity is disrupted, leading to a reduction in the effective excitation force on the riser.

Lamb and Bushell (1988) investigated these claims by testing a number of different types of buoyancy module coverage; both regular and random coverage were tried. Although vortex induced vibration and hydrodynamic drag were reduced by certain configurations, the level of reduction was disappointing (ASRs of about 20% were typical).

During these tests on conventional buoyancy modules, A. Sinker (a student working with the author during the summer of 1988) discovered that the vortex induced vibration of a 1 m long cantilevered cylinder could be greatly reduced by placing a single half buoyancy module near to the tip of the cantilever with the leading edge at about 45 degrees to the incident flow (Fig. 8.2).

As a result of this discovery, and the disappointing results of the conventional buoyancy module tests mentioned above, Lamb and Bushell decided to explore the possibilities of using various configurations of half modules to produce a device which would combine good suppression performance with buoyancy. The device was to become known as the

Buoyant Suppression Device (BSD).

For the BSD to be of maximum practical benefit its suppression performance would have to be omnidirectional (i.e. its suppression performance would not be sensitive to the direction of the incident current).

Simply placing a series of half modules at a flow angle of 45 degrees (Fig.8.2) was not acceptable, since early tests had shown that at other flow angles the effect on the cylinder's vortex induced vibration could be disastrous.

The problem was tackled by designing two independent sets of experiments. Firstly, subjective tests were carried out using a 1 m long cylinder in BHR Group's 1 m deep water flume. These tests were used to gain a basic understanding of the behaviour of a cylinder with half modules attached, and to identify a small number of potentially successful designs which would go forward to the second stage of testing.

The second stage was an objective set of tests on a large scale model in BHR Group's 6 m Deep Water Flume (DWF) which yielded performance data for each of the selected BSD configurations. Both sets of tests are described below.

3.2 Preliminary Small Scale Tests.

3.2.1 Flow visualisation.

The main objective of the small scale test programme was to establish the effect of flow angle (defined in Fig. 8.2) on the performance of the BSD, and to gain a better understanding of the way in which the BSD achieves its suppression.

The investigation began with a series of flow visualisation tests. A rigid cylinder (1 m long, 0.076 m diameter) was secured in the test section of BHR Group's 1 m deep flume and the flow velocity was set to an arbitrary value of 0.33 m/s ($Re = 2.5 \times 10^4$). Flow visualisation was achieved by allowing a thin stream of dye to flow from the tip of an hyperdermic needle which was placed a few centimetres upstream of the model cylinder. The flow patterns were recorded on video to allow subsequent analysis.

Tests were carried out on the bare cylinder, a single whole buoyancy module attached to the cylinder and a single half buoyancy module attached to the cylinder. For the latter case, the flow angle was varied from 0 to 90 degrees, in steps of 15 degrees.

Analysis of the video recording was used to answer the following questions for each of the above tests:-

- i) Are vortices shed in the near wake ?
- ii) Is the shedding alternate, or symmetric ?
- iii) How strong and stable is the shedding phenomenon ?
- iv) How wide is the wake ?

Questions i, ii and iii relate to the fluctuating forces on the cylinder and, therefore, to the likely amplitude response of the cylinder if it were allowed to vibrate.

The importance of question iv stems from the established link between wake width and hydrodynamic drag (chapter 2).

As expected, the bare and whole module tests exhibited a classical vortex street; strong alternate vortices were shed in the near wake of the cylinder as illustrated in Fig. 8.3.

For the half module tests, the flow pattern varied strongly with flow angle. With a zero degree flow angle, alternate vortex shedding still occurred, but it was weaker and less stable than for the bare cylinder.

Turning the half module to produce a flow angle of 15 degrees had the effect of decoupling the vortex shedding from each side of the cylinder, so that vortices were shed as two independent streets in the near wake of the cylinder.

This pattern of independent vortex shedding also occurred at a flow angle of 30 degrees, although this time the wake was deflected slightly.

At a flow angle of 45 degrees there was a dramatic change in the flow pattern. For the first time, large scale vortex shedding was eliminated; in its place, smaller tip vortices were shed in a free shear layer from the leading (sharp) edge of the half module, whilst on the other side of the module, large stationary vortices were formed which would occasionally move off downstream (Fig. 8.4).

This pattern of flow was repeated for flow angles of 60 and 90 degrees although, as the flow angle increased, the wake became wider due to outward jetting from the sharp leading edges of the half module (Fig. 8.5).

3.2.2 Suppression performance.

Although it is possible to draw conclusions as to which flow angles are good and which are bad from the flow visualisation tests on the stationary cylinder, described above, these conclusions must remain tentative because of the interaction which occurs between flow pattern and cylinder motion when the cylinder is allowed to vibrate.

In order to investigate the suppression performance of the half modules, at various flow angles, a series of tests were performed using BHR Group's 1 m deep flume.

A pin-pin supported model cylinder was designed for these tests; it was 1 m long and had a diameter of 0.014 m. The model is shown positioned in the test section of the 1 m flume in figure 8.6.

The model was segmented to make it extremely flexible and its natural frequencies were controlled by the axial tension applied to its upper end. For these tests the tension was set to 70 N, giving a first mode natural frequency of 3.0 Hz.

A pin-pin support was chosen so as to place the vibration antinodes away from the ends of the model and, thus, away from possible disturbances due to end or surface effects.

Eight half modules were made, sufficient to cover one third of the model's length when placed end to end. Each was made in waterproof expanded foam with the following dimensions (see Fig. 8.2 for definition of terms):-

Module diameter = 2 x cylinder diameter.
Module length = 3 x cylinder diameter.

The test procedure was as follows :-

- i) Set working tension of 70 N.
- ii) Secure 8 half modules to the central third of the cylinder to give the required configuration (see table below).
- iii) Set test velocity.
- iv) Record the cylinder's motion using a video camera.
- v) Repeat steps iii and iv to cover required range of velocity.

A total of six half module configurations were tested in this way, as shown in the following table:-

Table 8.1 Test programme to assess effect of flow angle on vortex induced vibration.

Configuration number	Flow Angle (degrees)
1	Bare cylinder
2	+/- 45
3	+/- 90
4	+/- 135
5	+/- 180
6	+ 45

Note that, in configurations 2 to 5, the +/- notation means that the half modules were arranged in alternate angles of (+) and then (-) the quoted angle. This was done to make the effective flow angle for all the modules in each test the same, whilst eliminating the

asymmetric sideways force which would occur if all the modules had exactly the same flow angle.

For each of the six configurations, tests were carried out with velocities in the range 0.15 m/s to 0.85 m/s (Within this range, the velocity was incremented in steps of about 0.15 m/s). This was equivalent to a Reduced Velocity range of 4 to 20 when based on the cylinder diameter of 0.014 m, and a Reduced Velocity range of 2 to 10 based on the diameter of the half modules.

Thus, the Reduced Velocity range was large enough to include the full excitation range ($4.5 < V_r < 10$) for the bare cylinder and the half buoyancy module.

Clearly, configurations 2 to 5 could not possibly be expected to provide omnidirectional suppression, but they could be used to assess the performance of a particular flow angle and, indeed, this was the object of the test programme.

The object of testing configuration 6 was to assess the importance of the axial correlation of the half modules by a direct comparison with the results of configuration 2.

Analysis of the results was by means of the video recording. The cylinder's motion was played back in slow motion which allowed measurements of the screen image to be made. In this way the peak amplitude of vortex induced vibration was estimated with an accuracy of about $\pm 5\%$.

The results of the analysis are shown in the table below :-

Table 8.2 Results of the flow angle tests.

Config. No.	Flow Angle (degrees)	Y/D		ASR (%)	
		Peak 1	Peak 2	Peak 1	Peak 2
1	Bare cylinder	0.70	-	-	-
2	+/- 45	0.01	0.25	99	64
3	+/- 90	0.07	0.50	90	29
4	+/- 135	0.07	0.50	90	29
5	+/- 180	0.07	0.38	90	46
6	+ 45	0.01	-	99	-

In general, two peaks were observed in the cylinder's first mode response; the first occurred at a Reduced Velocity of about 7 (based on cylinder diameter) and was,

therefore, probably caused by vortex shedding from the bare portion of the cylinder. The second peak occurred at a higher Reduced Velocity and was thought to be caused by shedding from the half modules, rather than from the bare cylinder.

NB The ASR for both peaks has been calculated with reference to the bare cylinder's first peak response.

Table 8.2 shows that all 6 configurations were successful at suppressing the first peak, but that the best performance was clearly obtained by the 45 degree flow angle.

The fact that configuration 6 gave as good a suppression as configuration 2 demonstrates that the half modules do not rely on being staggered along the length of the cylinder for their operation. It appears, therefore, that the suppression is achieved by a weakening of the local vortex shedding strength rather than by disrupting the axial correlation of shedding along the cylinder's length.

In all the tests, the second peak in the first mode response occurred during multi-modal vibration. Again, best results were obtained with the 45 degree flow angle. The response for configuration 6 was complicated at higher velocities by a new mode of vibration which was in a direction parallel to the flat faces of the half modules (i.e. at 45 degrees to the flow). This type of vibration was quite different from that of any of the other tests and was not, therefore, included in table 8.2 above.

3.2.3 Conclusion.

From the results of the flow visualisation and vibration suppression tests described above, the following conclusions have been drawn:-

- i) With a half module fitted to a stationary cylinder, large scale vortex shedding ceases to occur at flow angles of 45 degrees or greater.
- ii) The width of the wake tends to increase with flow angle, particularly for angles greater than 45 degrees.
- iii) All the configurations tested produced a significant degree of suppression, although the best performance was achieved by the +/- 45 degree flow angle.
- iv) Staggering the half modules to disrupt the axial correlation of the vortex shedding is not essential for the successful operation of the device at eliminating classical vortex induced vibration, however, if the modules are not staggered, it is possible that new modes of vibration will be excited.
- v) Conclusions i and iv suggest that the half module achieves its suppression performance by reducing the strength of the local vortex shedding, rather than by disrupting axial correlation.

Note that configurations of the half buoyancy modules are referred to as the Buoyant Suppression Device (BSD) in the remainder of this chapter.

3.3 Large Scale Suppression Performance Tests on the BSD.

3.3.1 Test programme.

Following the relatively informal and subjective small scale tests, described above, it was necessary to test the performance of the BSD more rigorously and under more realistic conditions. To this end, a series of tests was carried out using BHR Group's 6 m Deep Water Flume.

Model 1, the 0.076 m diameter model described in Chapter 6 was used for these tests. Using this model had the added benefit of allowing a direct comparison to be made between the BSD and the traditional suppression devices which had already been tested as part of this thesis.

For a description of the Deep Water Flume, the model and its location arrangement please refer to Chapter 6.

The half modules which made up the BSD were made from sheets of foam using a hot forming process. The modules were attached to the model cylinder by clamping them with two plastic cable ties. Each module had a small indentation on its inner surface which engaged with the head of a hose clip which was clamped around each segment of the model cylinder. In this way, the modules were positively located, both axially and rotationally, with respect to the model cylinder. The components used to make the BSD are shown in Fig. 8.7 and a particular BSD configuration (45° stagger angle) is illustrated in Fig. 8.8.

The instrumentation, data processing and analysis for these tests was identical to that used to test the three traditional suppression devices (Chapter 6), with one exception. The Wilcoxon accelerometers, used to measure the vibration of the model cylinder, were replaced with RDP JTF-S/3633 units which were more accurate (+/- 1% compared with +/- 5%).

In all, 18 tests were carried out, as detailed in table 8.3 below. Each configuration consisted of 24 individual half buoyancy modules which were attached to the model cylinder at particular angles to the flow. The cylinder was completely covered by the buoyancy modules.

Two parameters were required to fully define each configuration of BSD:-

- Top flow angle, the flow angle of the top module.
- Stagger angle. Each module was offset relative to its neighbour by a fixed angle (fixed within each configuration); this was known as the stagger angle.

Variation in flow angle was required in order to investigate the omnidirectionality of the BSD.

Table 8.3 Large scale BSD test programme.

Test No.	Config. Name	Flow profile	Stagger angle (degrees)	Top Flow angle (degrees)
1	15/0	Uniform	15	0
2	90/0	"	90	0
3	90/45	"	90	45
4	45/0	"	45	0
5	45/22	"	45	22.5
6	135/0	"	135	0
7	"	"	135	0
8	"	"	135	0
9	135/35	"	135	35
10	135/70	"	135	70
11	45/0	LS15	45	0
12	135/35	"	135	35
13	45/0	LS30	45	0
14	135/35	"	135	35
15	45/0	SS24	45	0
16	135/35	"	135	35
17	45/0	SS40	45	0
18	135/35	"	135	35

For each test, the flume velocity was varied from 0.5 to 2.0 m/s in steps of 0.15 m/s, and the model's first mode vibration response was measured (The Reduced Velocity range was 4.2 to 17). The total drag force on the model was also recorded to allow the drag coefficient to be calculated.

3.3.2 Results.

The performance of the BSD, like that of the traditional suppression devices (chapter 6), was judged by two criteria, its ASR and maximum drag coefficient. Values of ASR and C_d (max) for all tests are shown in table 8.4, below.

In table 8.4, the ASR is based on the Model 1 bare cylinder results detailed in chapter 6. The drag coefficient is based on the projected area of the individual configuration.

Table 8.4 BSD Test results.

Test No.	Config. Name	Flow profile	ASR (%) (mode 1)	C_d (max)
1	15/0	Uniform	47	1.56
2	45/0	"	63	1.32
3	45/22	"	63	1.29
4	90/0	"	49	1.44
5	90/45	"	85	1.06
6	135/0	"	57	1.32
7	"	"	64	1.30
8	"	"	63	1.24
9	135/35	"	65	1.23
10	135/70	"	68	1.29
11	45/0	LS15	67	1.14
12	135/35	"	61	1.21
13	45/0	LS30	64	1.03
14	135/35	"	65	0.87
15	45/0	SS24	93	1.10
16	135/35	"	90	1.03
17	45/0	SS40	77	1.14
18	135/35	"	69	1.18

(a) UNIFORM FLOW PROFILE.

All the BSD configurations gave ASR's of at least 47% and maximum drag coefficients of below 1.6, however, by far the best performance was obtained by the 90/45 configuration (90 degree stagger angle, 45 degree flow angle) which achieved an ASR of 85% and a maximum drag coefficient of only 1.06.

This configuration was not omnidirectional, however, and was judged to be unsuitable for general offshore use. In contrast, the 45 degree stagger angle design performed well at both flow angles, giving an ASR of 63% and a maximum drag coefficient of about 1.3.

Similar results were obtained for the 135 stagger angle design; the average ASR was 63% and the average value of C_d (max) was 1.28.

(b) SHEARED FLOW PROFILES.

The two best configurations (stagger angles of 45 and 135 degrees) were tested in sheared flow to determine its effect on their performance.

Linear shear had very little effect on the performance of either of the two BSD designs, although the maximum drag coefficient was slightly reduced. The stepped shear profiles increased the ASR of both designs and there was a corresponding reduction in the maximum drag coefficient, as one would expect.

3.3.3 Conclusion.

- i) The BSD can be used as an omnidirectional suppression device, giving an ASR of 63% and a maximum drag coefficient of about 1.3.
- ii) Greater performance is possible, but only at the expense of a reduction in omnidirectionality.
- iii) Linear shear does not greatly affect the performance of the BSD. Stepped shear improves the performance, in terms of both ASR and drag coefficient.

The BSD was patented in 1988 by the author and Mr. A.Sinker.

4 THE FLEXIBLE SHROUD SUPPRESSION DEVICE.

4.1 Introduction.

The flexible shroud suppression device (FSD) was invented by the author in 1987. The original design (FSD Mark 1) consisted of a flexible, perforated plastic sheet which was located around spacers attached to the parent cylinder (Fig. 8.9).

Following early trials, this design was refined to form a flexible axial rod shroud (FSD Mark 2, Fig. 8.10). The Mark 2 shroud consisted of three basic components:-

- Axial rods.
- Circumferential ties.
- Spacers.

These components are arranged so as to form a flexible shroud around the cylinder which, because of its flexibility is able to weathervane, regardless of the direction of the incident current.

The operation of the FSD is similar to that of a conventional rigid shroud (chapter 3) in that it affects all three stages of the vortex shedding process; vortex initiation, vortex growth and vortex shedding itself.

The FSD was designed and developed because it appeared to offer a number of advantages over conventional suppression devices, both in terms of suppression performance and economics, as listed below.

a) SUPPRESSION PERFORMANCE.

The FSD has the potential to outperform conventional shrouds because of its flexible trailing edge. There are two reasons for this:-

- i) The trailing edge of the shroud is further aft, increasing its disruptive effect on vortex growth.
- ii) The trailing edge is flexible, enabling it to move as vortices are shed from the cylinder. This movement absorbs energy from the vortices through hydrodynamic and structural damping, without transmitting it to the cylinder.

b) ECONOMIC EFFECTS.

The FSD is a simple device with several attractive features for use in a wide variety of offshore applications:-

- i) The shroud is a loose, self locating fit around the riser and does not, therefore, require expensive design change to be made before it can be used.
- ii) The FSD is ideally suited to retro-fitting which is sometimes necessary to suppress unforeseen vibrations.
- iii) Because of its flexibility and location mechanism, the FSD can be used on any shape of riser, including risers with choke and kill lines and riser bundles.
- iv) Although the shroud has three basic components, it is possible to manufacture it as a single plastic moulding or extrusion.

4.2 Development and Testing of the FSD.

In order to gain the financial support from industry to fund the FSD's development and testing it was necessary to carry out an informal test programme to prove that the device was capable of effective suppression.

Preliminary tests have been completed using a 0.075 m diameter model in BHR Group's 1 m deep flume. These tests were not rigorous and only set out to measure amplitude suppression; the drag force was not recorded.

The results showed that the FSD was capable of producing ASRs in excess of 90%, considerable better than conventional perforated shrouds and at least as good as the most effective rigid axial rod shrouds (chapter 3).

The tests also showed that, although it was important to have a reasonably sized gap between the shroud and the cylinder (produced by the spacer), other design variables such as rod diameter and rod spacing did not appear to be critical.

As a result of these encouraging preliminary results, two sponsors have agreed to fund a development project. The project is currently in progress and aims to optimise the design of the FSD before it goes into commercial production.

The basic concept of the FSD has been successfully patented by the author and Mr G. Jones, who is now working on the development project.

5 CONCLUSION.

Two novel suppression devices have been invented, developed and patented. The BSD offers, for the first time, a device which is capable of providing riser buoyancy and amplitude and drag suppression, thus effectively eliminating the costs associated with using conventional suppression devices.

The FSD, although only at an early stage of development, has the potential to act as a very effective suppression device. It also offers considerable advantages over conventional devices in economic terms; it should be very versatile and cheap to produce and fit.

CHAPTER 9

CONCLUSIONS AND SUGGESTIONS FOR FURTHER RESEARCH

1 CONCLUSIONS.

The primary objective of this project was to increase the current understanding of the behaviour of fluid dynamic suppression devices and to develop analytical techniques which will enable device performance to be reliably predicted, without the need for expensive and time consuming model and prototype testing (chapter 1).

As can be seen from the conclusions listed below, this primary objective has been achieved.

1.1 Conclusions from the Experimental Work.

The experimental test programme described in chapter 6 was designed to achieve five secondary objectives (chapter 6, section 1). Sections 1.1.1 to 1.1.5 below relate directly to these secondary project objectives.

1.1.1 Suppression Device Performance.

- i) An ideal suppression device for use on a marine riser would have a high ASR and a low drag coefficient. Using these two criteria, the ranking order of the three suppression devices tested was:-

1st:	Fairing	(ASR = 98%, C_d (max) = 0.84)
2nd:	Strake	(ASR = 73%, C_d (max) = 1.68)
3rd:	Shroud	(ASR = 60%, C_d (max) = 1.72)

(The bracketed figures are for 100% device coverage in uniform flow.)

- ii) The wake behind a cylinder fitted with a suppression device does appear to become stronger when the cylinder undergoes vortex induced vibration. The increase in the drag force on the suppression device, which accompanies the vibration, is more than that which would be expected due to the increase in relative velocity between the cylinder and the flow.

1.1.2 Partial Suppression Device Coverage.

- i) For the shroud and strake, whose maximum ASR was well below 100%, the relationship between regular device coverage and ASR was approximately linear. For the better performing fairing, however, the relationship became non-linear as the ASR approached 100%.
- ii) The position of the suppression device relative to the cylinder's mode shape is very important in determining the device performance. Placing the device close to the modal antinode gives best results. For example, the effect of moving 50% strake coverage from the modal node to the antinode increased the ASR from virtually zero to 60%.
- iii) The maximum drag coefficient is strongly related to the maximum Reduced Amplitude. Consequently, the effect of device coverage on C_d (max) is very similar to its effect on amplitude response, as described in i) and ii) above.

1.1.3 Sheared Current Profiles.

- i) The introduction of shear in the current profile reduces the peak amplitude of vortex induced vibration of a bare cylinder, but tends to broaden the RVRR. This is true of both linear and stepped shear profiles.
- ii) Linear shear reduces the maximum drag coefficient of a bare cylinder and a cylinder fitted with suppression devices. For stepped shear flows, the effect on drag depends on the ratio (excitation length / total length).
- iii) Hydrodynamic damping, which occurs over regions of the cylinder not subjected to vortex induced vibration, appears to be linear, rather than quadratic, at least in the type of sheared profiles used in this study. This is true for damping of both bare cylinders and those fitted with a suppression device.
- iv) Linear and stepped shear profiles increase the amplitude suppression performance of suppression devices. The degree of improvement depends upon the type of device and the nature of the sheared profile.

N.B. There is a double benefit in terms of reduced levels of vortex induced vibration when a suppression device is used and the flow profile is sheared. Firstly, there is a reduction in the level of vibration of the bare cylinder due to the shear and, secondly, an improvement in the ASR of the device.

1.1.4 Reynolds Number.

- i) Vortex induced vibration does occur in critical flow, but at reduced levels. For a bare cylinder, the peak amplitude appears to be reduced by about 30% to 40% as a result of fully critical flow.
- ii) Critical flow improves the amplitude suppression performance of a strake by between 20% and 100%, depending on the amount of coverage. It is likely that the ASR of any device whose wake is largely independent of Reynolds number would be improved by the presence of critical flow.
- iii) The drag coefficient of a cylinder in critical flow is amplified by vortex induced vibration. The degree of amplification can be predicted by Griffin's wake response method, provided suitable values of the Strouhal number and Reduced Velocity are used. This conclusion applies to a bare cylinder and a cylinder fitted with a suppression device.

1.1.5 Multi-Modal Vibrations.

- i) A bare cylinder can undergo large amplitude vortex induced vibration in more than one mode simultaneously, however, the presence of several modes appears to reduce the level of vibration in any one mode.
- ii) Where multi-modal vibrations occur the lower bound of the RVRR is unaffected ($V_r = 4.5$), but the upper bound can be reduced due to the dominance of a higher mode.
- iii) The fairing's performance was not affected by mode; it performed equally well in modes 1 and 2. Differences in the suppression performance between modes were observed, however, for the strake and shroud, but these were not large.
- iv) Multi-modal vibrations are more likely to occur in sheared flows due to the increase in the RVRR which is caused by the shear.

1.2 Prediction Methods.

- i) A computer based mathematical model has been developed which is capable of predicting the amplitude response of a slender cylinder which is undergoing vortex induced vibration. The following conditions have been successfully modelled:-
 - Bare cylinder.
 - The effects of mode shape.

- The effects of Stability Parameter.
- Cylinder fitted with suppression devices. The effects of full and partial coverage with shrouds, fairings and strakes can be predicted, as can the effect of the position of the devices along the cylinder's axis.
- The effects of sheared free stream flow profiles. Linear and stepped shear profiles can be modelled.

The program, known as the ARM, was designed principally to predict the maximum level of vortex induced vibration that will occur over a range of Reduced Velocity, but it is also capable of modelling the variation of vibration level which accompanies a change in the Reduced Velocity.

- ii) A second computer based mathematical model has been developed, based on Griffin's wake response parameter method, which can be used to estimate the maximum drag force which will act on the cylinder under all the conditions in i) above.

Both programs have been validated against the experimental data derived from this project and, to a lesser extent, against data from other research work.

1.3 Novel Designs.

In addition to the study of traditional suppression devices, this project included an investigation into the feasibility of developing novel designs specifically for use on marine risers (Chapter 8).

Two new types of suppression device have been developed and patented which offer significant advantages over traditional devices.

The BSD enables vortex induced vibration to be effectively suppressed whilst, at the same time, providing riser buoyancy which is essential for many deep water marine risers. Thus, in applications requiring buoyancy the costs associated with the use of a conventional suppression device are virtually eliminated.

The FSD is designed to compete with traditional devices such as the strake and shroud. It offers superior amplitude and drag suppression performance and is quicker and cheaper to make and fit. The FSD is extremely versatile and can be used on a variety of riser shapes and sizes without major design changes.

1.4 Economics.

- i) Vortex induced vibration can cause operational problems which significantly increase the risks involved in offshore oil field exploitation. The increased costs associated with vortex induced vibration are usually dominated by the opportunity cost of delayed oil production.
- ii) Fitting suppression devices, although expensive, is by far the most economic solution to serious vortex induced vibration problems in marine risers.
- iii) This project has shown that satisfactory performance can be obtained from using only partial suppression device coverage. In the case of fairings, this information could reduce the cost of using the suppression device on a typical drilling riser by about 40% (1 million pounds).
- iv) In certain applications, the two new suppression devices developed as part of this project, could be used in place of traditional devices, resulting in a very substantial cost saving.

2 SUGGESTIONS FOR FURTHER RESEARCH.

2.1 Prediction Methods.

The ARM and DFM mathematical models developed as part of this project will, it is hoped, be of use to marine engineers, particularly in the field of riser fatigue analysis. In some offshore situations the models should be of direct use, however, further development would make them applicable to a wider variety of marine risers.

At present, both models treat a single mode of vibration at a time, with the implicit assumption that there are no inter-modal effects. For very deep water risers, where the response is thought to be highly multi-modal, this assumption is likely to lead to erroneous predictions. It may be possible to develop the ARM and DFM models to enable these conditions to be modelled, or it may be more appropriate to use a different type of mathematical model, such as that proposed by Vandiver and Chung (1988, see chapter 5). Further work is required in this area.

In recent years, oil companies have begun to monitor marine risers using specially developed instrumentation packages. It is now possible to measure full scale riser behaviour and ocean currents over a range of water depths. In order to give increased confidence in the predictions of the ARM and DFM models, it is recommended that their predictions be compared with full scale measurements.

2.2 Riser Geometry.

In order to study some of the fundamental phenomena associated with the vortex induced vibration of slender structures in ocean environments, this project has made use of model tests involving two relatively simple circular cylindrical structures.

Some marine risers, such as conductor pipes and deep water risers fitted with buoyancy modules, do have a simple circular cross-section, however, others have more complicated shapes. Deep water risers are commonly fitted with external choke and kill lines which are bound to alter the flow pattern in the vicinity of the main riser. Their effect on vortex induced vibration is unknown and should be investigated.

2.3 Novel Suppression Devices.

The BSD and FSD suppression devices have both demonstrated their potential for use in full scale offshore applications. It is recommended that their development be continued to enable this potential to be realised. By the time the current FSD test programme is completed at BHR Group LTD, both suppression devices will have reached the stage where prototype designs are required, followed by full scale testing and commissioning.

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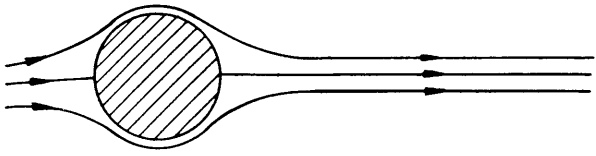
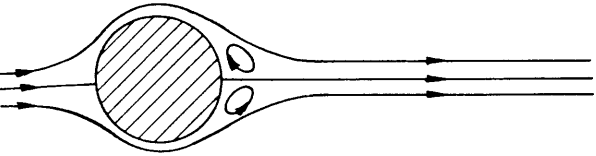
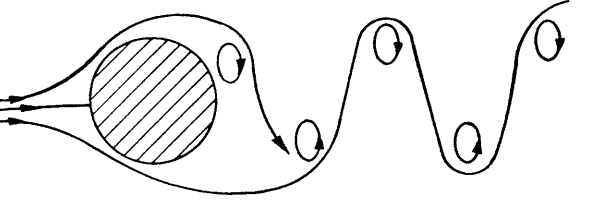
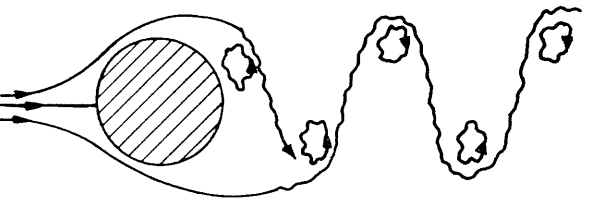
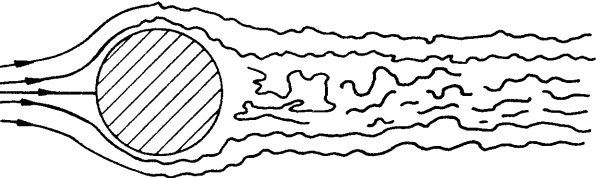
No.	Reynolds number	Flow regime
1	$Re < 5$ Unseparated flow with no vorticity	
2	$5 < Re < 40$ Föppl vortices form at rear of cylinder but no vortices are shed	
3	$40 < Re < 150$ Laminar vortex shedding	
4	$150 < Re < 3 \times 10^5$ Turbulent vortex street. Turbulent transition point moves progressively upstream as Re increases	
5	$3 \times 10^5 < Re < 3.5 \times 10^6$ Cylinder boundary layer becomes turbulent. Vortex shedding ceases. The wake narrows and is very disorganised.	
6	$Re > 3.5 \times 10^6$ Vortex shedding returns. Both the vortex street and the cylinder boundary layer are fully turbulent. The wake is narrow.	

Fig. 2.1 The variation of flow regime around a circular cylinder with Reynolds Number (Lienhard, 1966).

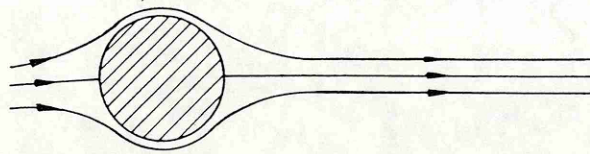
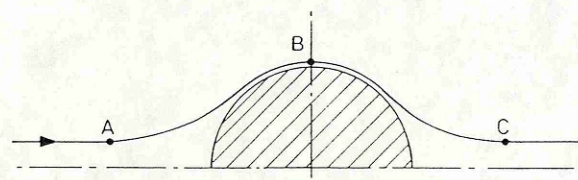
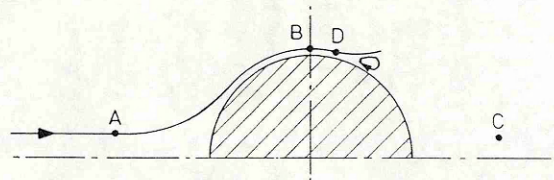


Fig. 2.2 Potential flow around a circular cylinder.



(a) Potential flow



(b) Real flow

Fig. 2.3 Boundary layer separation in an adverse pressure gradient.

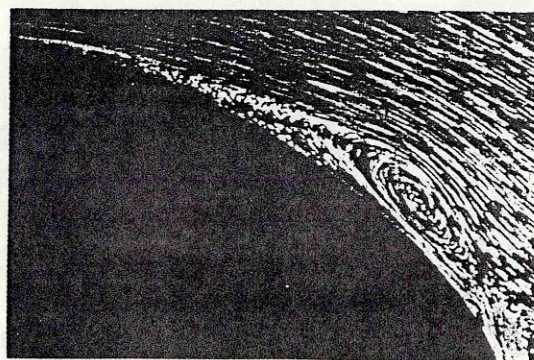


Fig. 2.4 Vortex initiation downstream of the flow separation point on a circular cylinder (Prandtl & Tietjens, 1957).

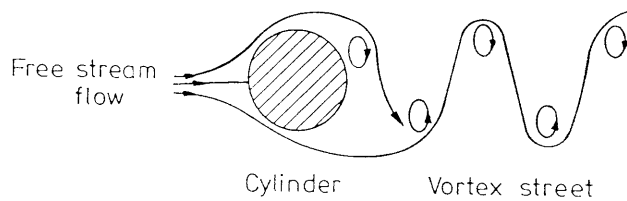


Fig. 2.5 Vortex shedding from a circular cylinder.

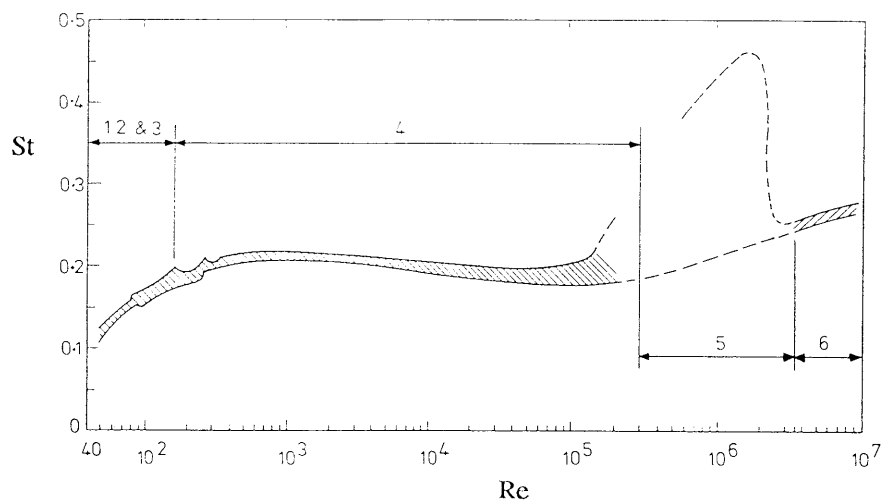


Fig. 2.6 The effect of Reynolds number on the Strouhal number of a circular cylinder (Lienhard, 1966).

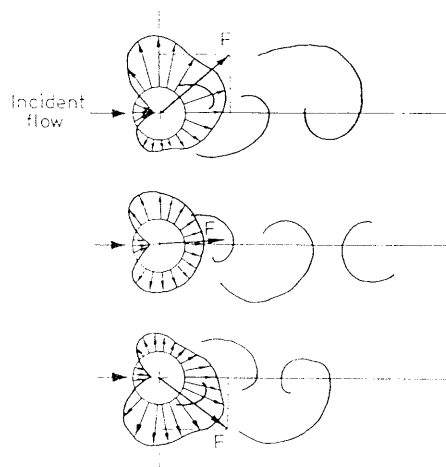


Fig. 2.7 The pressure distribution around a circular cylinder during vortex shedding (Blevins, 1977).

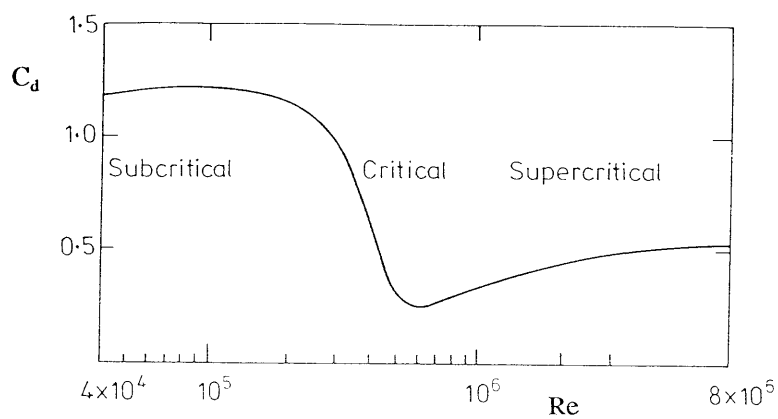


Fig. 2.8 The variation of mean drag coefficient with Reynolds number (Wootton et al, 1972).

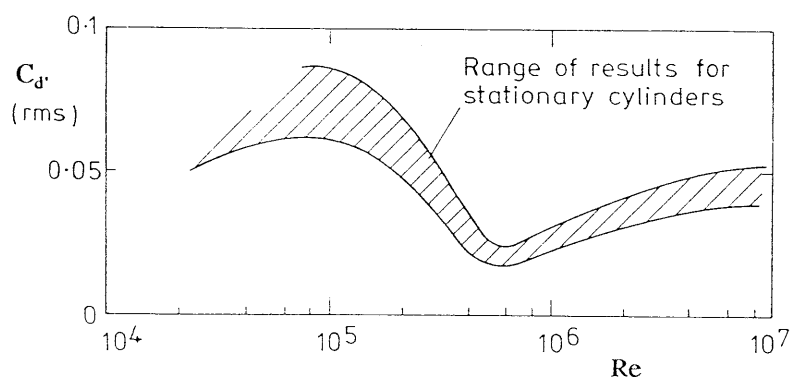


Fig. 2.9 The variation of fluctuating drag coefficient with Reynolds number (Wootton et al, 1972).

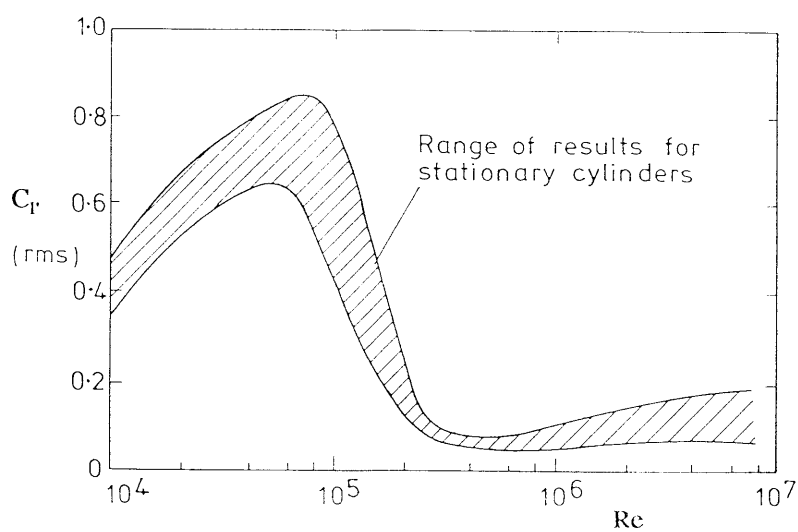


Fig. 2.10 The variation of fluctuating lift coefficient with Reynolds number (Wootton et al, 1972).

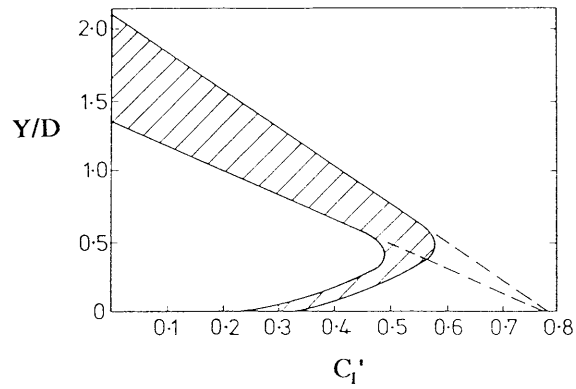


Fig. 2.11 Calculated values of fluctuating lift coefficient versus Reduced Amplitude (King, 1974).

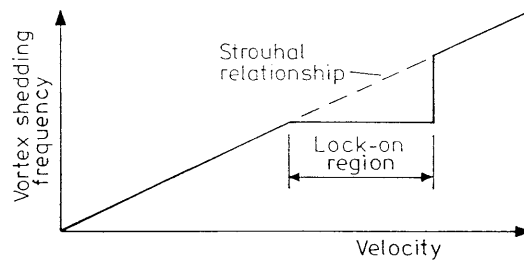


Fig. 2.12 The locking-on of the vortex shedding frequency to the structural natural frequency.

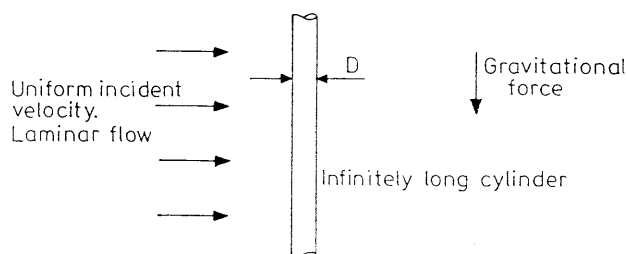


Fig. 2.13 Circular cylinder in a flowing fluid - an idealised case.

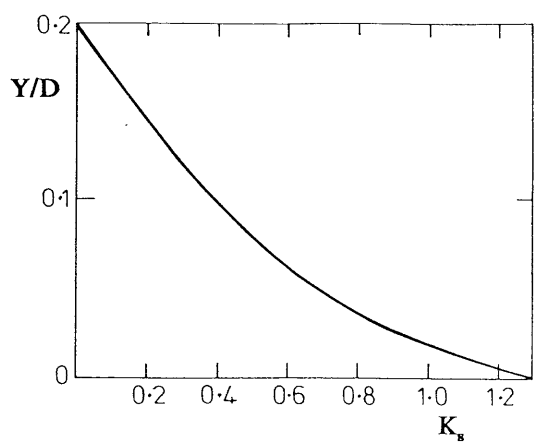


Fig. 2.14 Reduced Amplitude versus Stability Parameter for pure in-line vibration (King 1974).

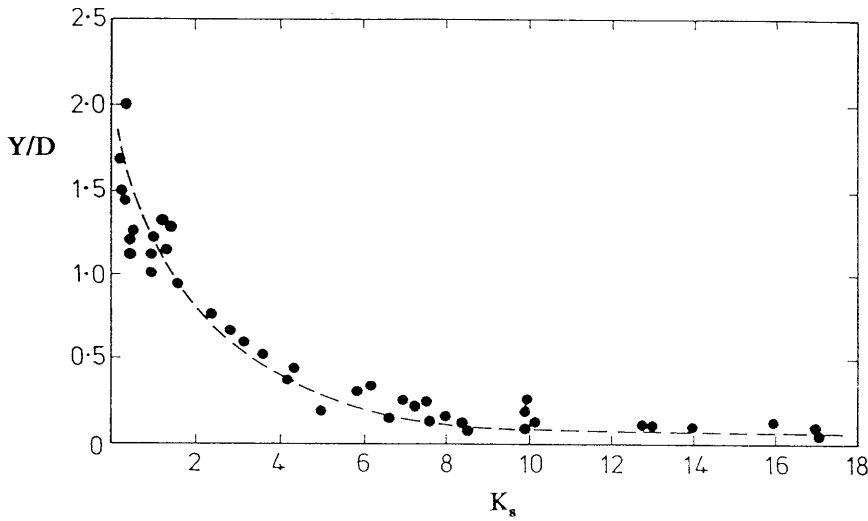


Fig. 2.15 Reduced Amplitude versus Stability Parameter for cross-flow vibration (King 1974).

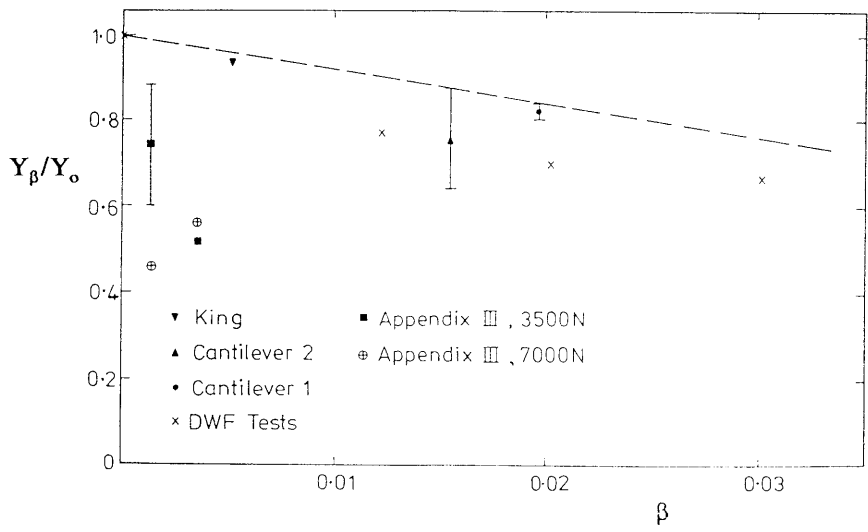


Fig. 2.16 Normalised shear amplitude versus shear parameter (Humphries 1987).

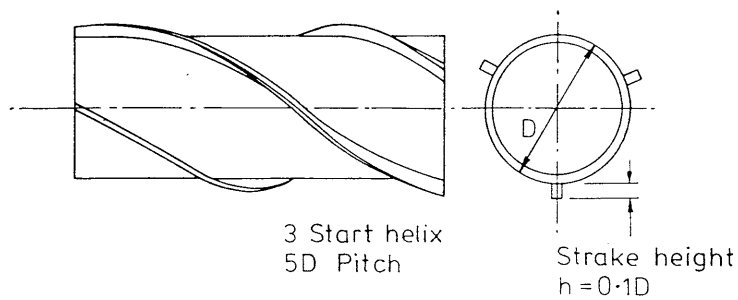


Fig. 3T.1 Rectangular strake with optimum dimensions.

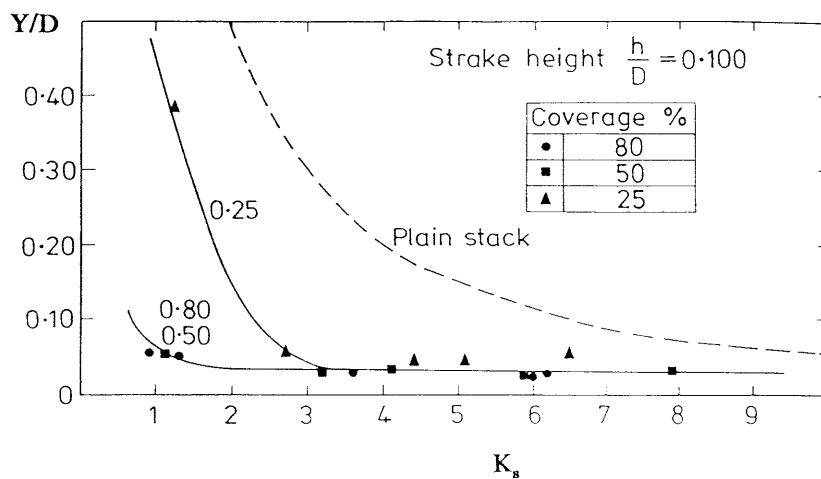


Fig. 3T.2 The effect of rectangular strake coverage on Reduced Amplitude (Vickery & Watkins, 1964).

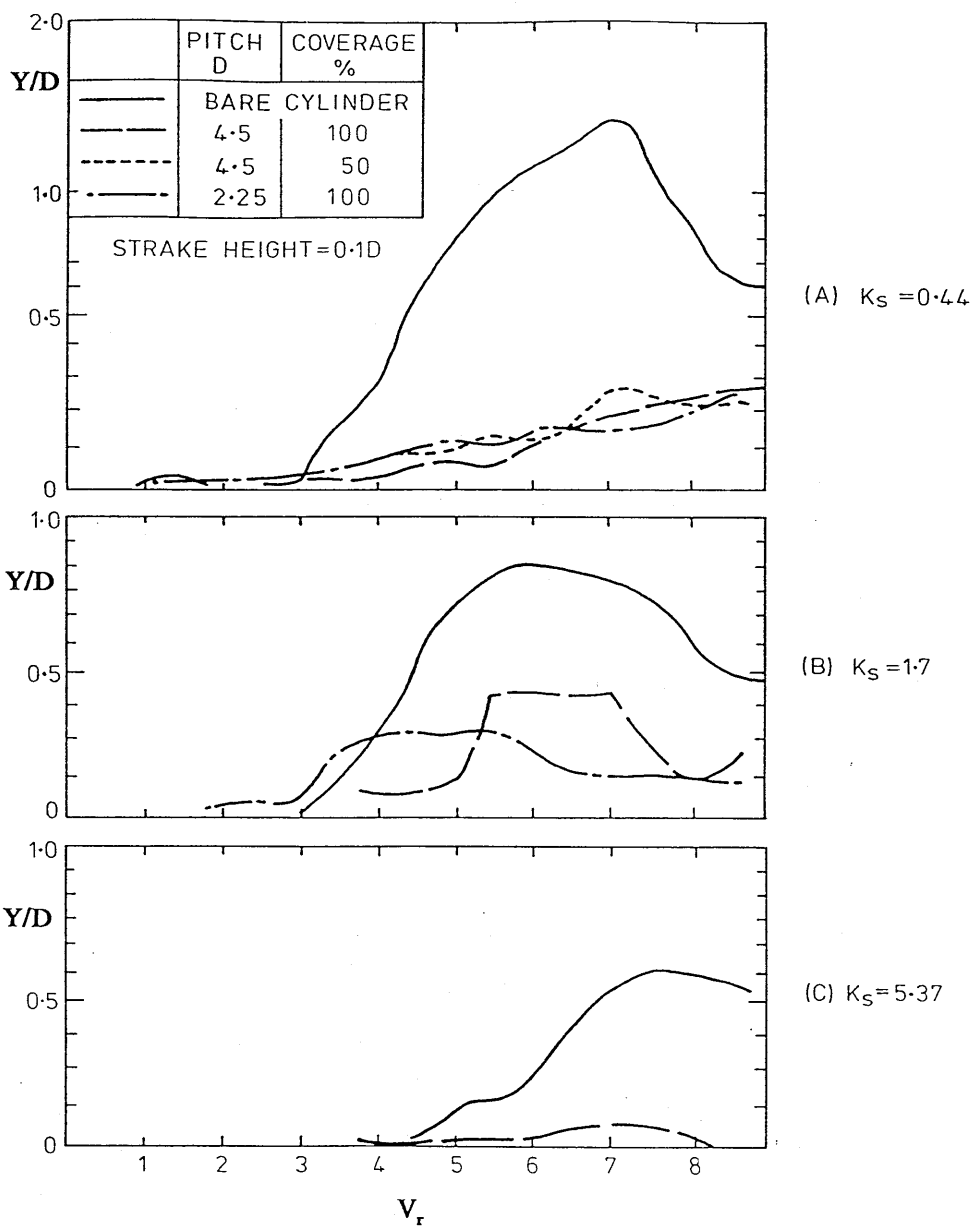


Fig. 3T.3 The effect of rectangular strakes on Reduced Amplitude (Every & King, 1979).

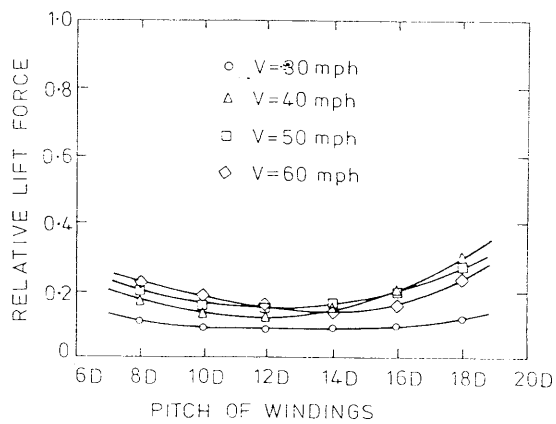
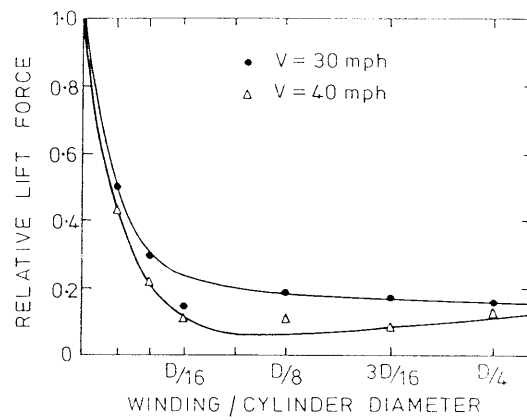
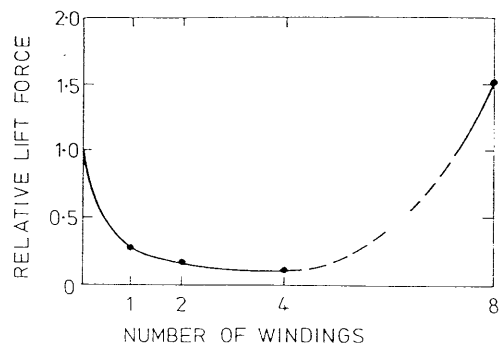


Fig. 3T.4

The effect of various round stakes on lift force (Weaver, 1961).

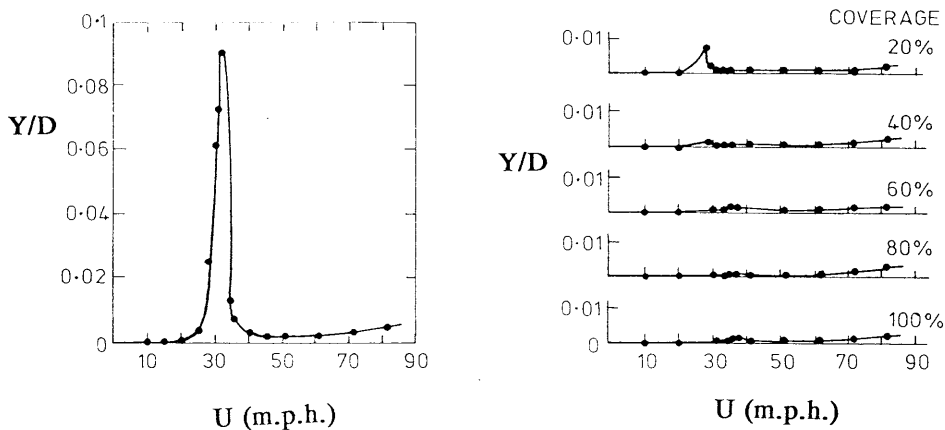


Fig. 3T.5 The effect of round stake coverage (optimum design) on Reduced Amplitude (Weaver, 1961).

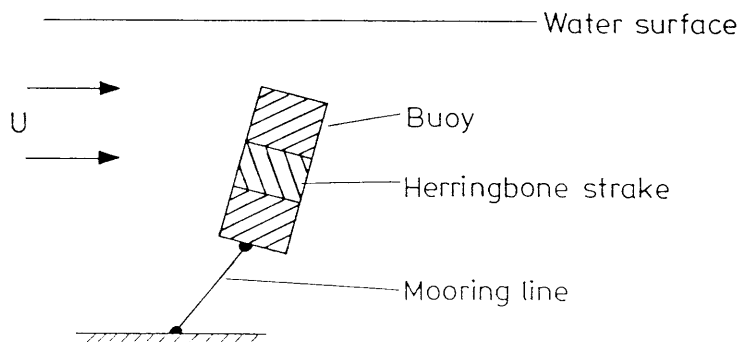


Fig. 3T.6 Herring-bone strakes fitted to a moored buoy (Sallet & Berezow, 1972).

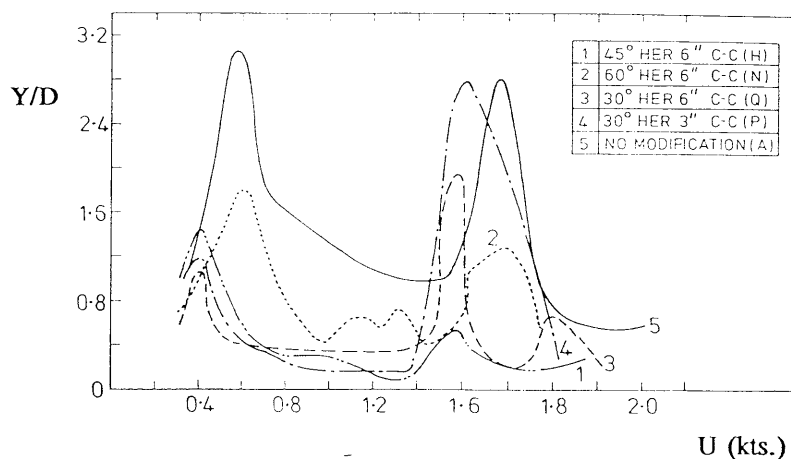


Fig. 3T.7 The effect of various herring-bone strakes on Reduced Amplitude (Sallet & Berezow, 1972).

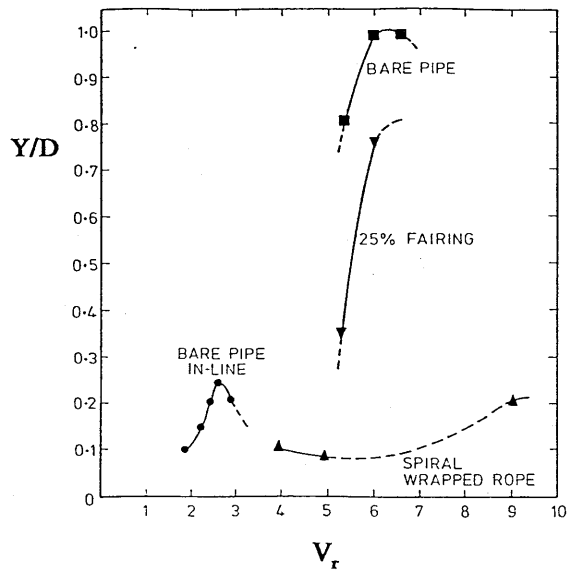


Fig. 3T.8

The effect of a round strake (and fairing) on Reduced Amplitude (Halkyard, 1987).

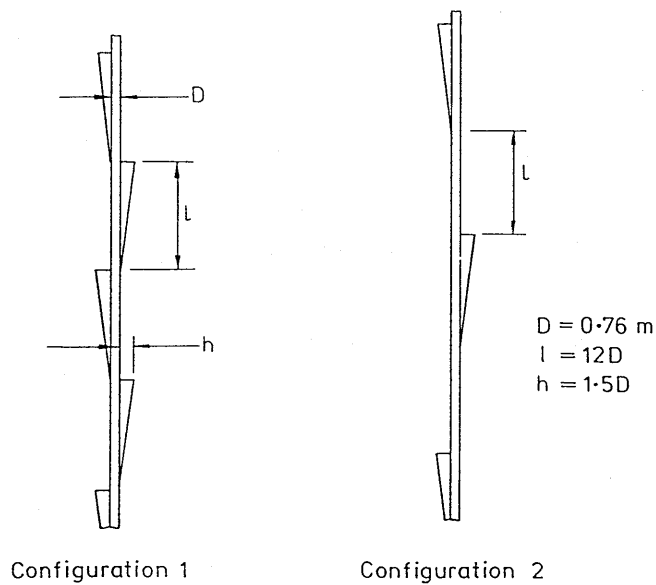


Fig. 3T.9

Baird's sawtooth fin (1955).

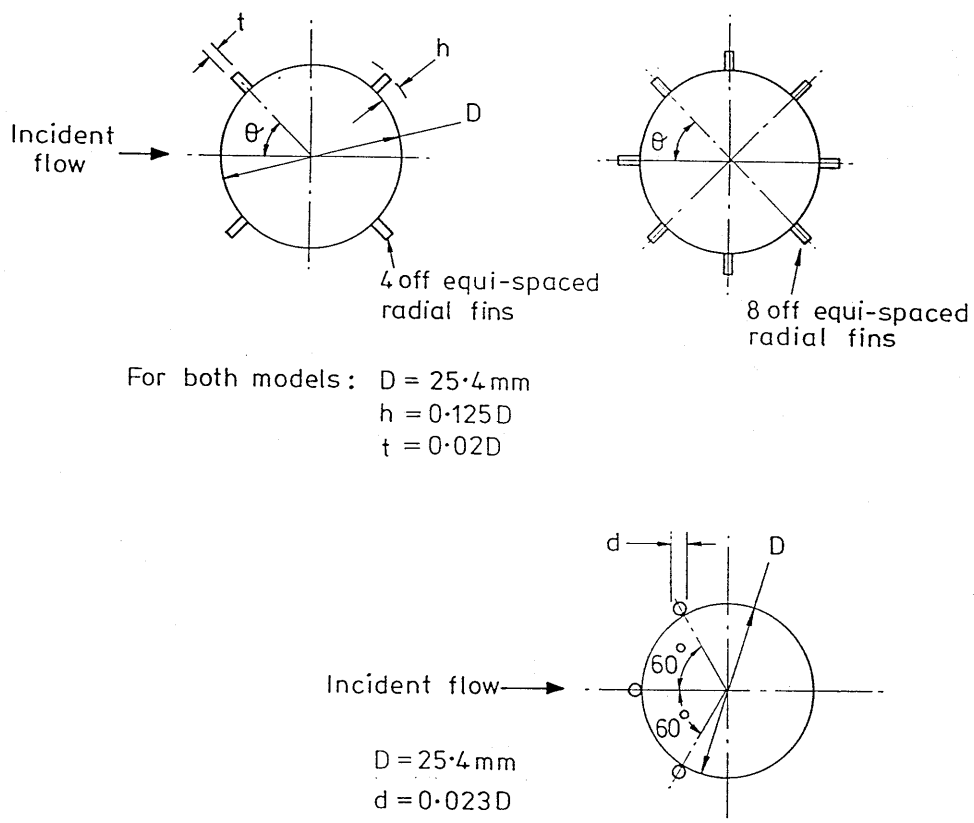


Fig. 3T.10

Price's rectangular and round sectioned fins (1956).

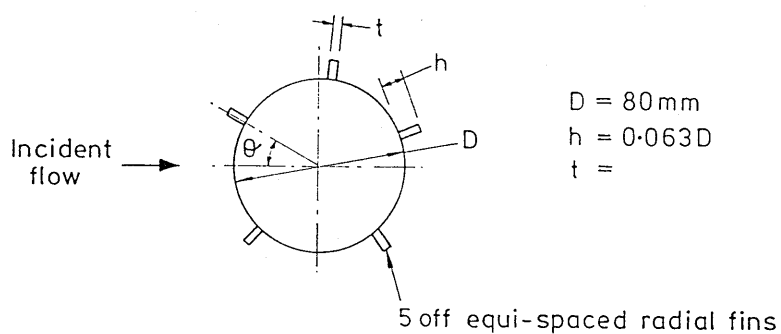


Fig. 3T.11

Mahrenholtz & Bardowick's radial fins (1978).

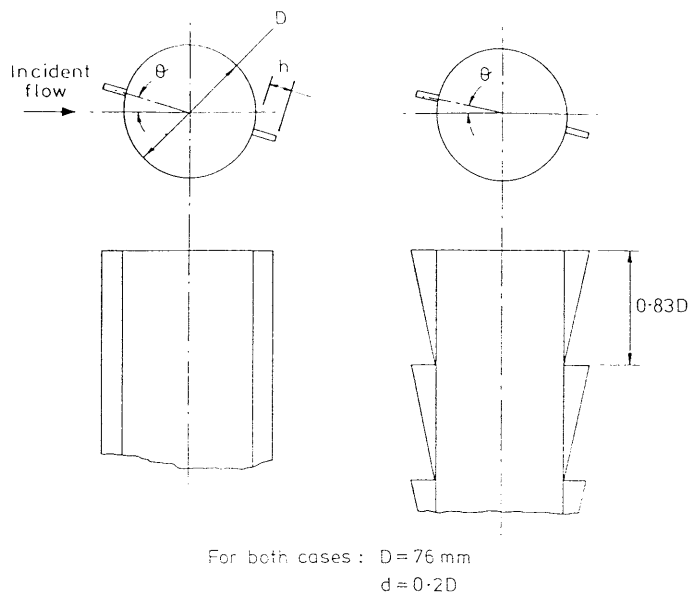


Fig. 3T.12 Gartshore's ordinary and sawtoothed fins (1978).

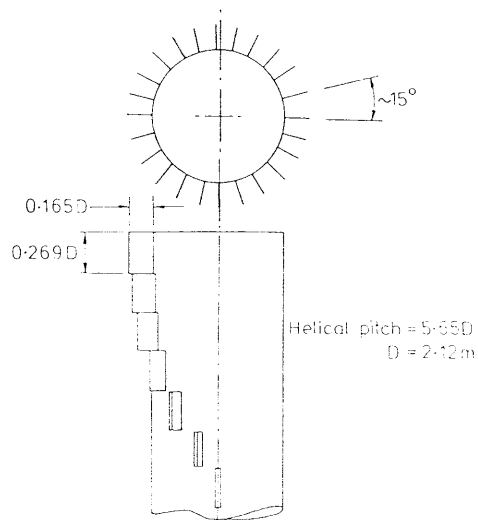


Fig. 3T.13 Novak's staggered radial fins (1966).

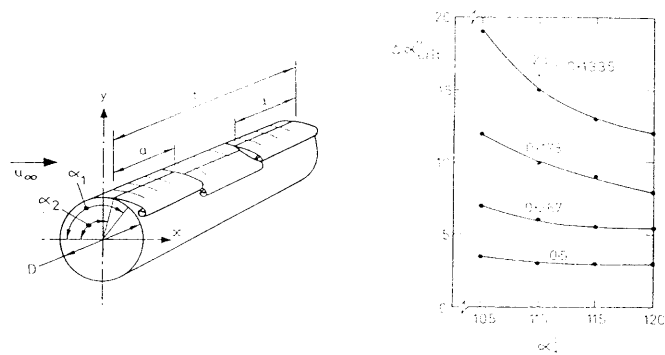


Fig. 3T.14 Naumman's staggered boundary layer tripping wires (1966).

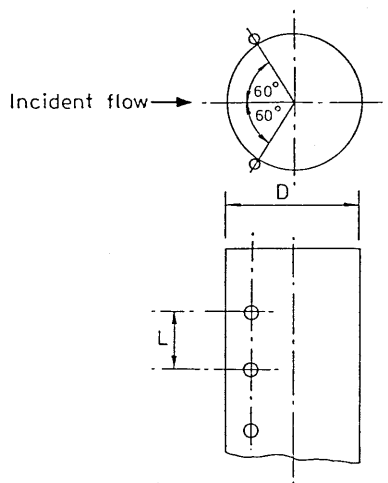
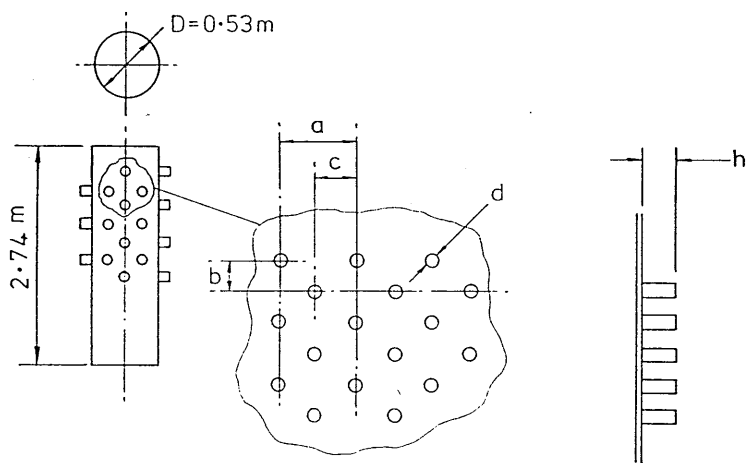


Fig. 3T.15 Mizuno's spheres (1970).



CONFIG. No.	a	b	h	d	c
1	152	76	12.7	12.7	a/2
2	152	76	12.7	6.4	a/2
3	152	76	50.8	12.7	a/2
4	152	76	50.8	6.4	a/2
5	76	76	12.7	12.7	a/2
6	76	76	12.7	6.4	a/2
7	140	152	152	25.4	0
8	140	76	152	25.4	0

Fig. 3T.16 Sallet & Berezow's cylindrical studs (1972).

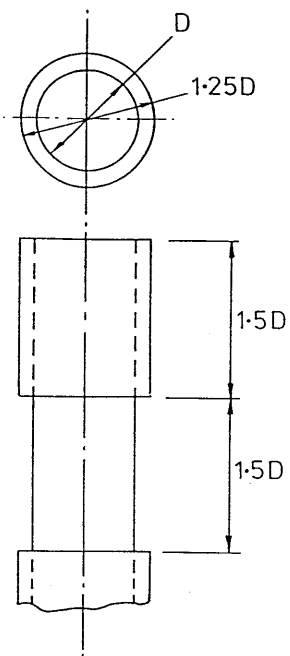


Fig. 3T.17 Price's bushings (1956).

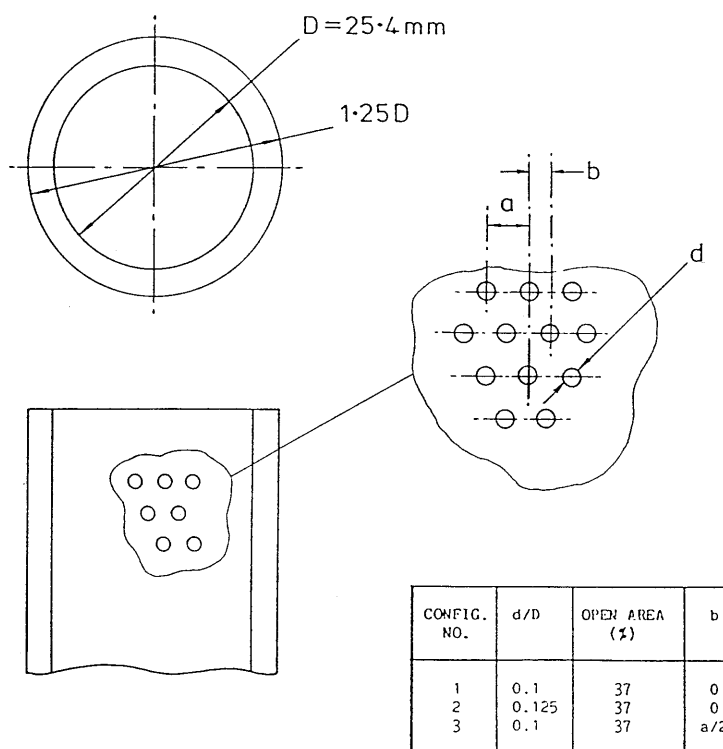


Fig. 3S.1 Price's round hole perforated shroud (1956).

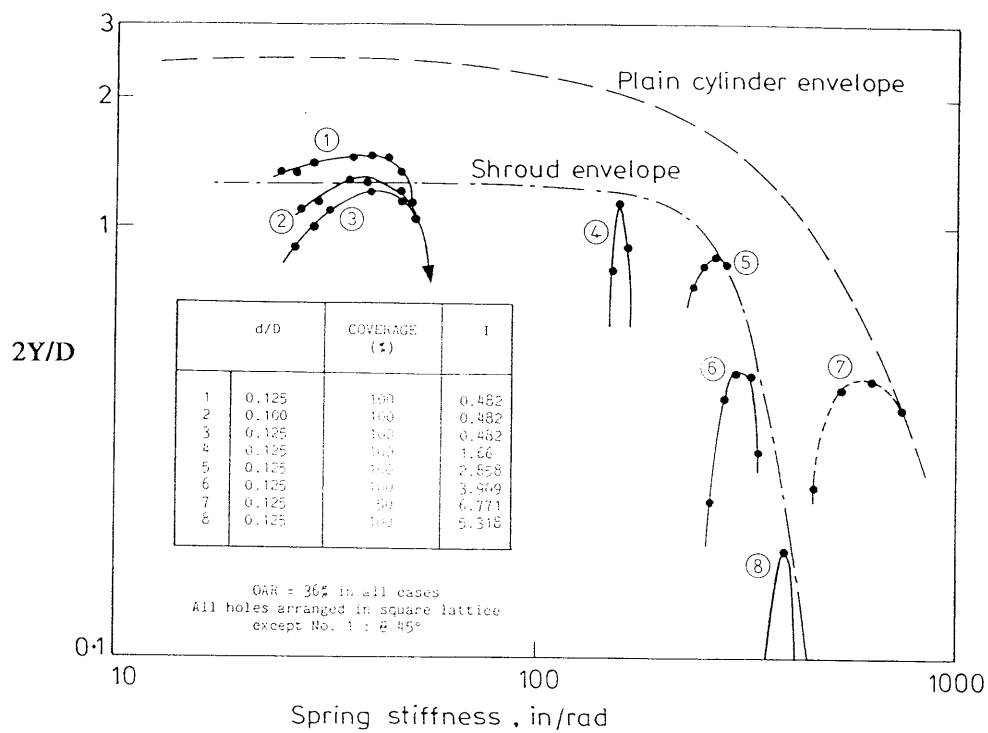


Fig. 3S.2 The effect of various round hole perforated shrouds on Reduced Amplitude (Price, 1956).

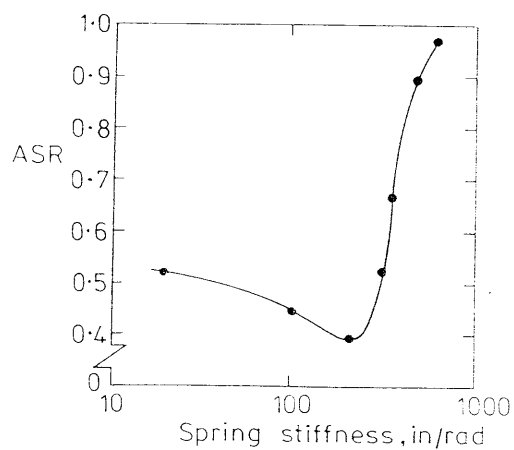


Fig. 3S.3 The variation of Suppression Ratio with structural stiffness for a shrouded cylinder (Price, 1956).

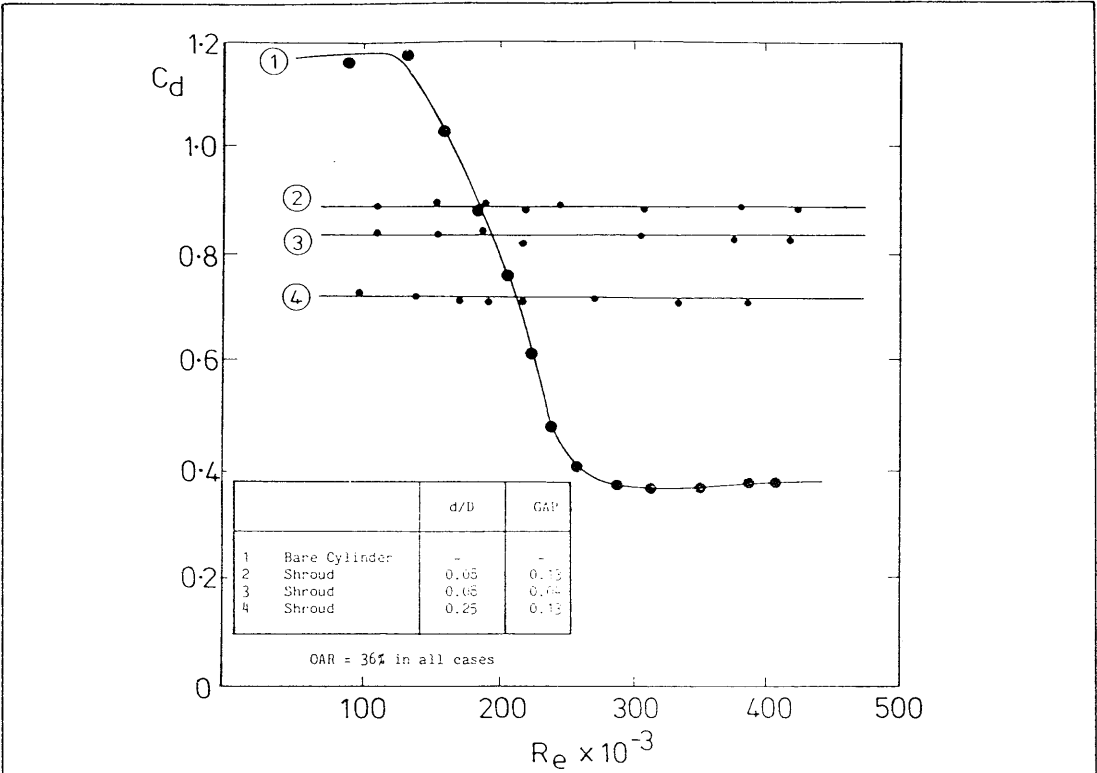


Fig. 3S.4 The drag coefficient of a shrouded cylinder (Price, 1956).

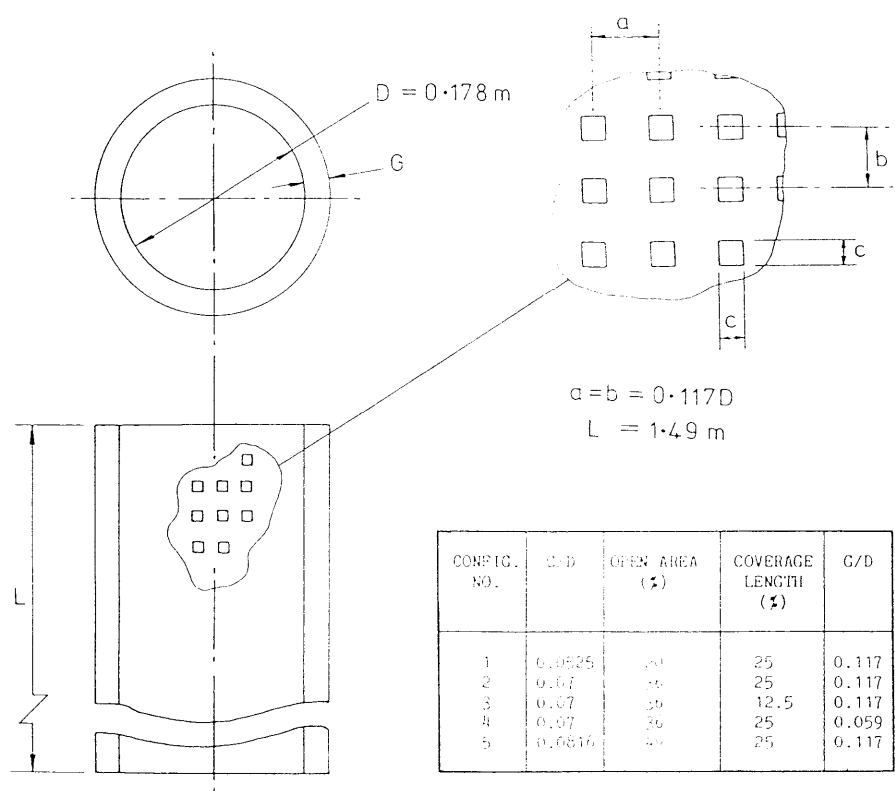


Fig. 3S.5 Walshe's square hole perforated shroud (1968).

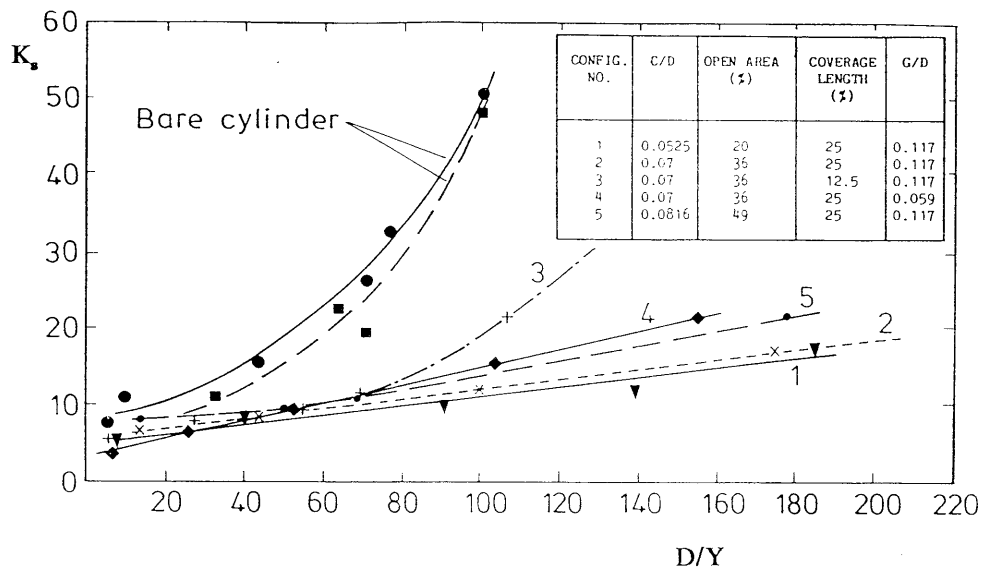


Fig. 3S.6 The effect of various square hole perforated shrouds on vibration amplitude (Walshe, 1968).

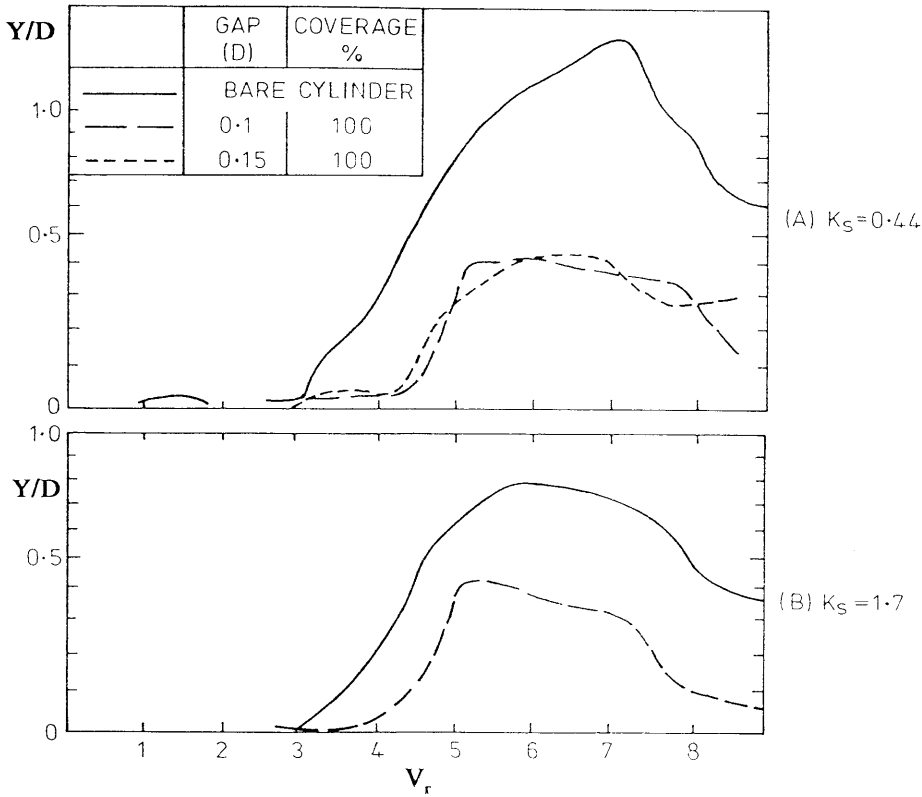


Fig. 3S.7 The effect of square hole perforated shrouds on Reduced Amplitude (Every & King, 1979).

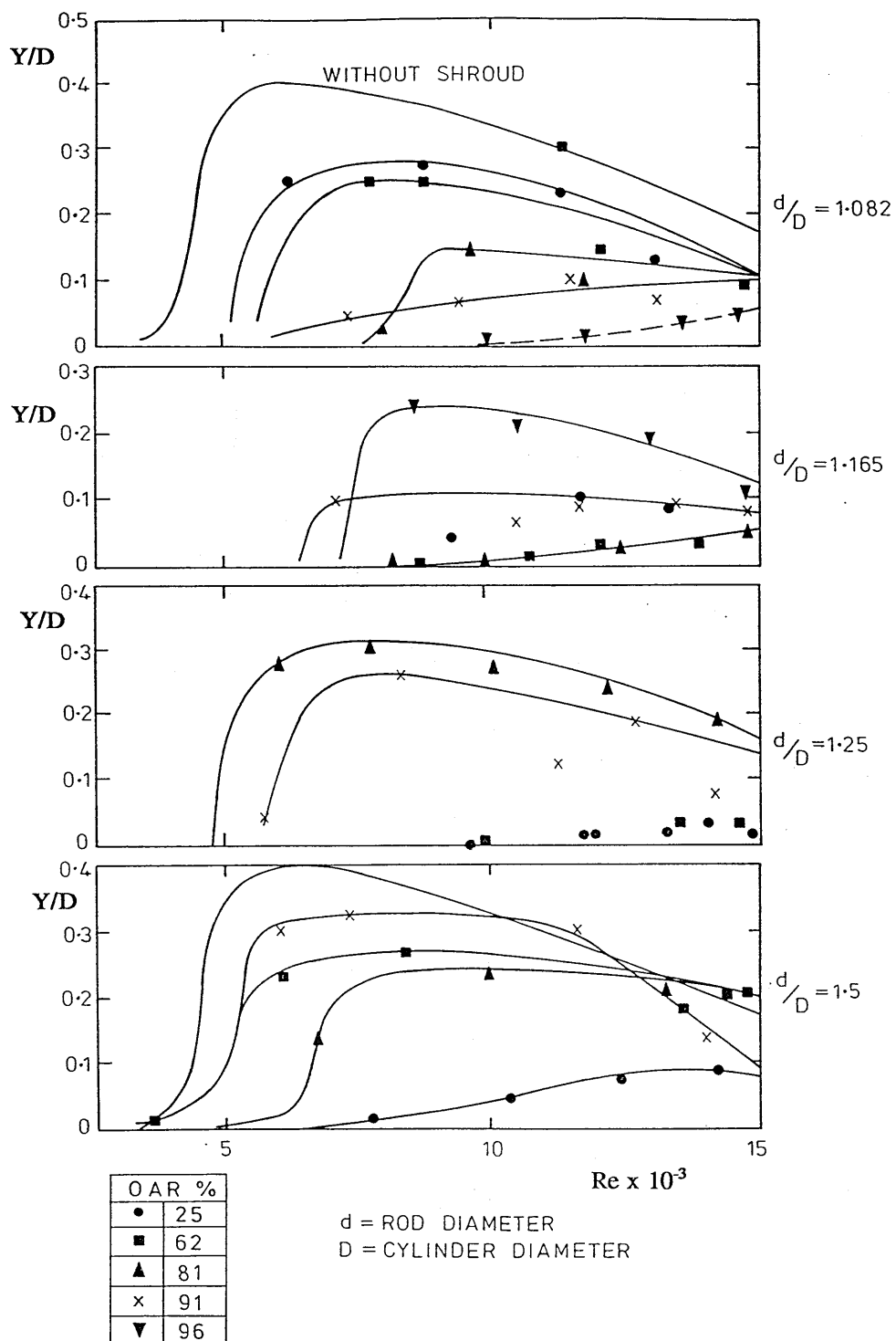


Fig. 3S.8

The effect of various axial rod shrouds on Reduced Amplitude (Zdravkovich, 1971).

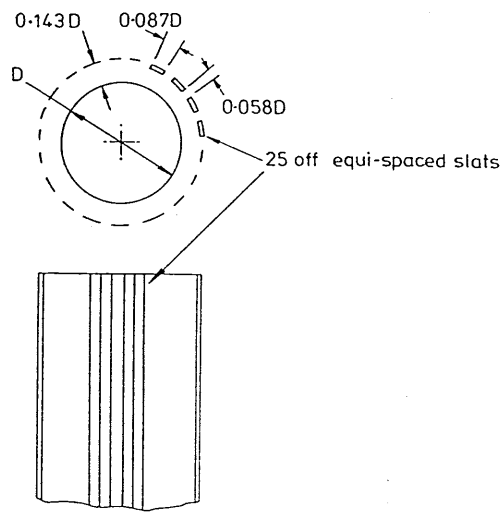


Fig. 3S.9 Wong's slat shroud (1977).

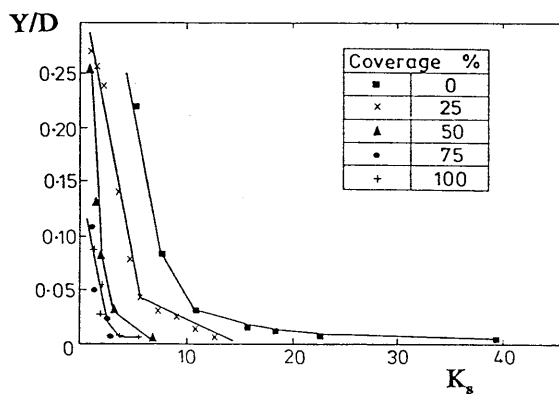


Fig. 3S.10 The effect of slat shroud coverage on Reduced Amplitude (Wong, 1977).

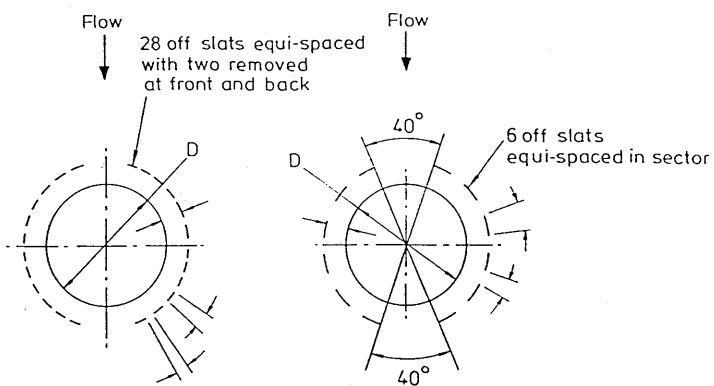


Fig. 3S.11 Every & King's slat shroud (1979).

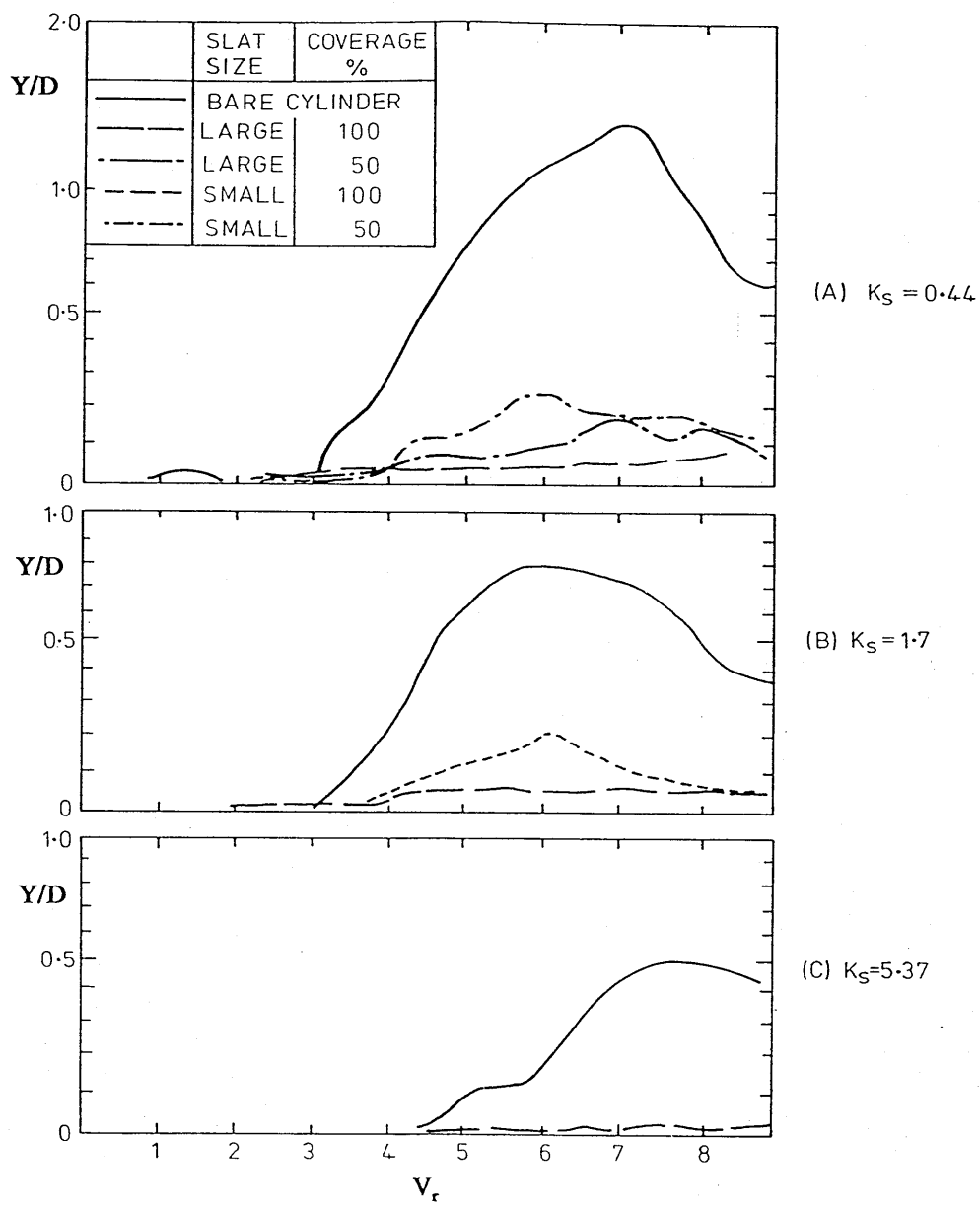


Fig. 3S.12

The effect of slat shrouds on Reduced Amplitude (Every & King, 1979).

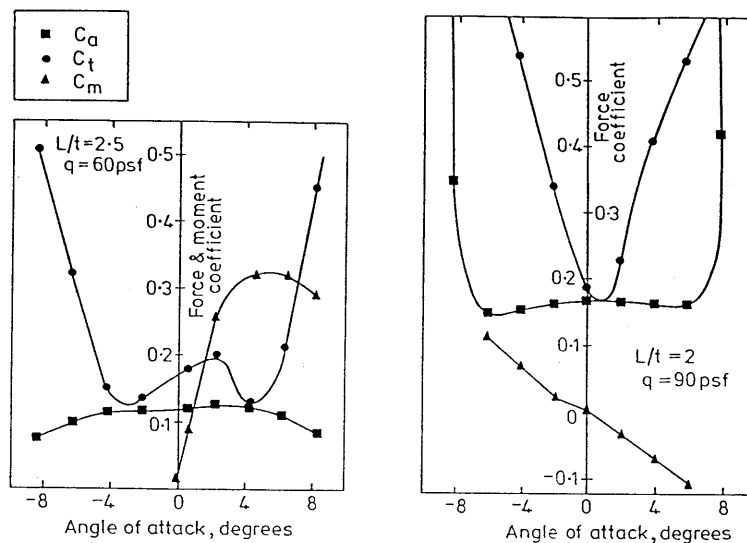


Fig. 3W.1 The effect of angle of attack on the force coefficients for model fairings (Grant and Patterson, 1977).

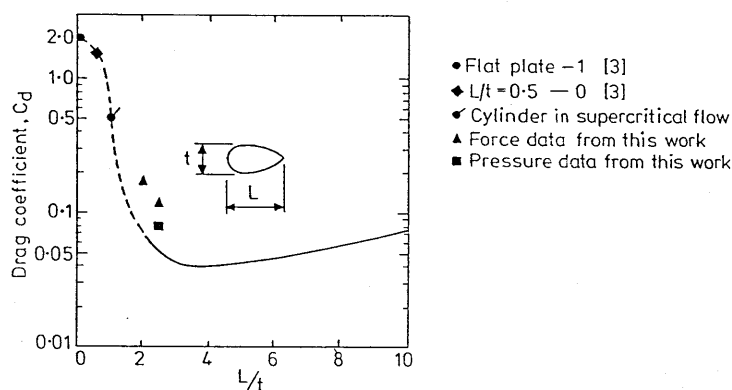


Fig. 3W.2 The variation of drag coefficient with c/t ratio for a general fairing design (Grant and Patterson, 1977).

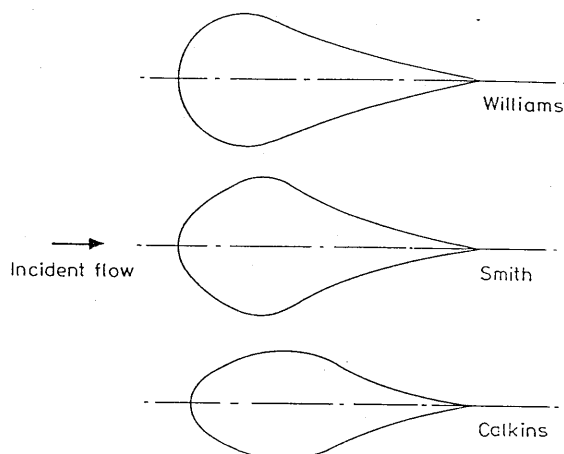


Fig. 3W.3 Various designs of stubby fairing (Packwood & Williams, 1986).

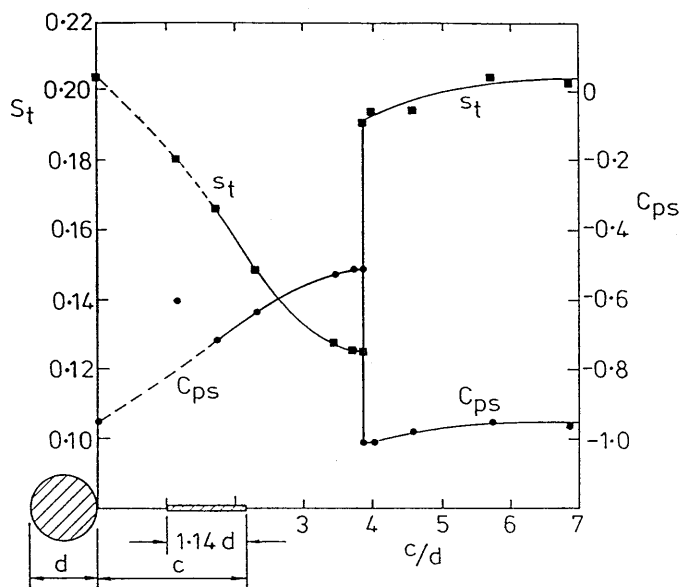


Fig. 3W.4

The effect of splitter plate position on the wake characteristics of a cylinder (Roshko, 1954).

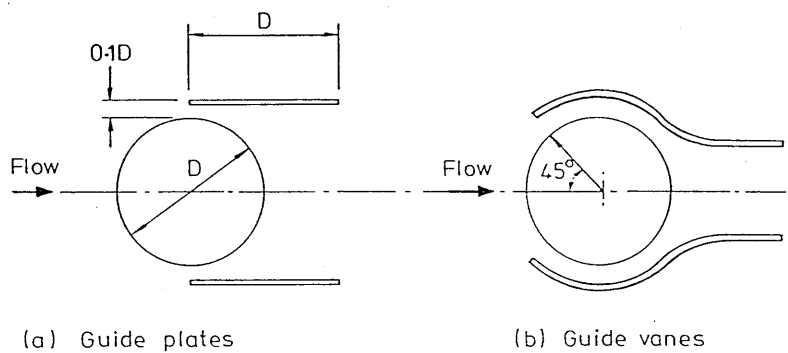


Fig. 3W.5

Grimminger's guiding vanes (1945).

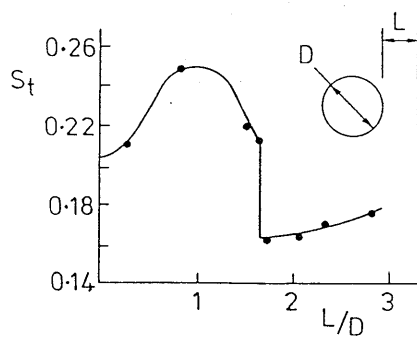


Fig. 3W.6

The effect of a perpendicular guide vane on Strouhal number (Gerrard, 1966).

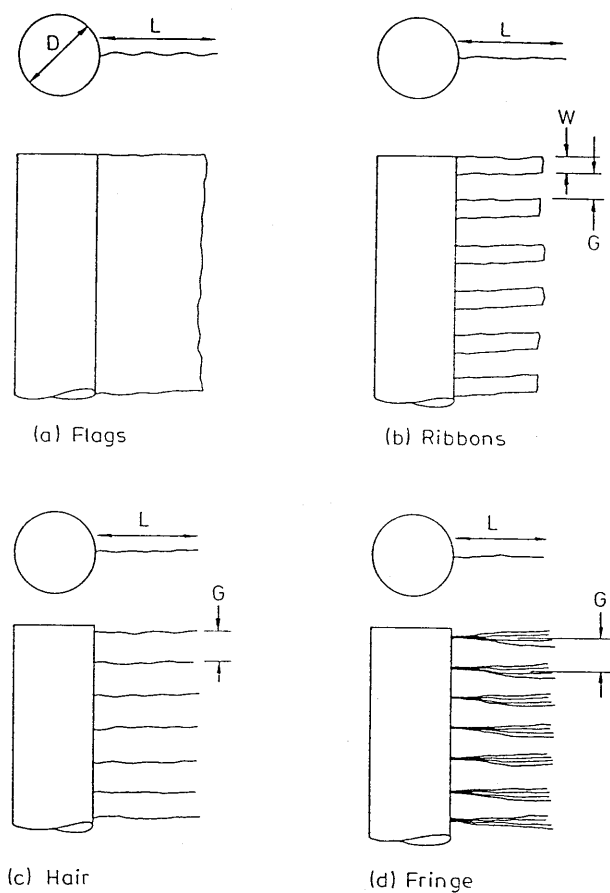


Fig. 3W.7 Flexible wake suppression devices.

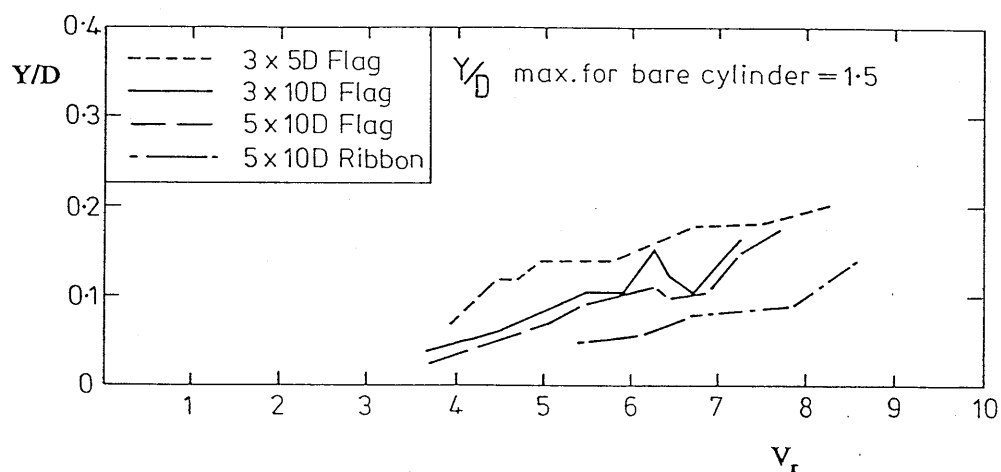


Fig. 3W.8 The effect of various flags on Reduced Amplitude (King, 1978).

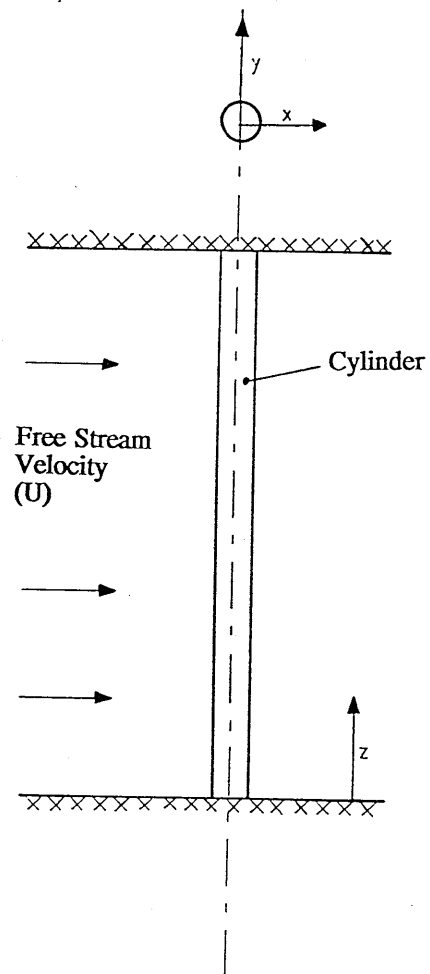


Fig. 5.1

Bare cylinder in a uniform flow.

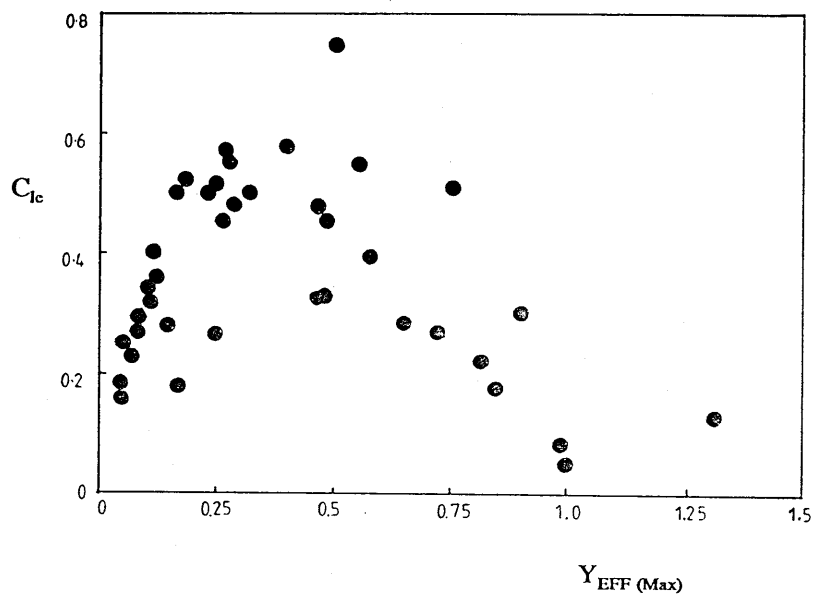


Fig. 5.2 Effective lift Coefficient vs. Modified Reduced Amplitude (Griffin, June 1985).

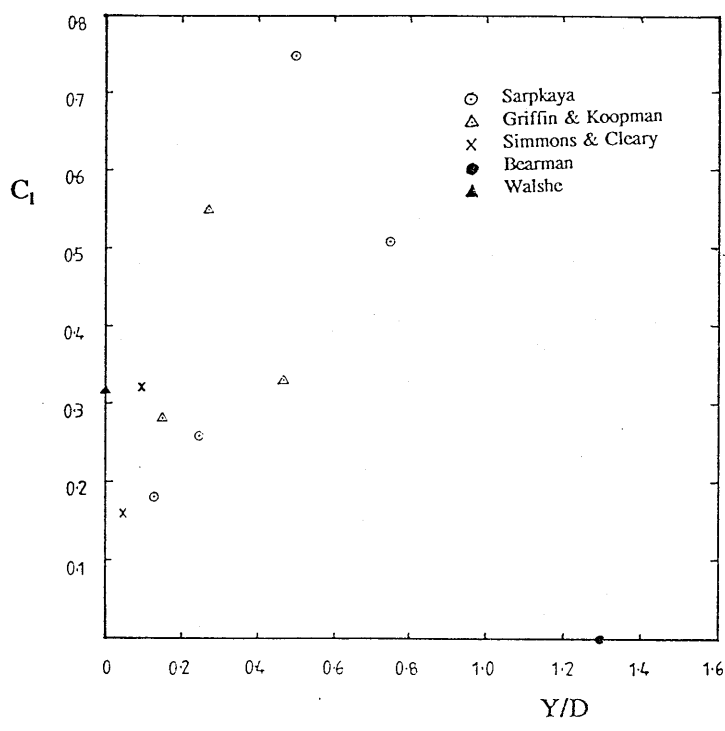


Fig. 5.3 Lift coefficient vs. Reduced Amplitude: Rigid cylinder results only.

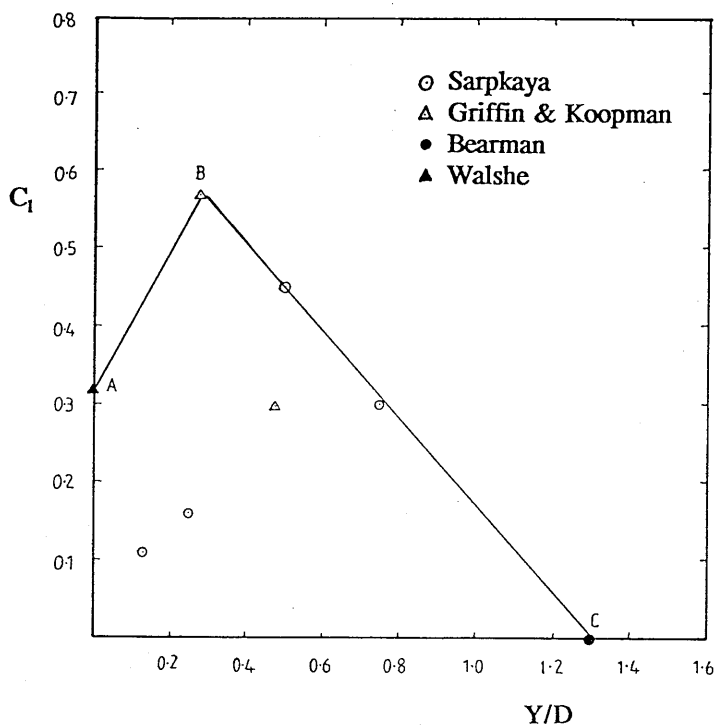


Fig. 5.4 Lift coefficient vs. Reduced Amplitude: Rigid cylinder results only, data points corrected for differences in V_r .

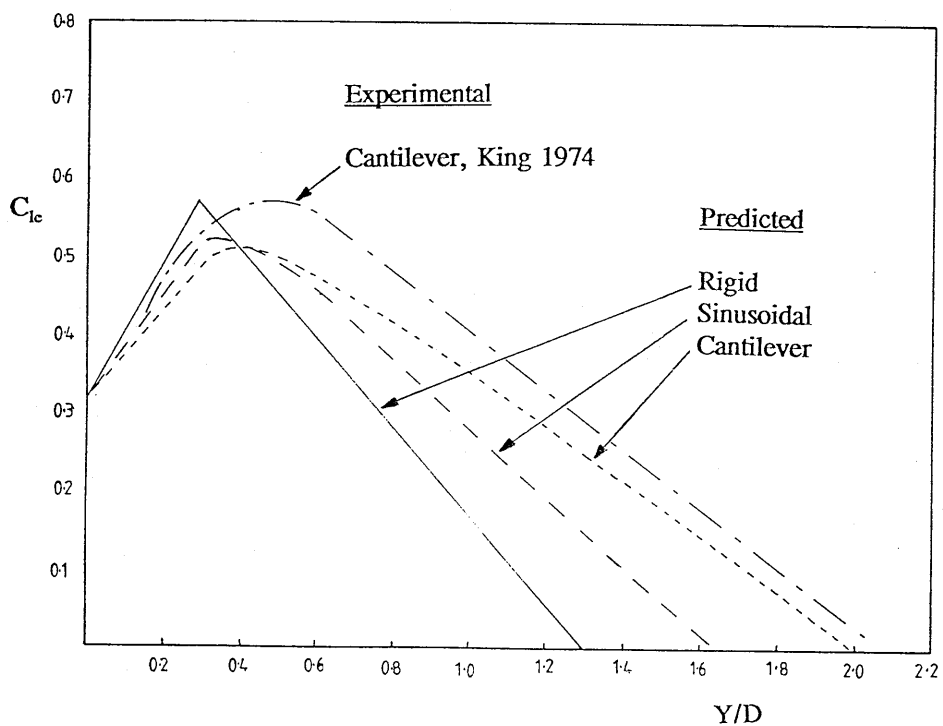
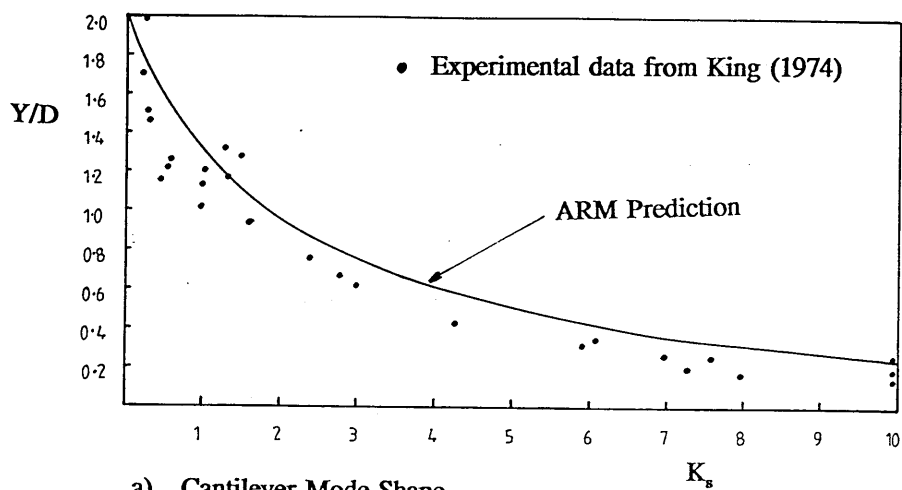
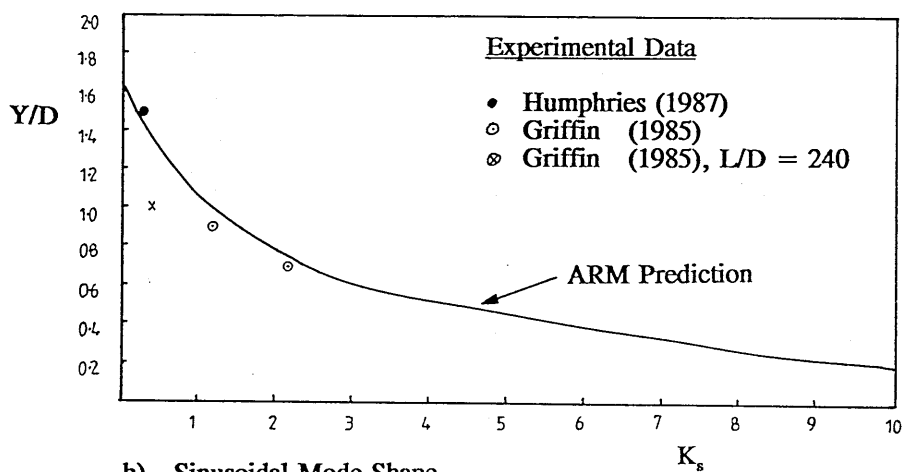


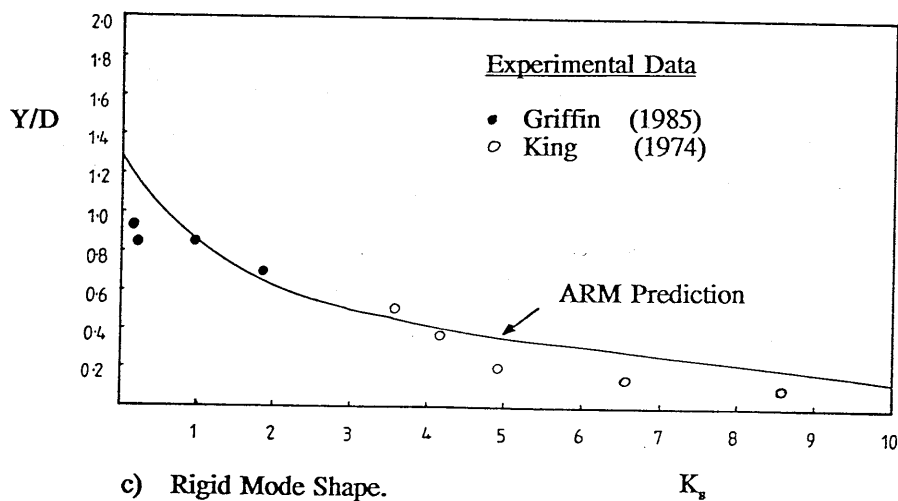
Fig. 5.5 Predicted Effective lift coefficient vs. Reduced Amplitude for a variety of mode shapes.



a) Cantilever Mode Shape.



b) Sinusoidal Mode Shape.



c) Rigid Mode Shape.

Fig. 5.6

Predicted and measured Reduced Amplitude vs. Stability Parameter for a variety of mode shapes.

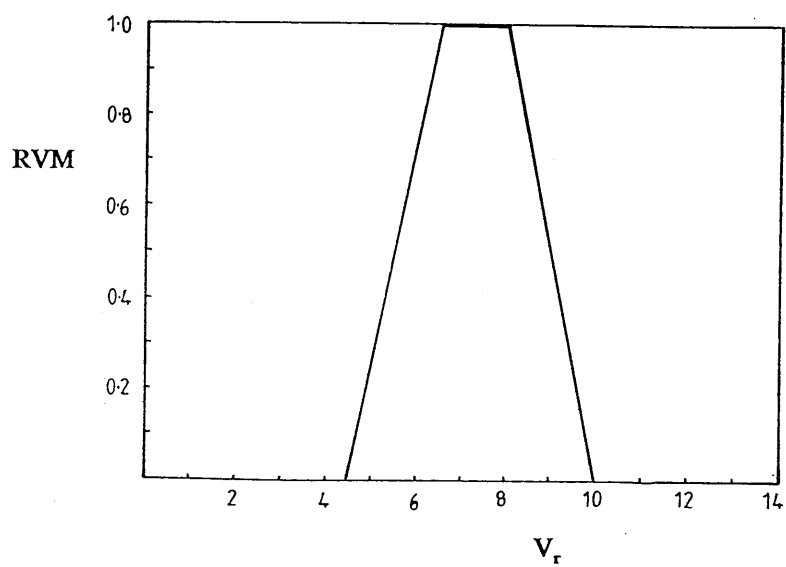
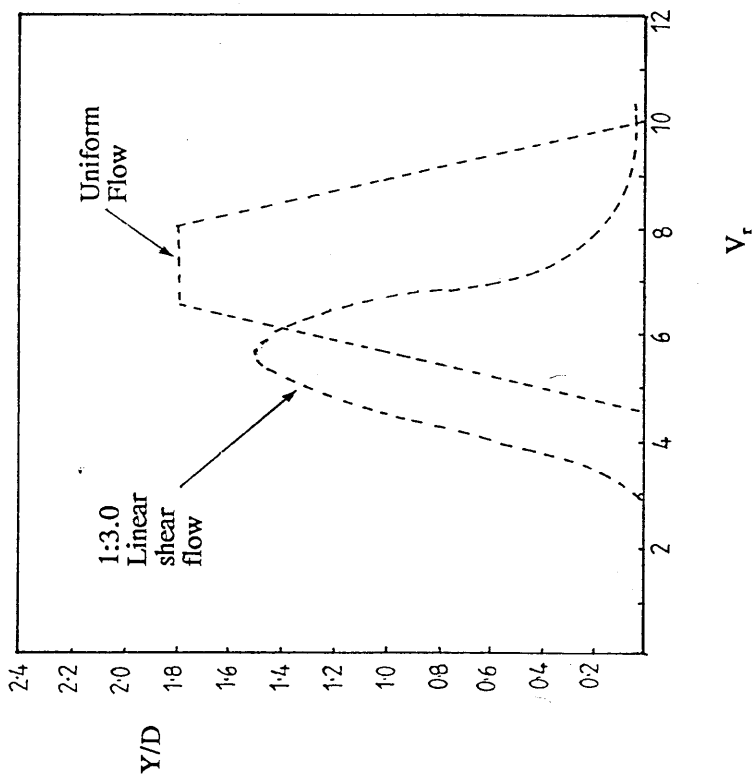
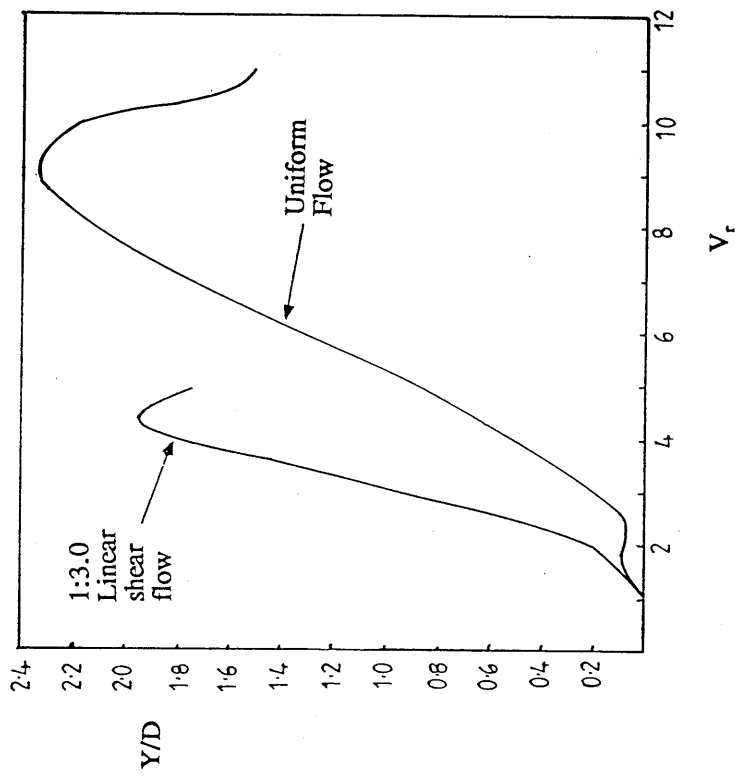


Fig. 5.7 Reduced Velocity multiplier (RVM) vs. Reduced Velocity.



a) Predicted



b) Measured

Fig. 5.8 A comparison between predicted and measured Reduced Amplitude in uniform and sheared flow: $K_s = 0.22$.

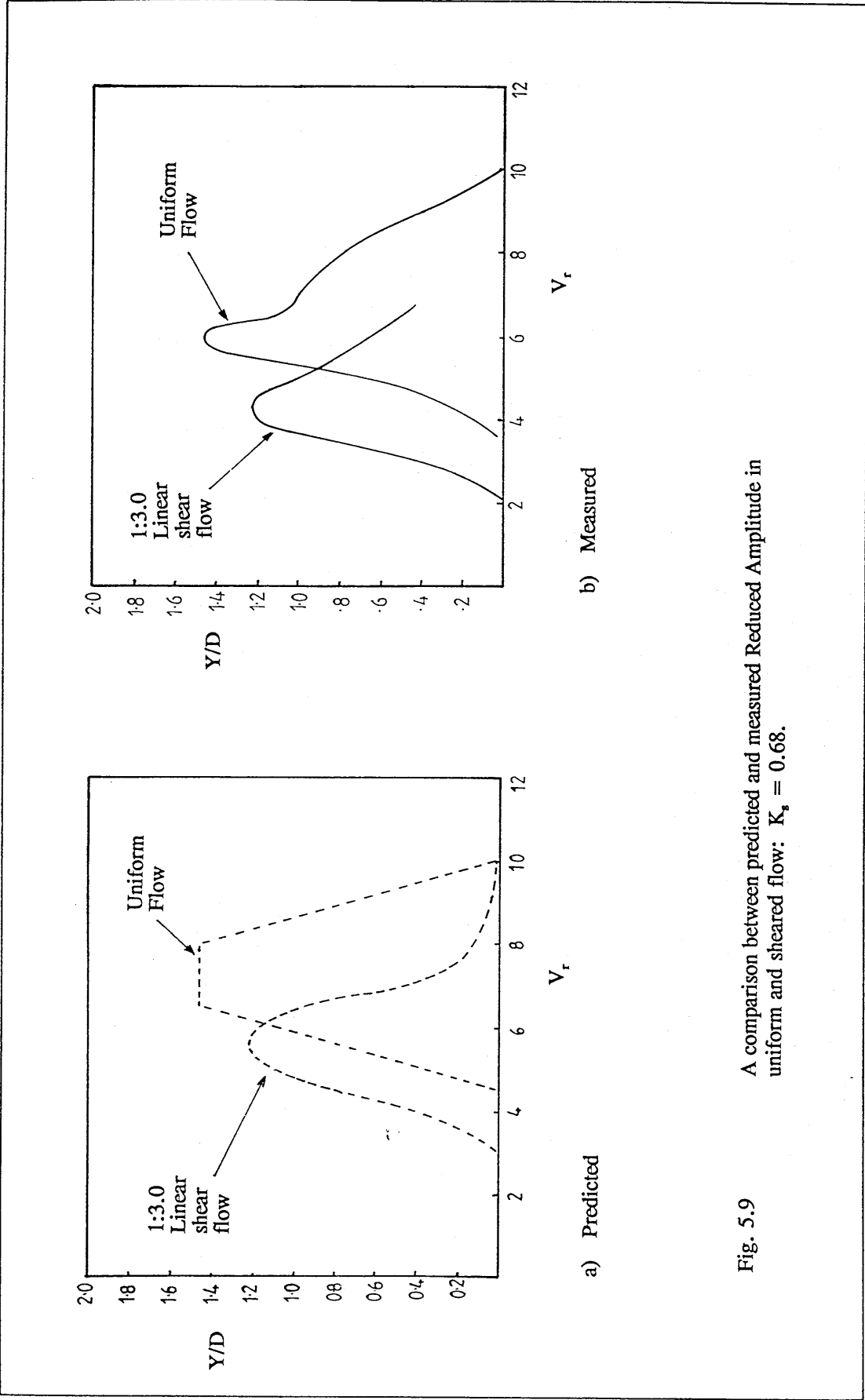
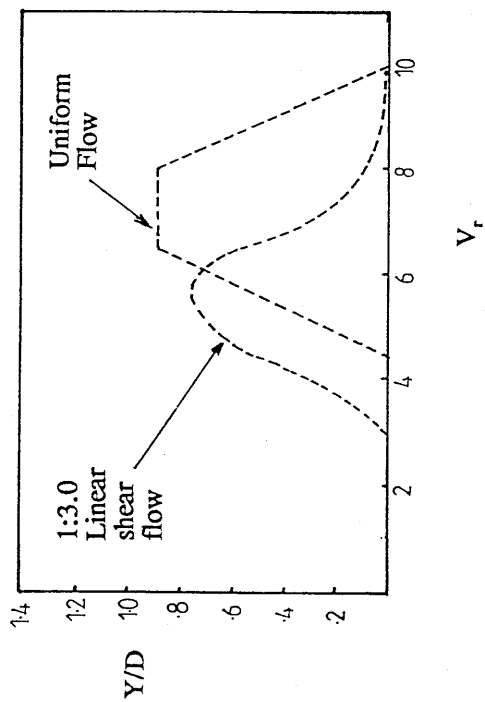
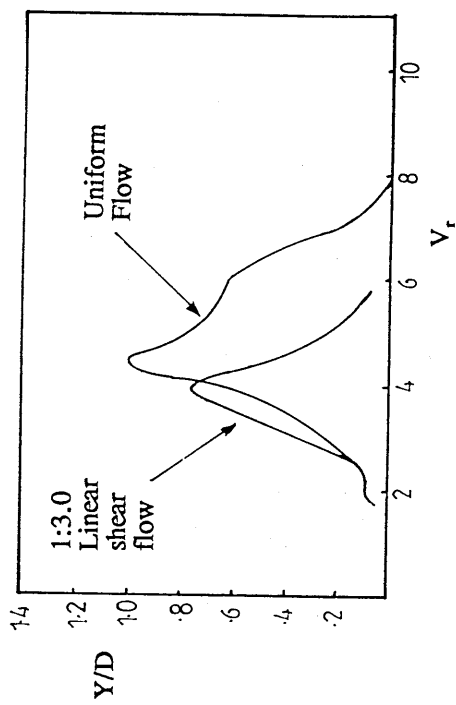


Fig. 5.9 A comparison between predicted and measured Reduced Amplitude in uniform and sheared flow: $K_s = 0.68$.



a) Predicted



b) Measured

Fig. 5.10 A comparison between predicted and measured Reduced Amplitude in uniform and sheared flow: $K_s = 2.30$.

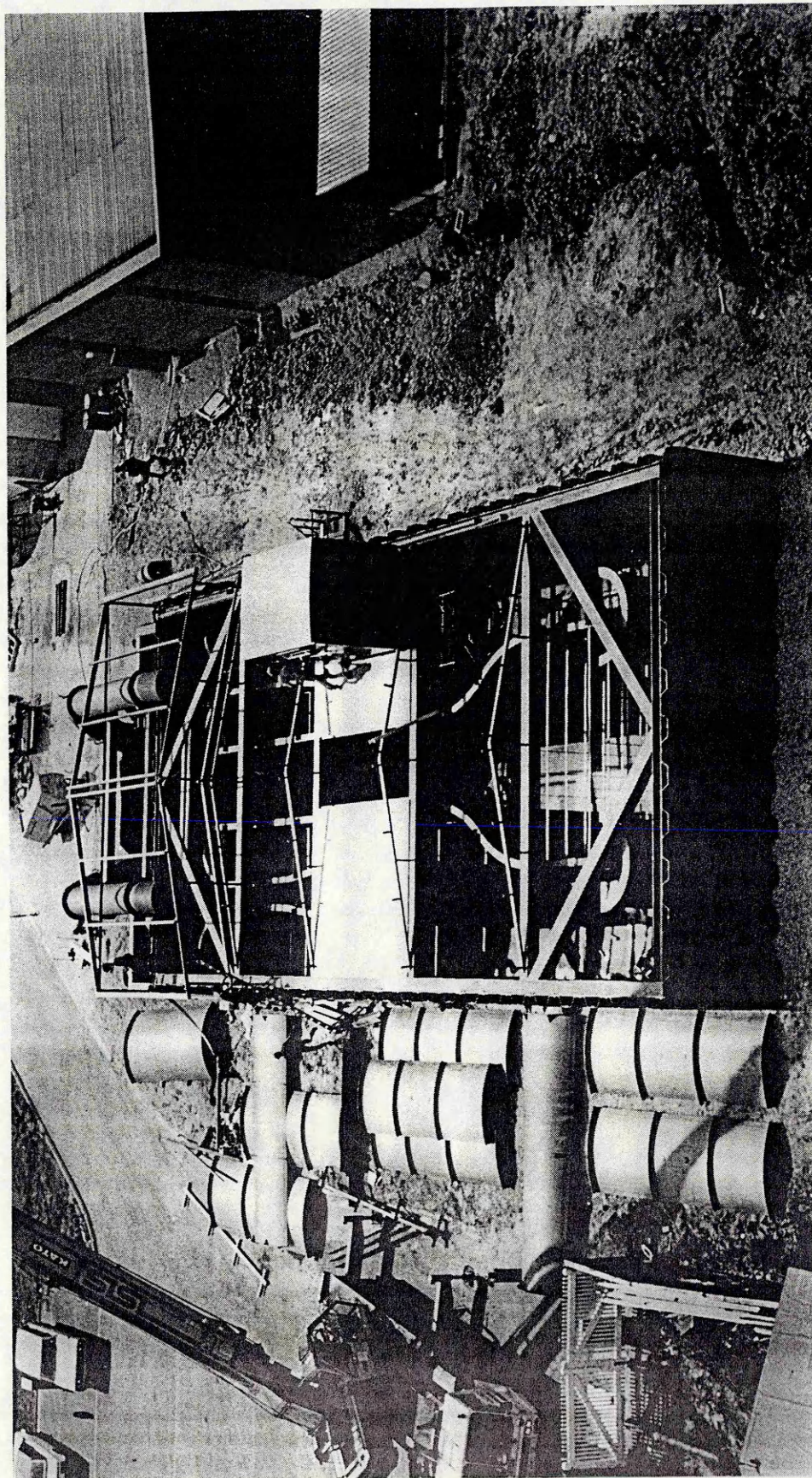


Fig. 6.2.1 The Deep Water Flume under construction.

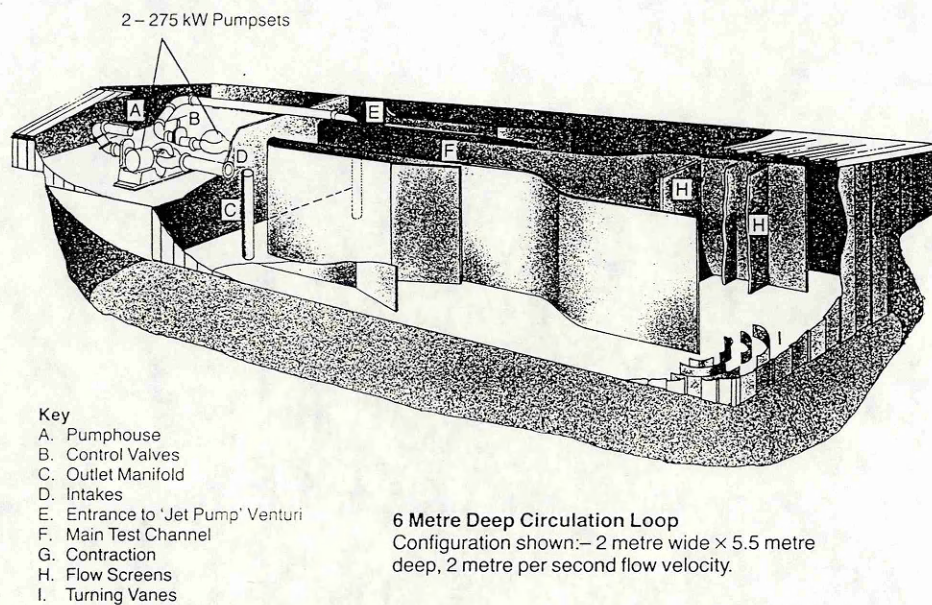


Fig. 6.2.2 Cutaway illustration of the Deep Water Flume showing the internal steel fabrication.

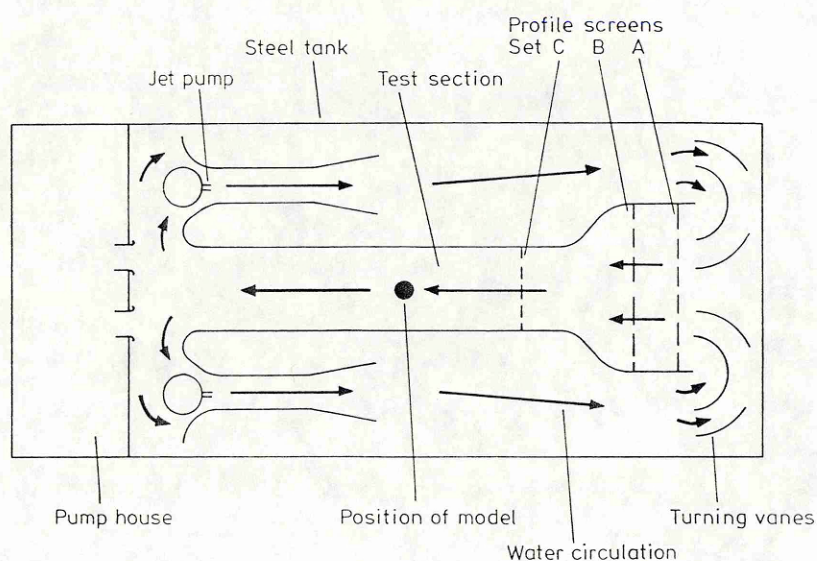


Fig. 6.2.3 Plan view of the Deep Water Flume showing the path of the circulating water.

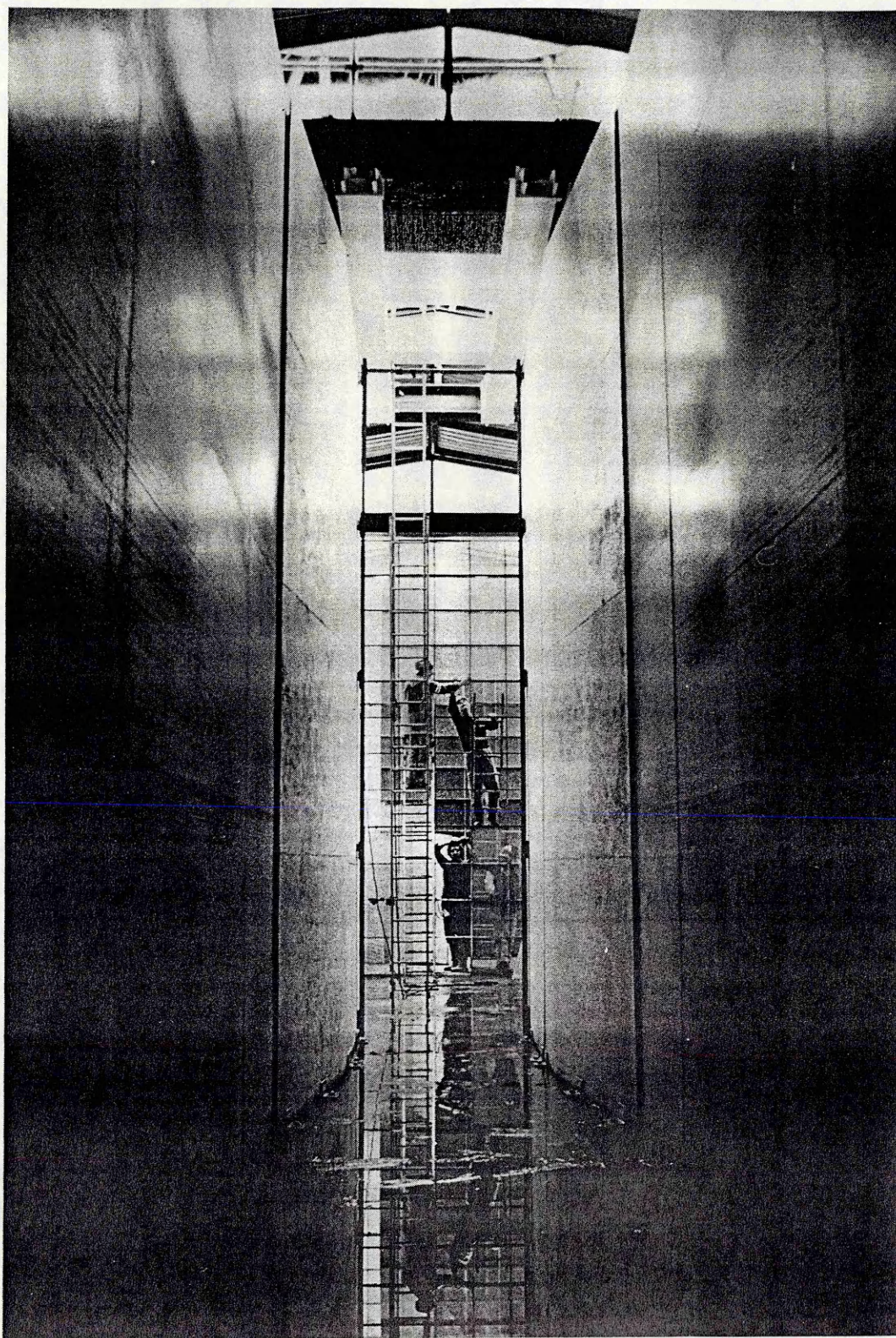


Fig. 6.2.4 The 6 m deep test section of the Deep Water Flume.

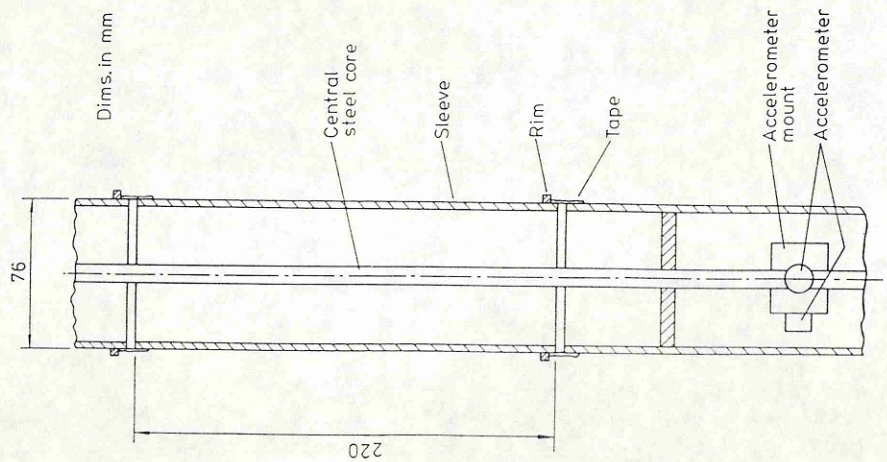


Fig. 6.2.5 Sectional drawing of Model 1.

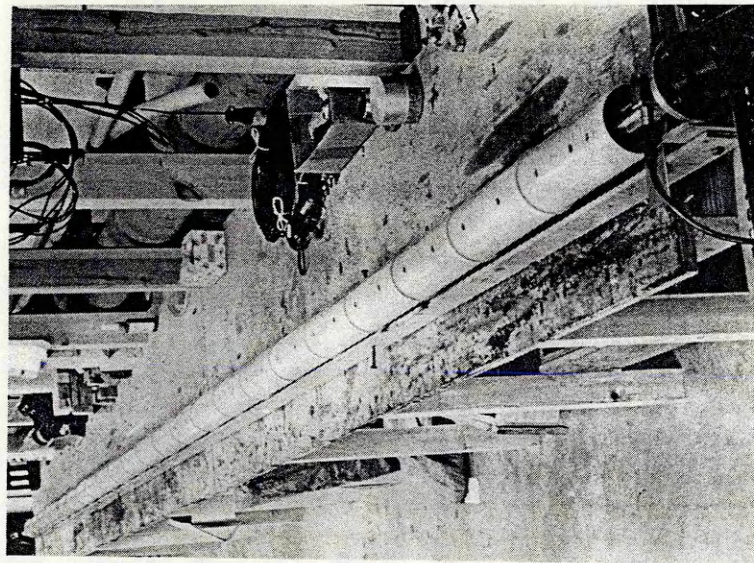


Fig. 6.2.6 Model 1 fully assembled prior to fitting in the DWF.

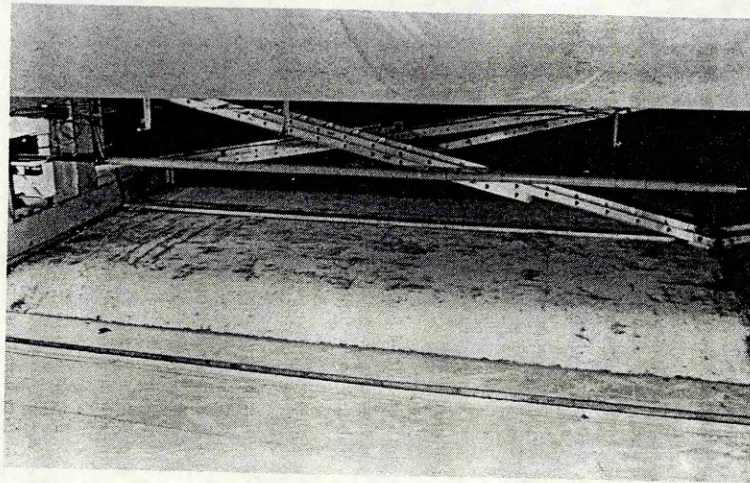


Fig. 6.2.7 Model 1 located in the test section of the Deep Water Flume.

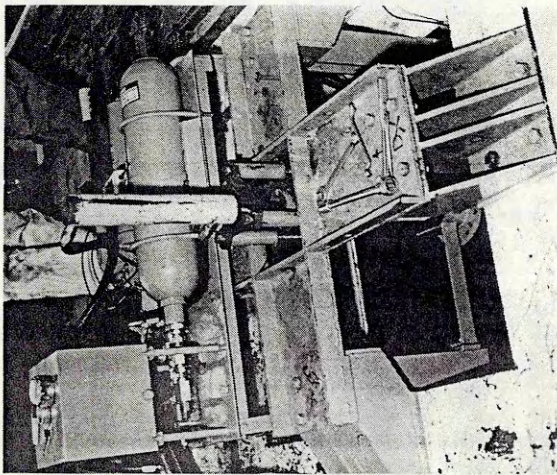


Fig. 6.2.8 The hydraulic tensioning unit.

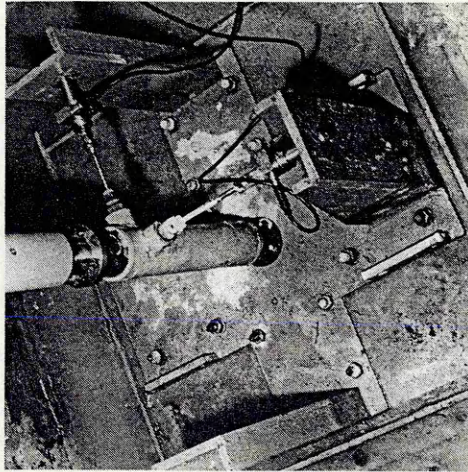


Fig. 6.2.9 Fixing arrangement for the bottom of the models.

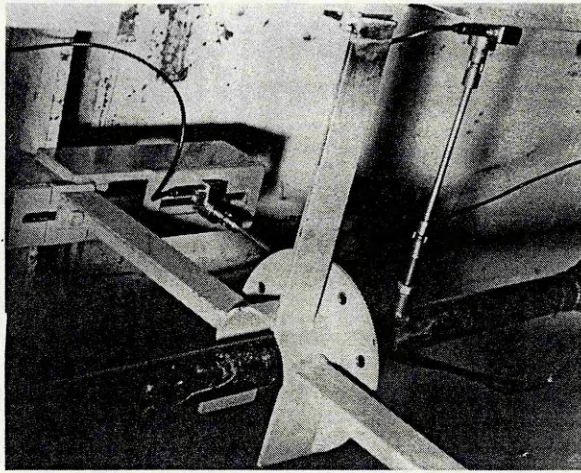


Fig. 6.2.10 Fixing arrangement for the top of the models.

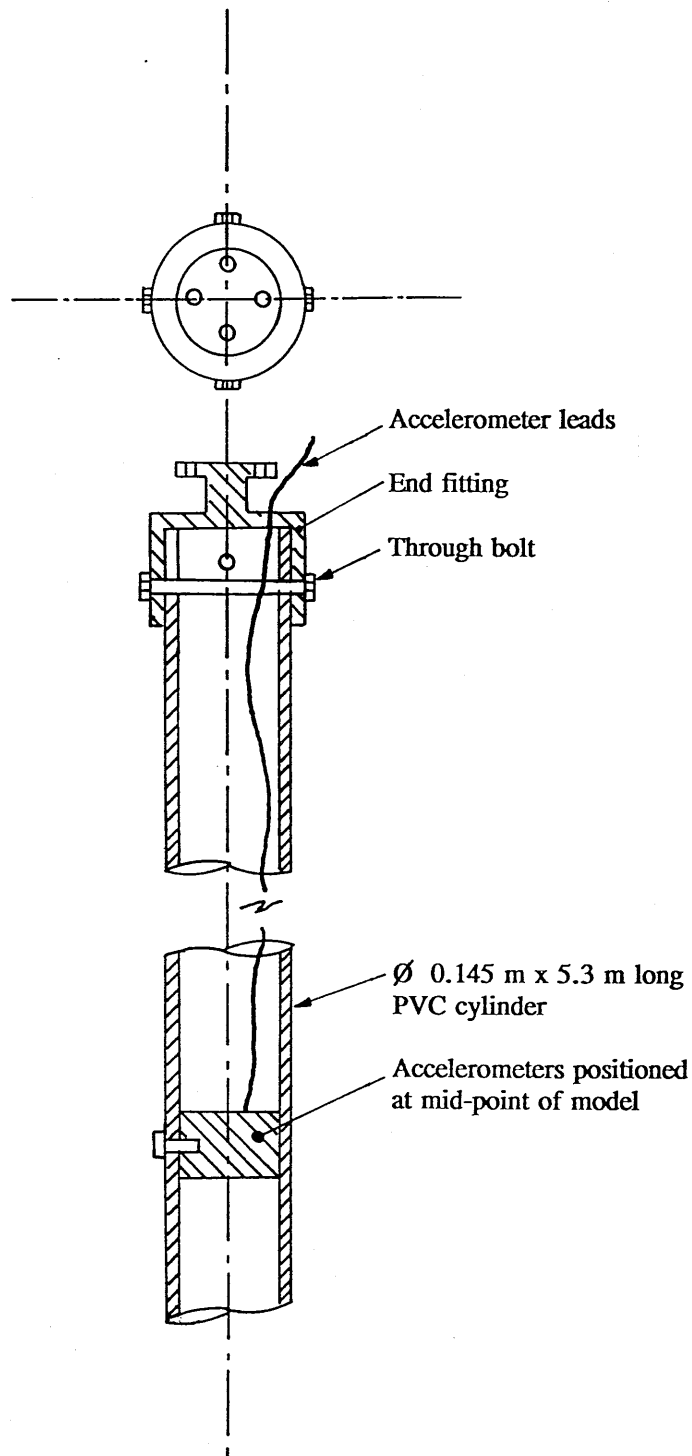


Fig. 6.2.11 Sectional drawing of Model 2.

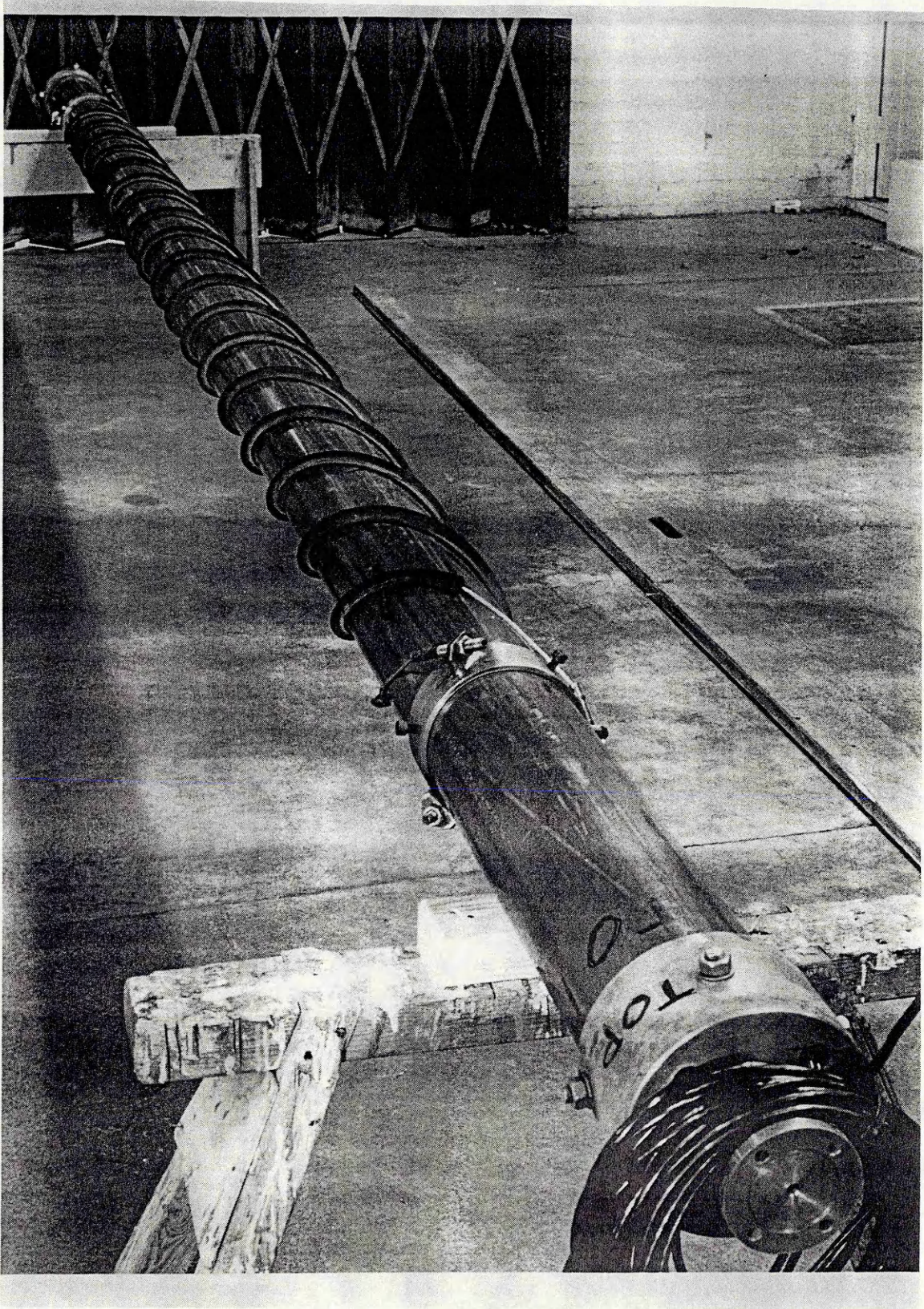


Fig. 6.2.12 Model 2 complete with 4.5 D pitch strake, prior to fitting in the DWF.

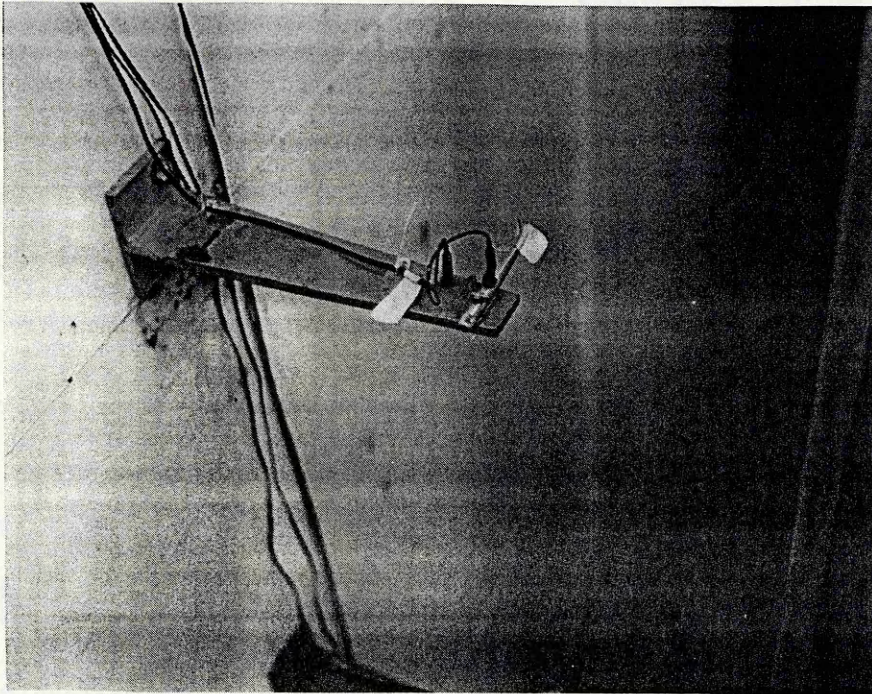


Fig. 6.2.13 An A.OTT propeller meter attached to the side of the DWF.

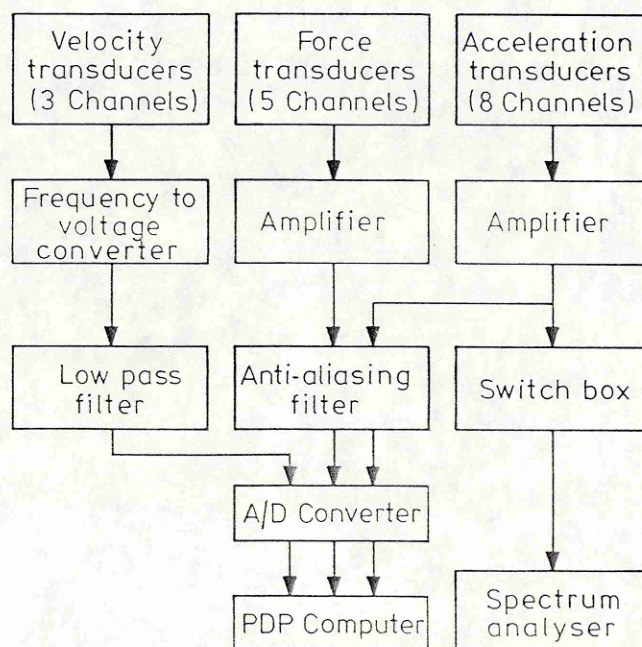
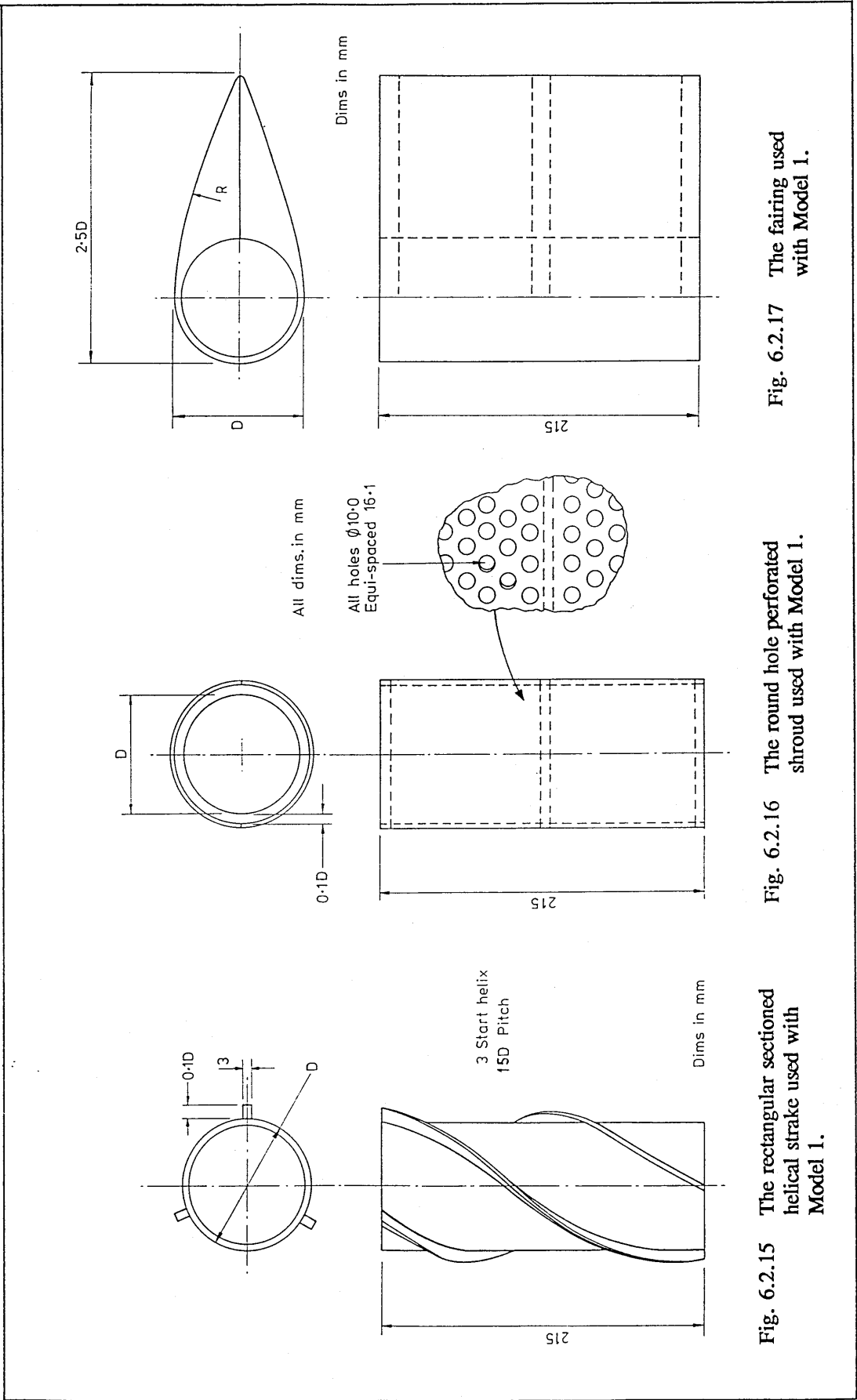


Fig. 6.2.14 Signal processing block diagram.



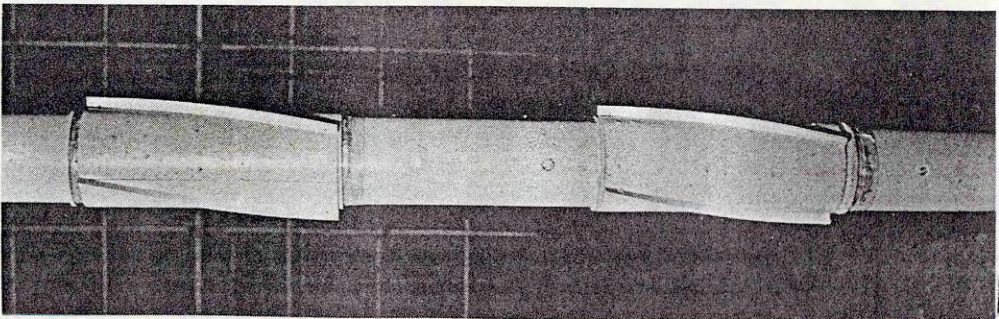
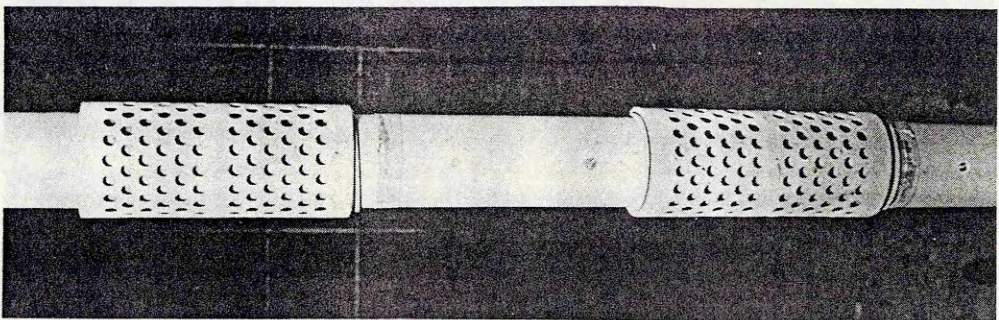
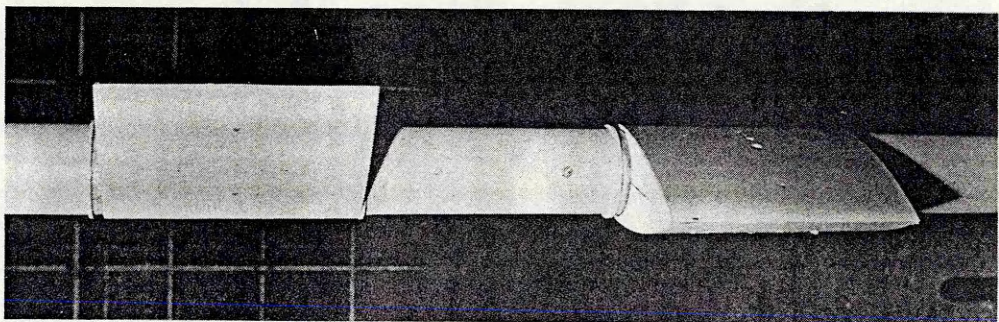
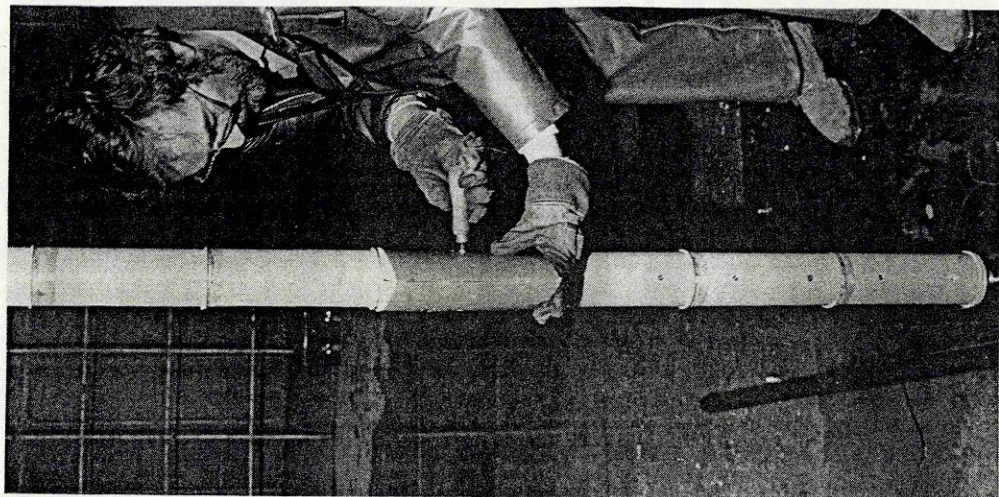


Fig. 6.2.18 Suppression devices fitted to Model 1.

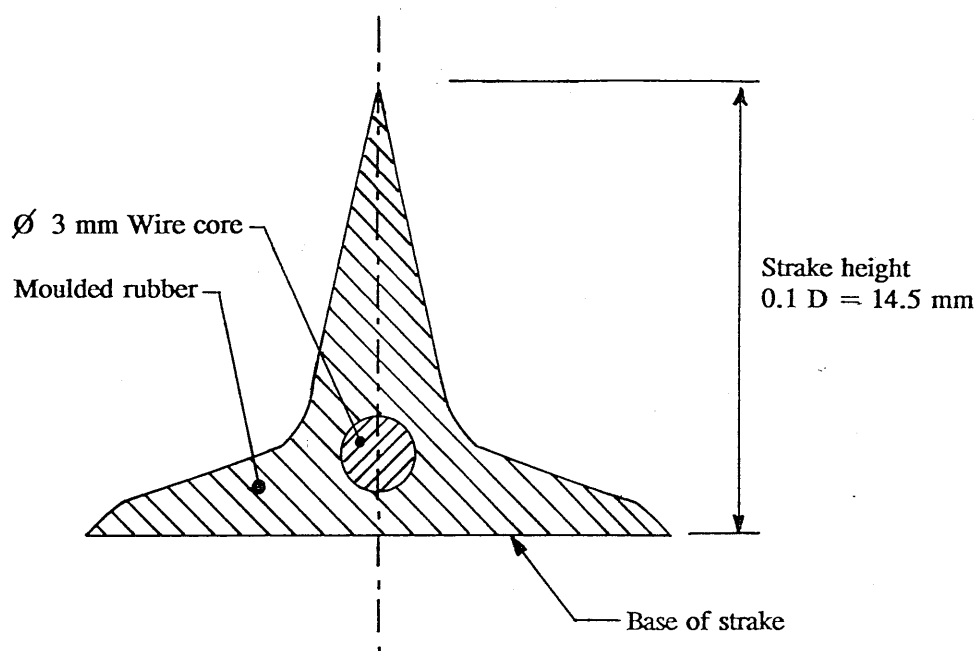


Fig. 6.2.19 Section through the Albrown Universal strake used with Model 2.

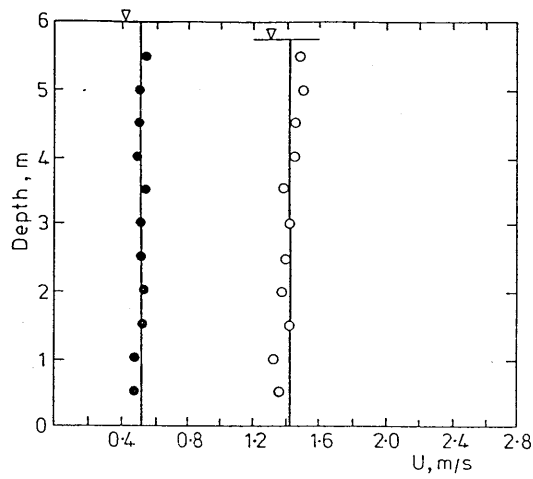


Fig. 6.4.1 The uniform velocity profile.

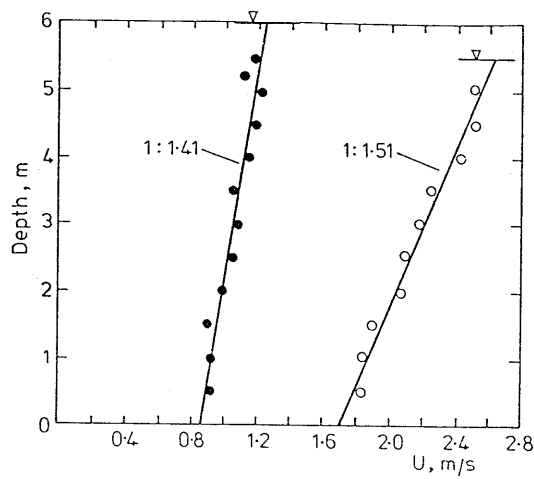


Fig. 6.4.2 The 1:1.5 linear shear velocity profile (LS15).

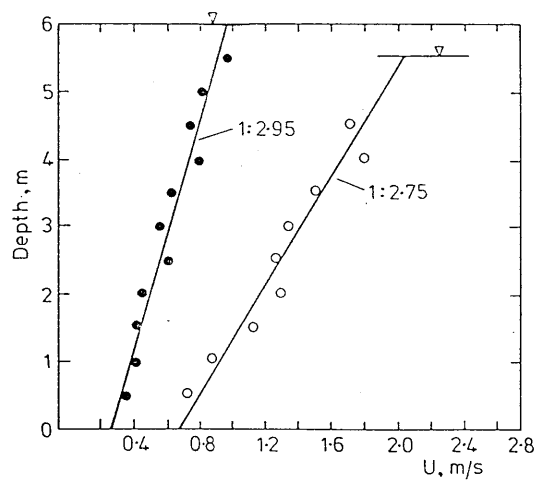


Fig. 6.4.3 The 1:3.0 linear shear velocity profile (LS30).

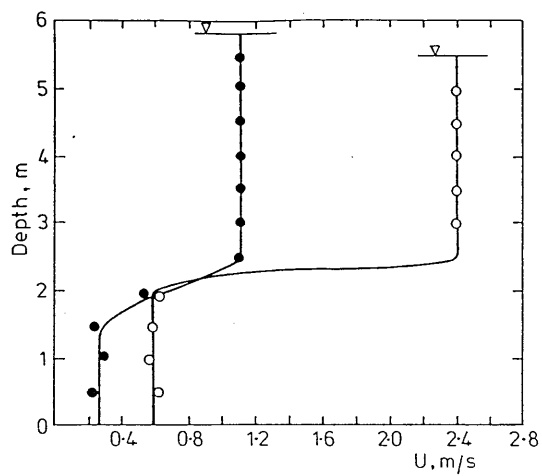


Fig. 6.4.4 The 55% stepped shear velocity profile (SS55).

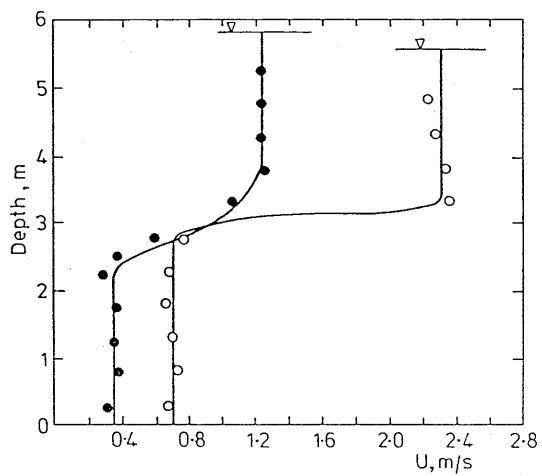


Fig. 6.4.5 The 40% stepped shear velocity profile (SS40).

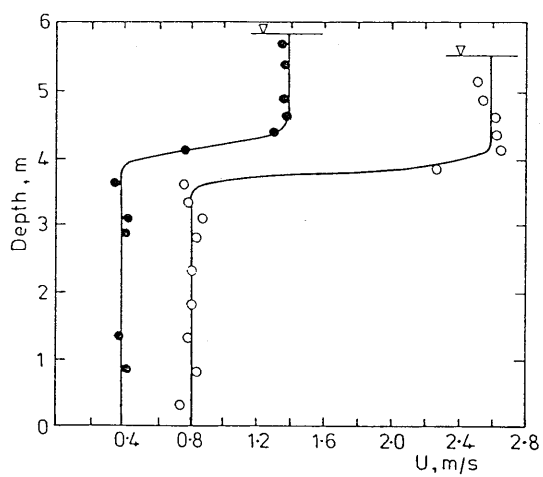
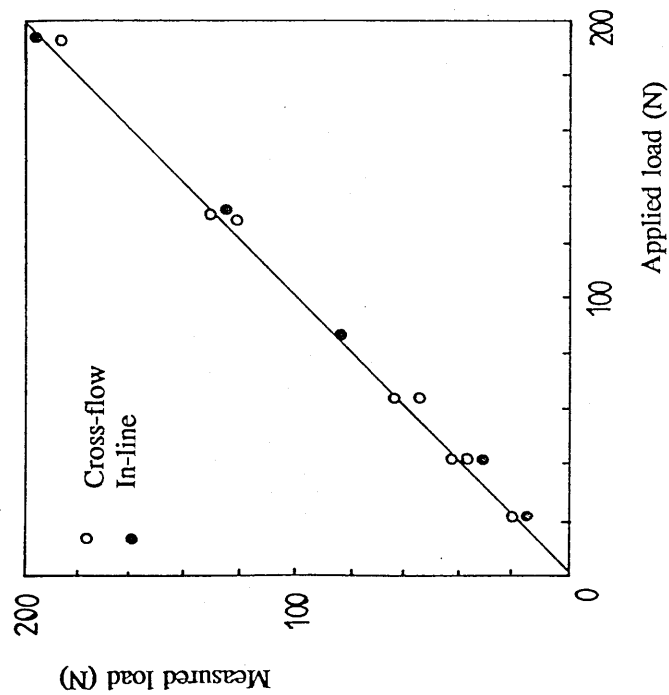
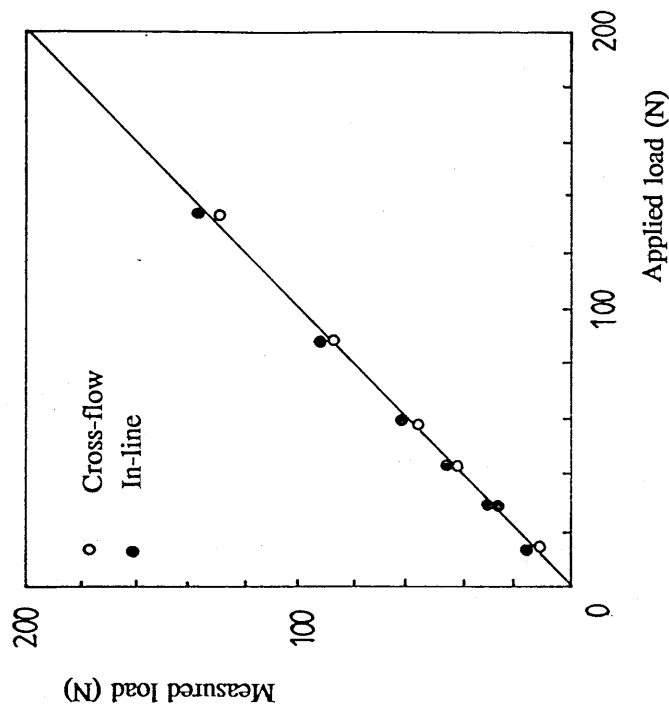


Fig. 6.4.6 The 24% stepped shear velocity profile (SS24).



a) Top load cells



b) Bottom load cells

Fig. 6.4.7 Load cell calibration: in-situ test.

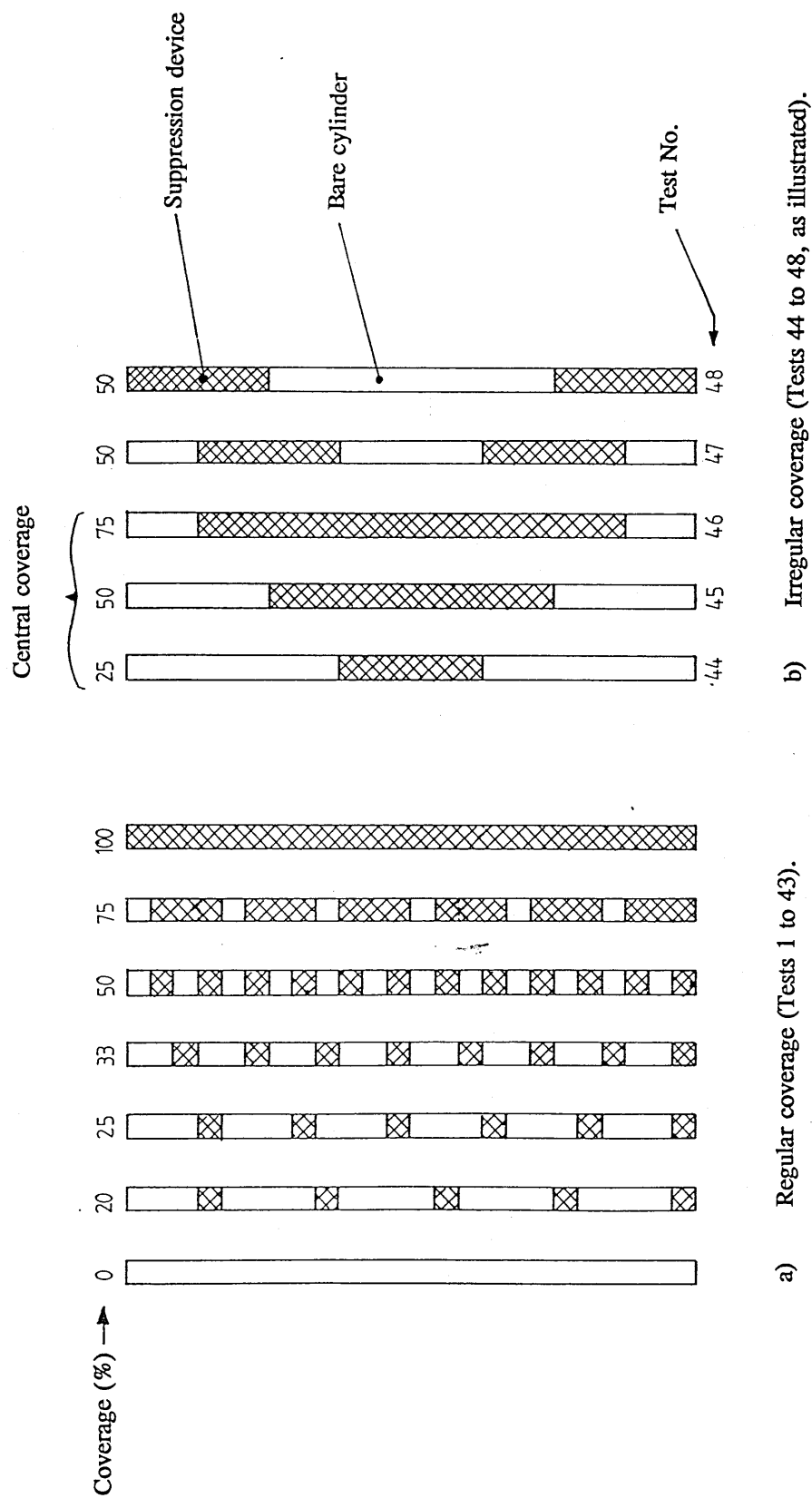


Fig. 6.6.1 Types of suppression device coverage used in the Model 1 test programme.

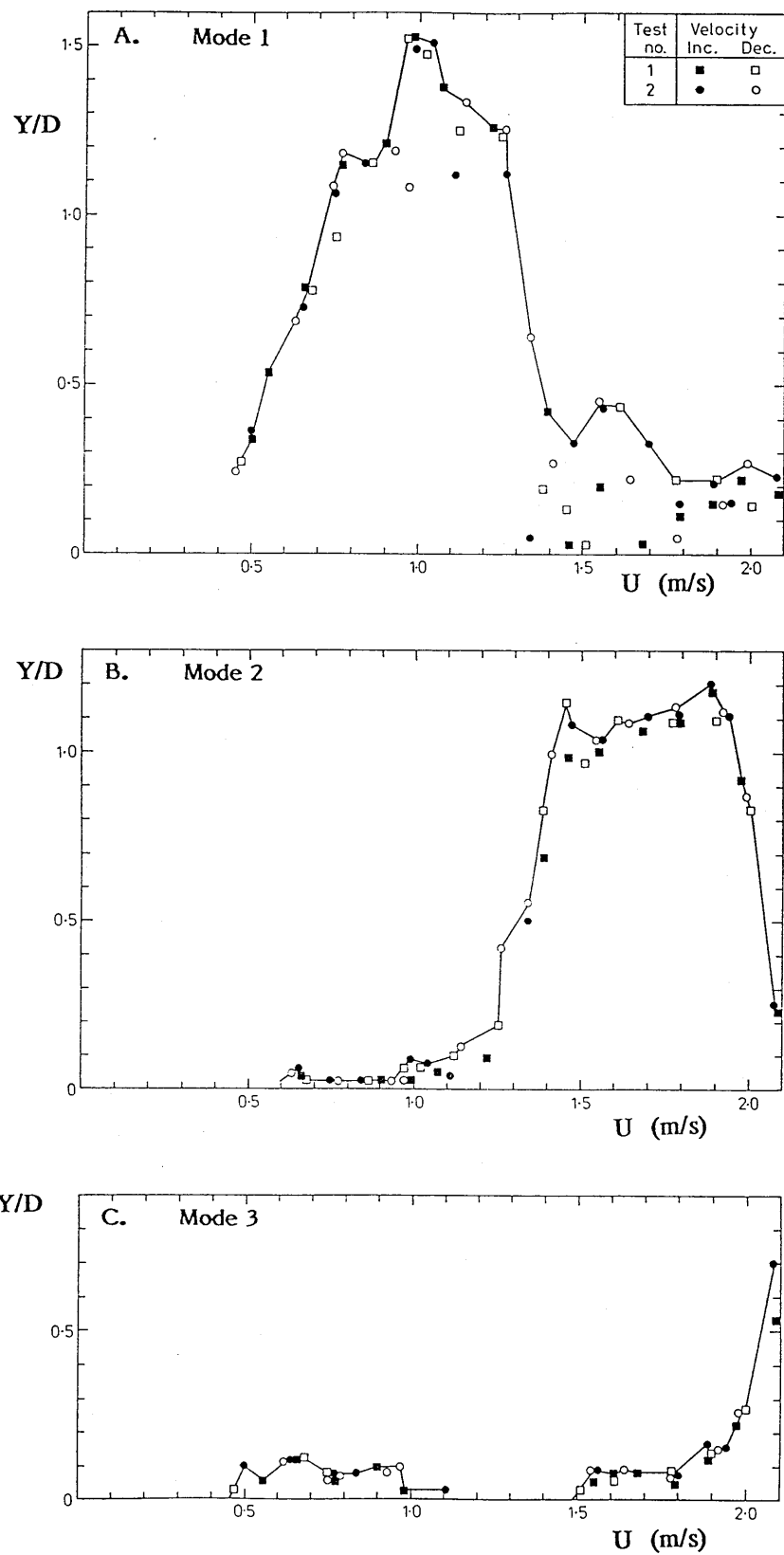


Fig. 6.7.1 Reduced Amplitude vs velocity.
(Uniform velocity profile, no device fitted)

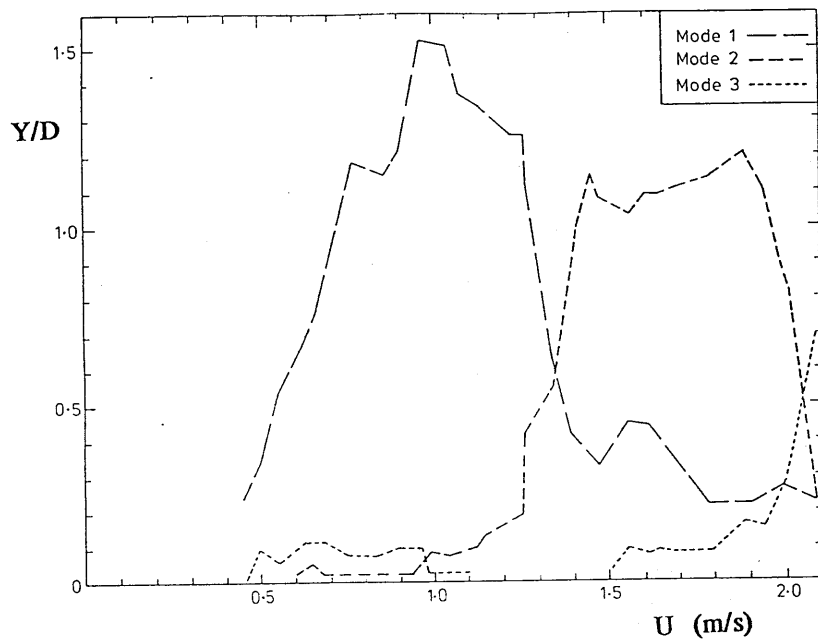


Fig. 6.7.2 Reduced Amplitude vs velocity.
(Uniform velocity profile, no device fitted)

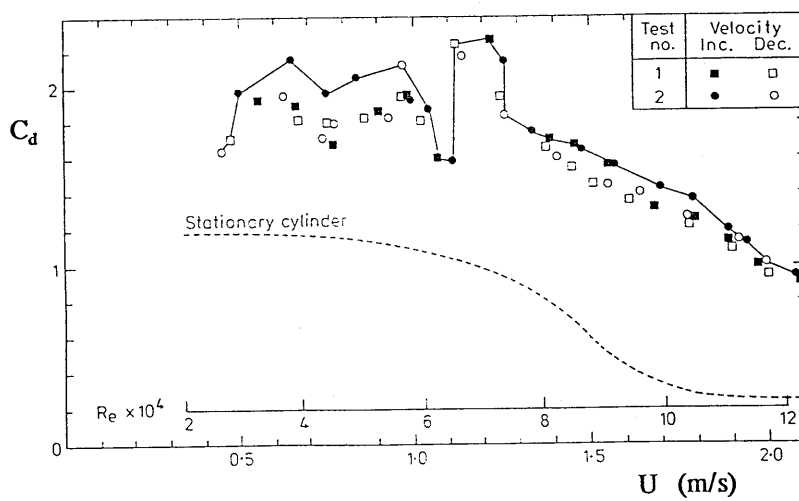


Fig. 6.7.3 Drag coefficient vs velocity.
(Uniform velocity profile, no device fitted)

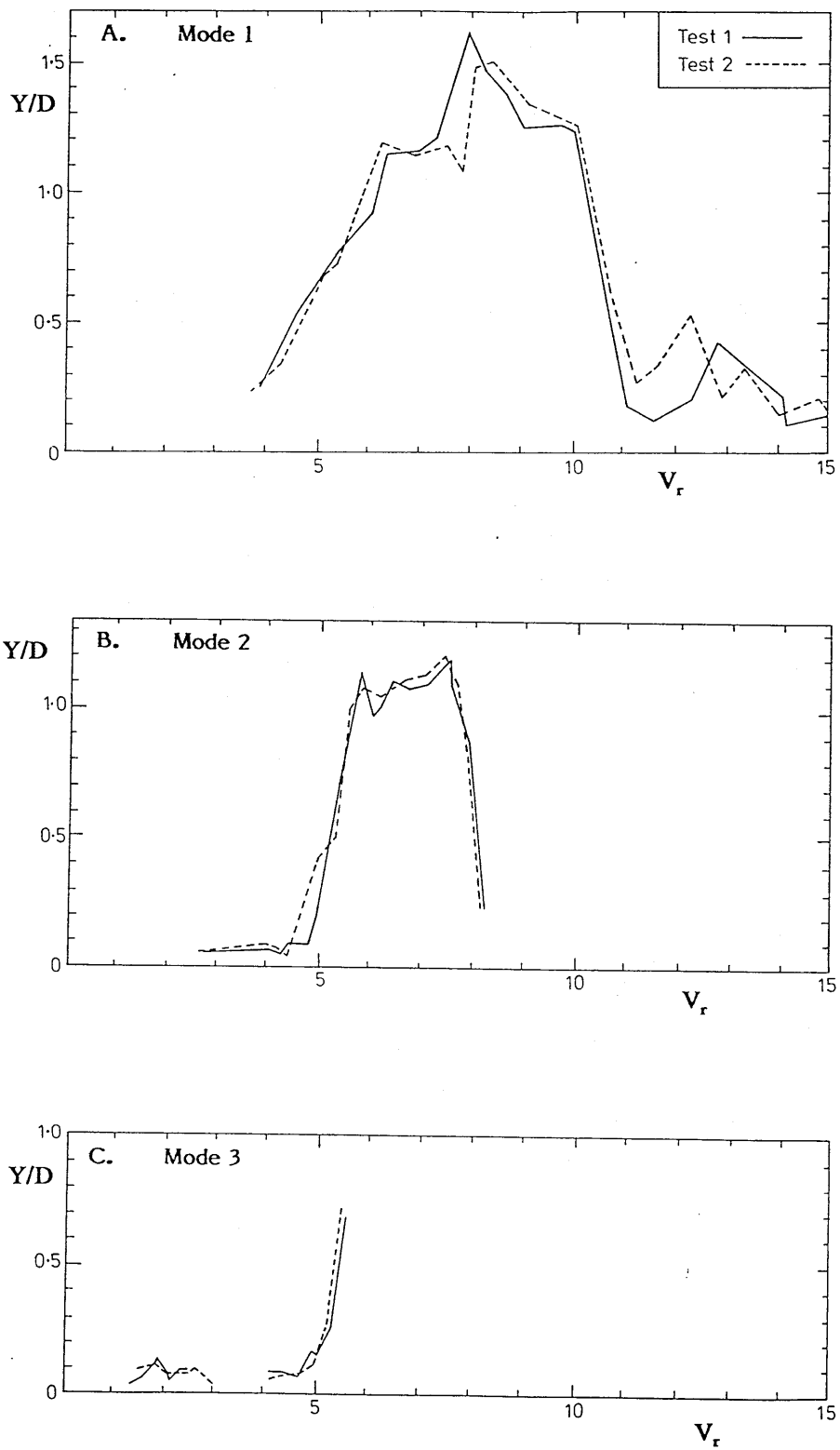


Fig. 6.7.4 Reduced Amplitude vs Reduced Velocity.
(Uniform velocity profile, no device fitted)

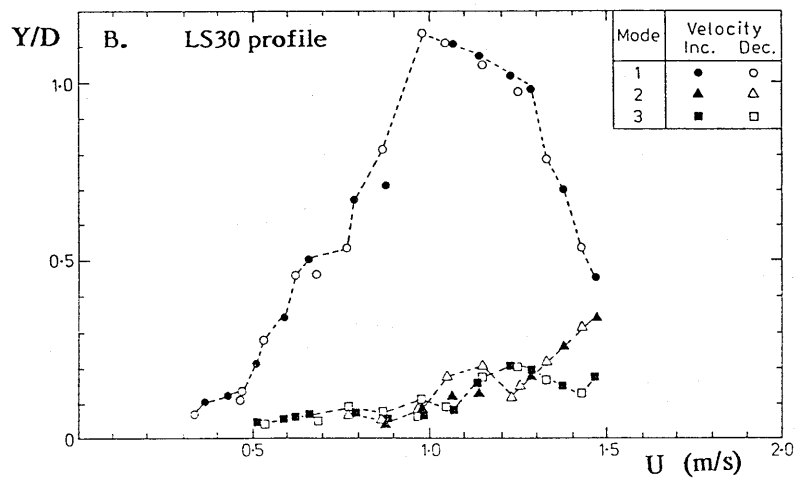
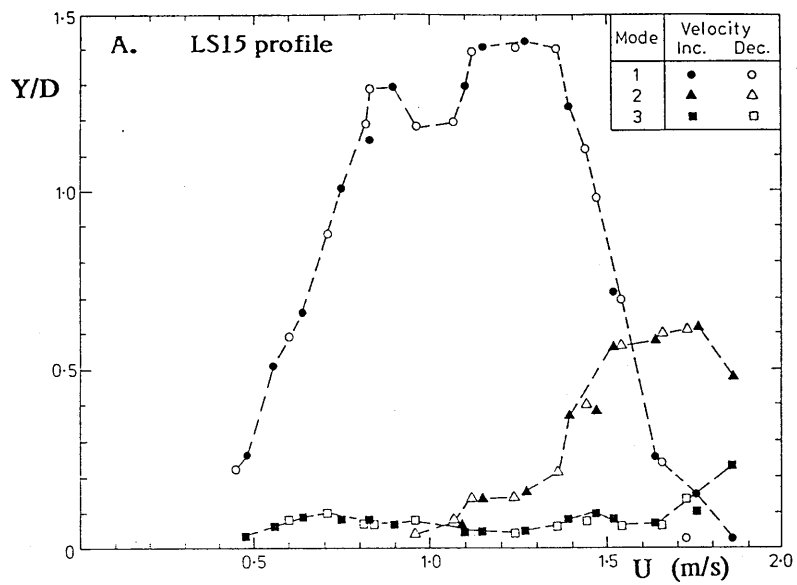


Fig. 6.7.5 Reduced Amplitude vs velocity.
(Linear shear velocity profiles, no device fitted)

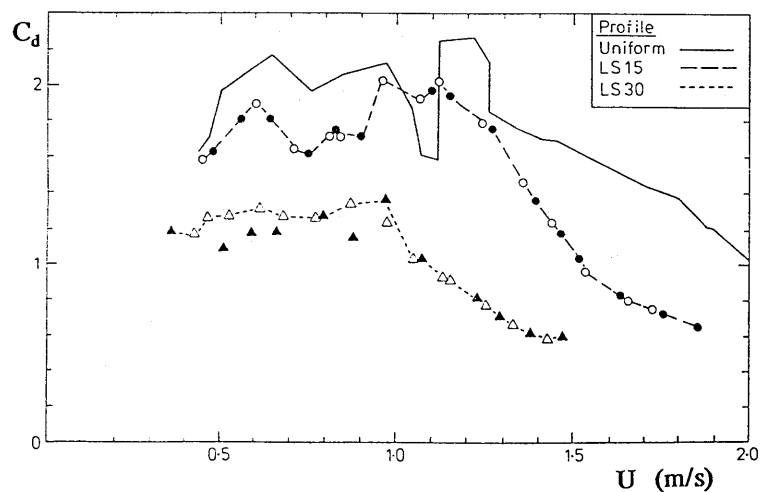


Fig. 6.7.6 Drag coefficient vs velocity.
(Linear shear velocity profiles, no device fitted)

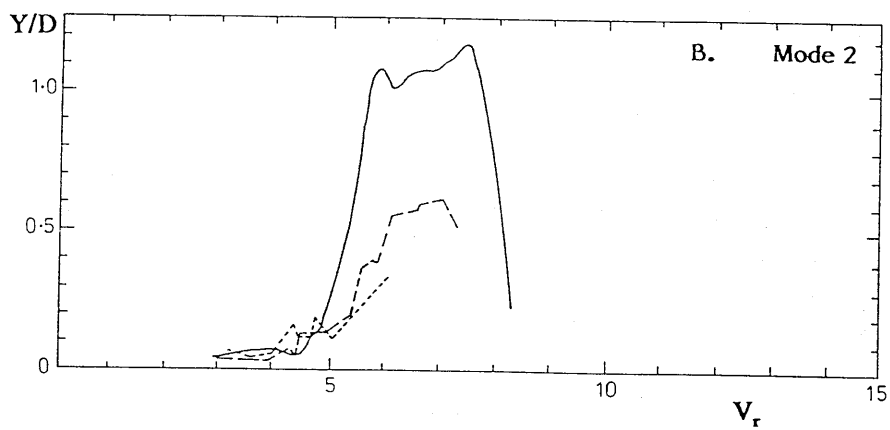
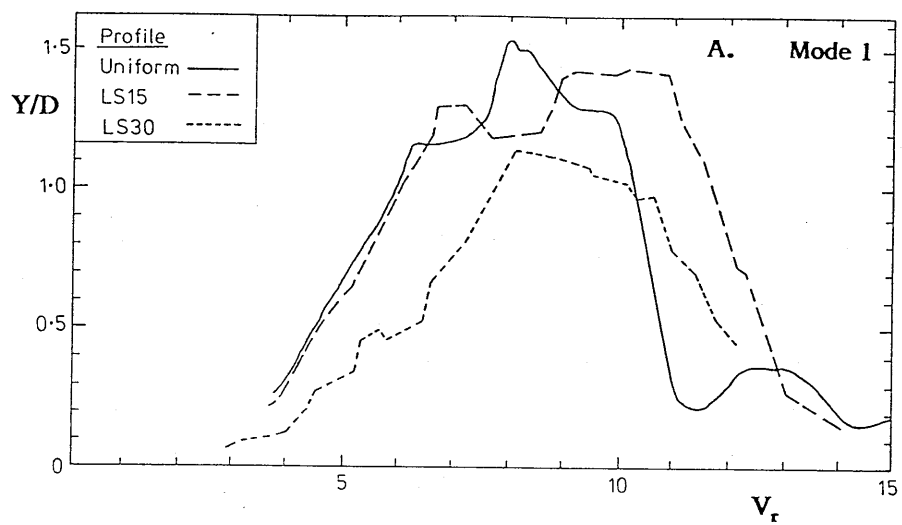


Fig. 6.7.7 Reduced Amplitude vs Reduced Velocity.
(Linear shear velocity profiles, no device fitted)

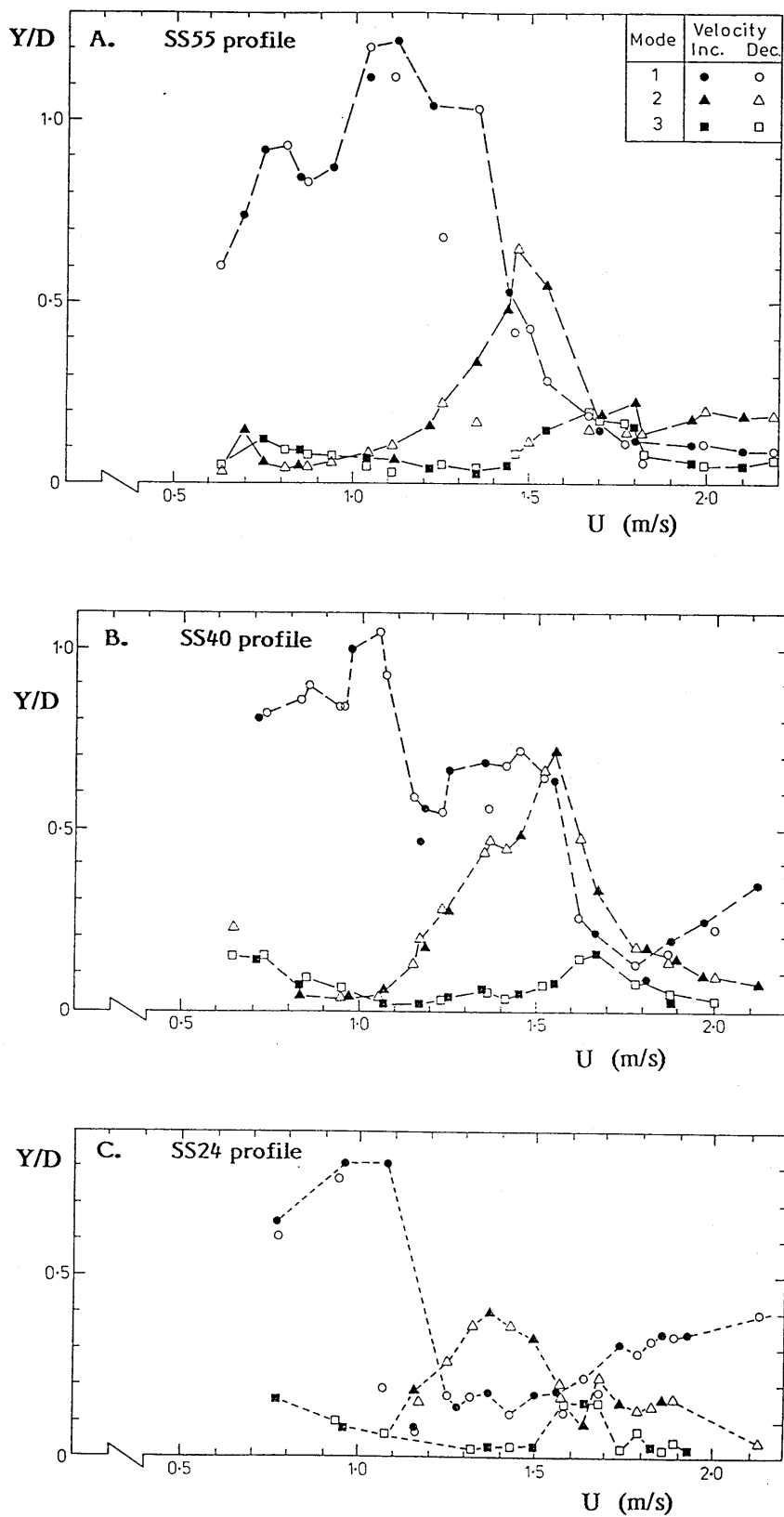


Fig. 6.7.8 Reduced Amplitude vs velocity.
(Stepped shear velocity profiles, no device fitted)

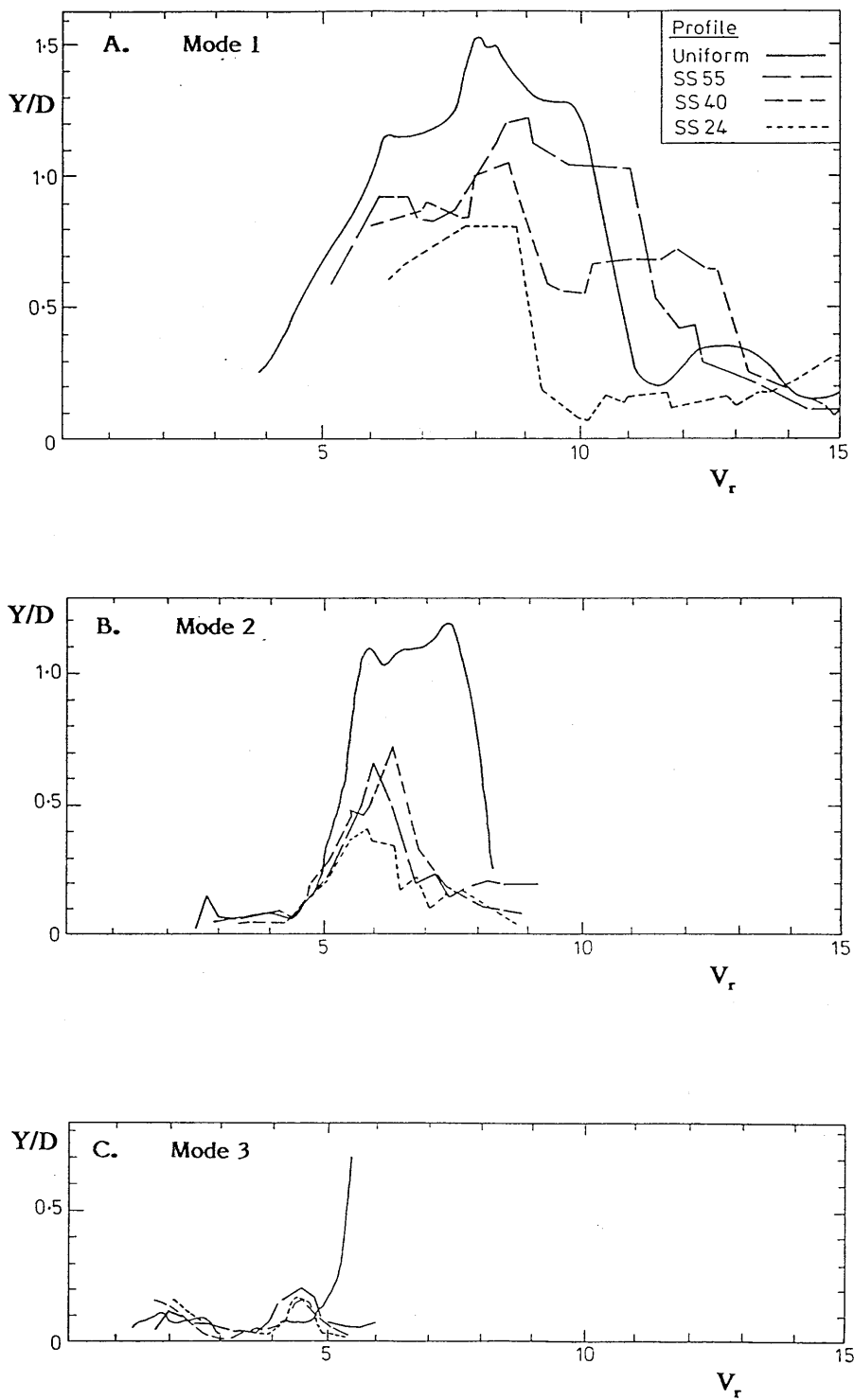


Fig. 6.7.9 Reduced Amplitude vs Reduced Velocity.
(Stepped shear velocity profiles, no device fitted)

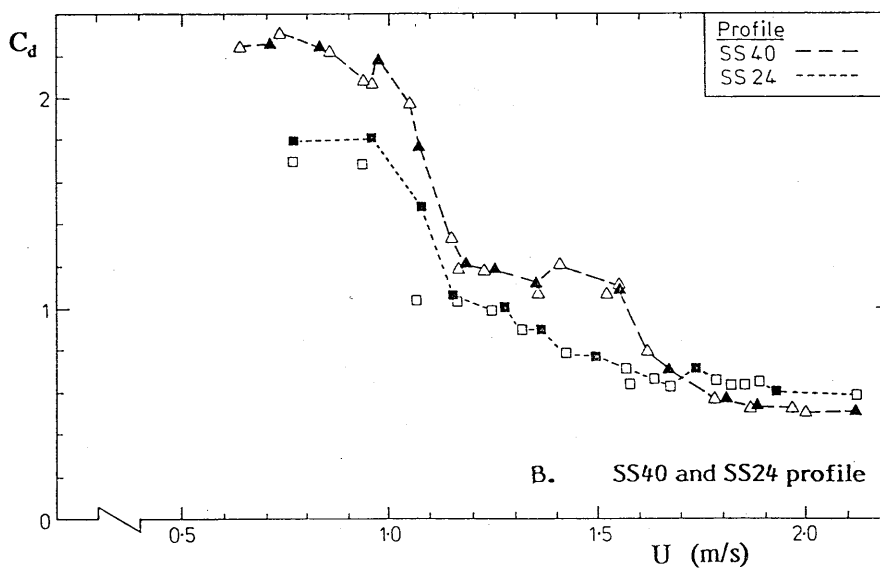
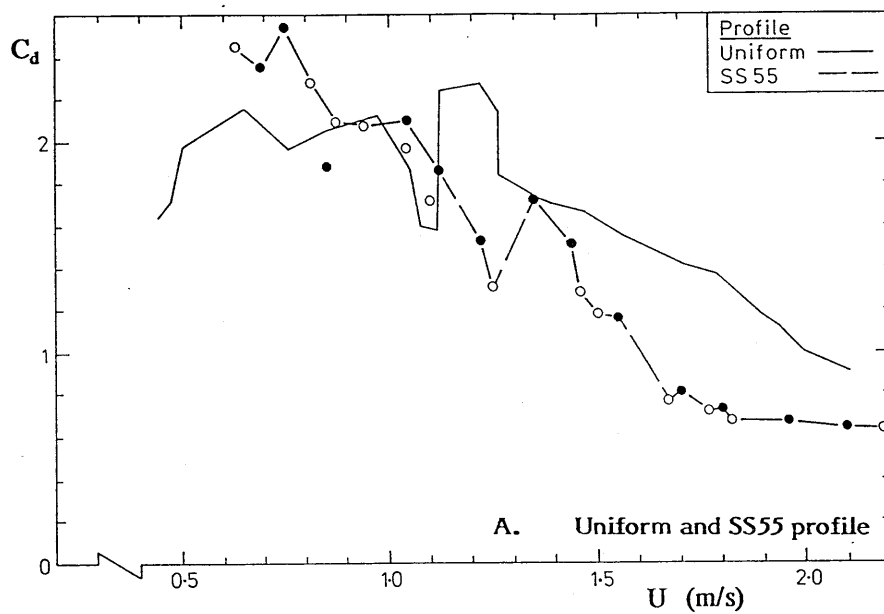


Fig. 6.7.10 Drag coefficient vs velocity.
(Stepped shear velocity profiles, no device fitted)

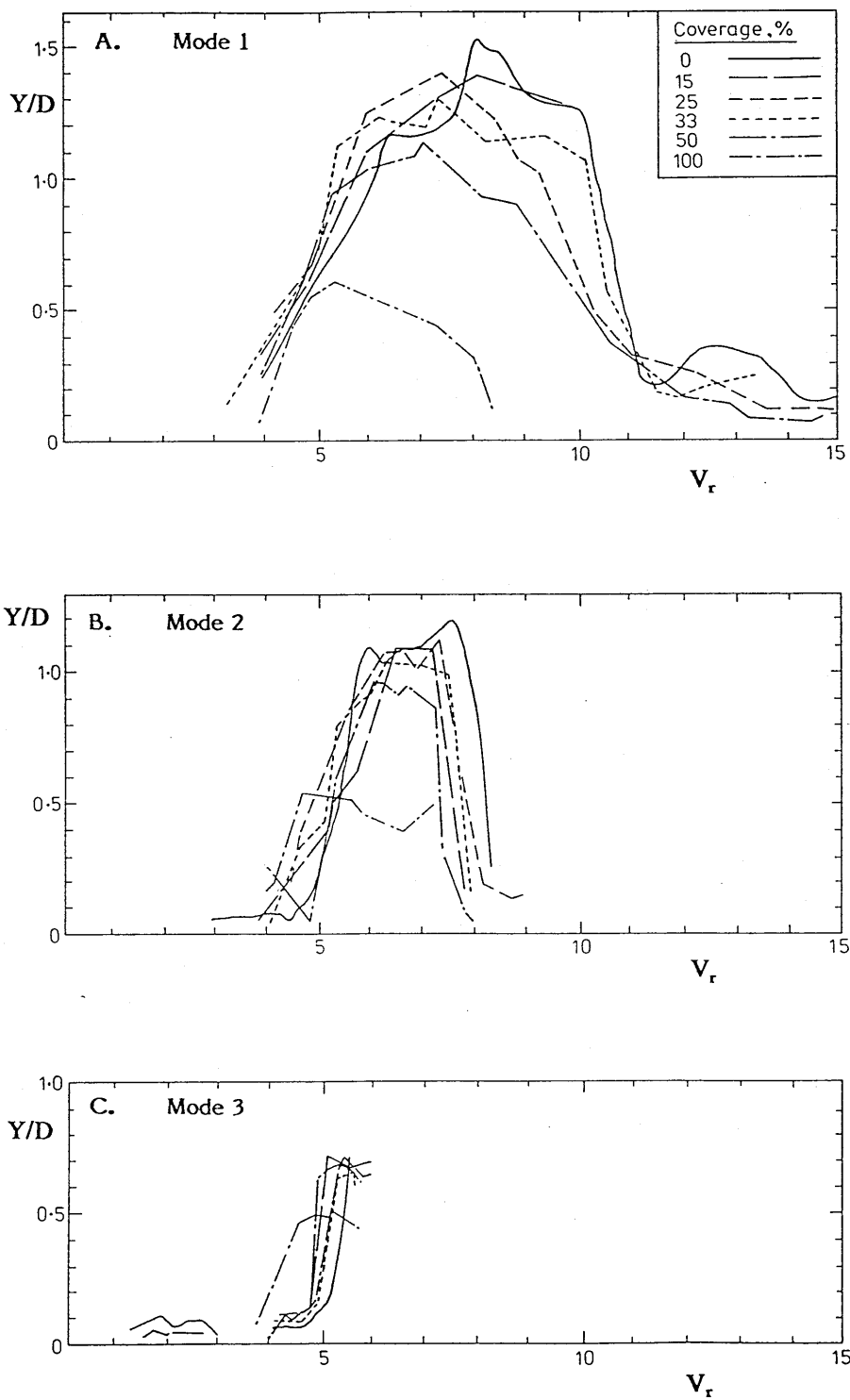


Fig. 6.7.11 Reduced Amplitude vs Reduced Velocity.
(Uniform velocity profile, shroud fitted)

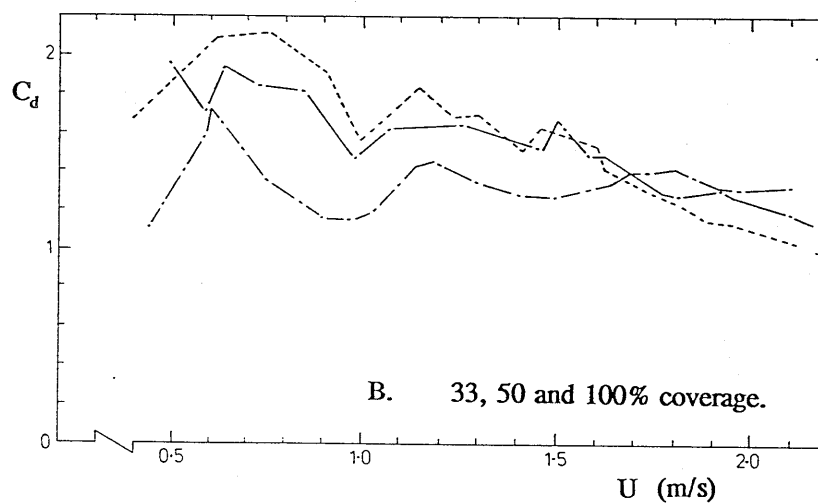
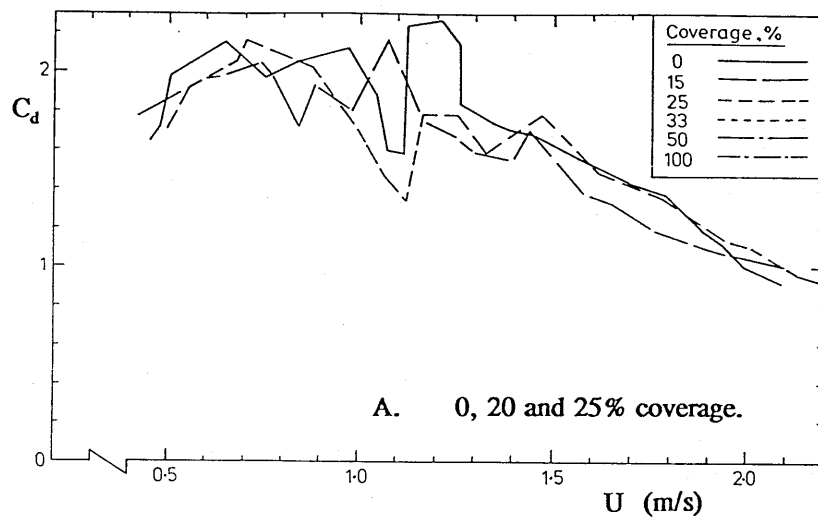


Fig. 6.7.12 Drag coefficient vs velocity.
(Uniform velocity profile, shroud fitted)

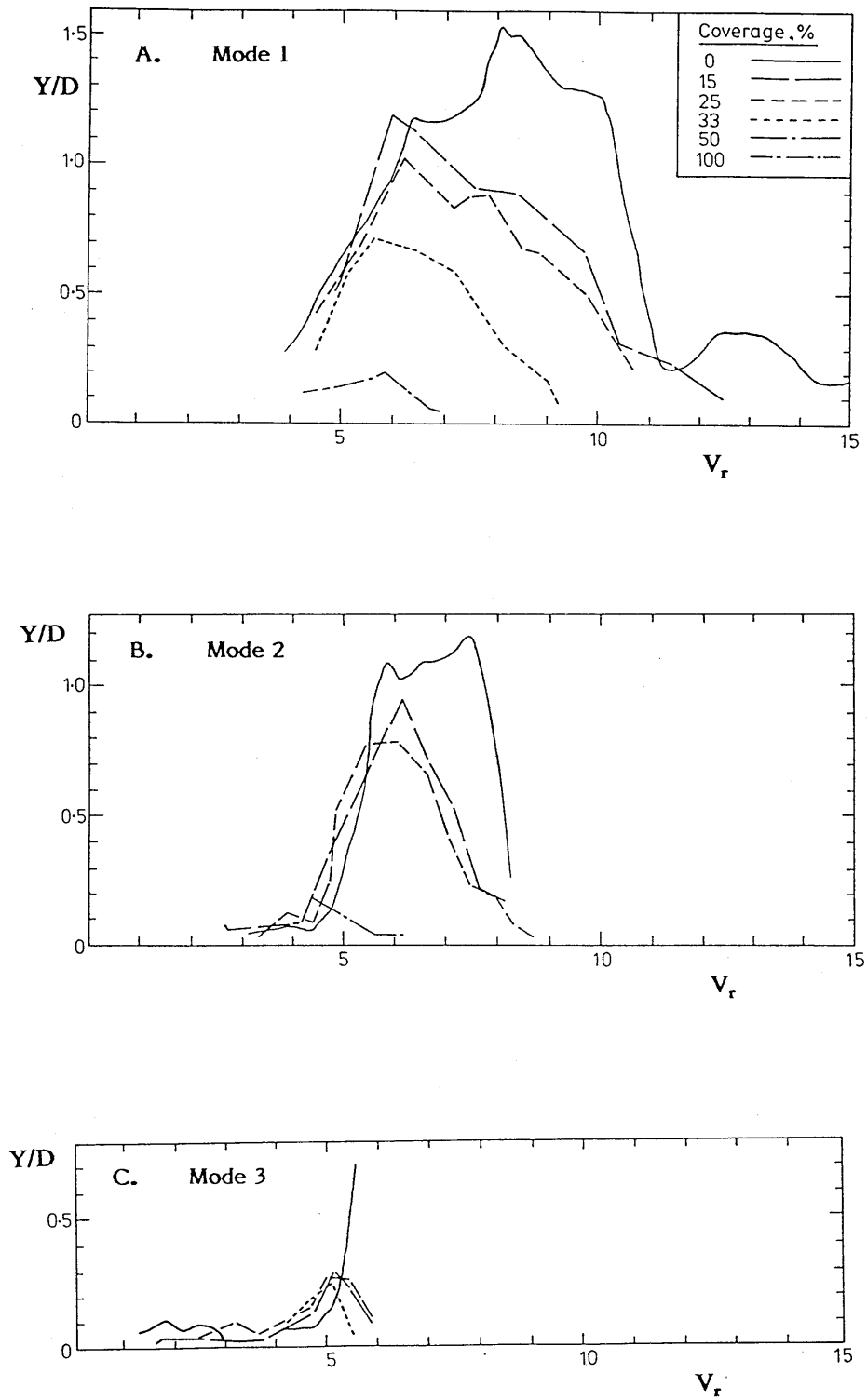


Fig. 6.7.13 Reduced Amplitude vs Reduced Velocity.
(Uniform velocity profile, fairing fitted)

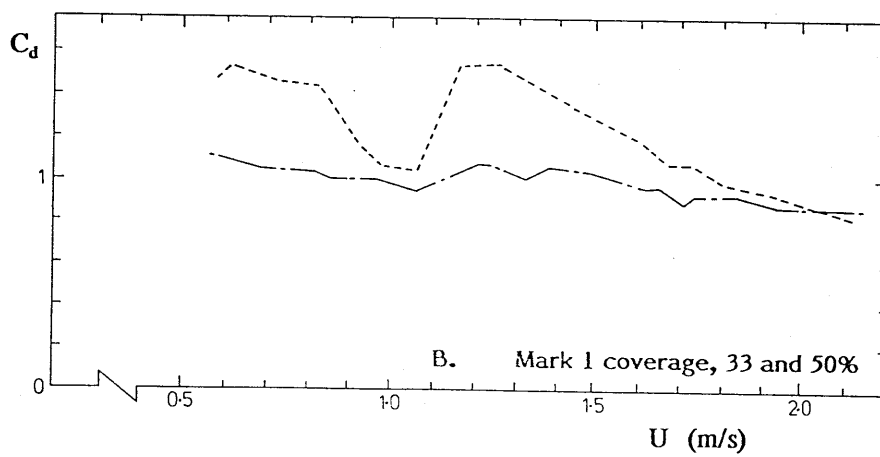
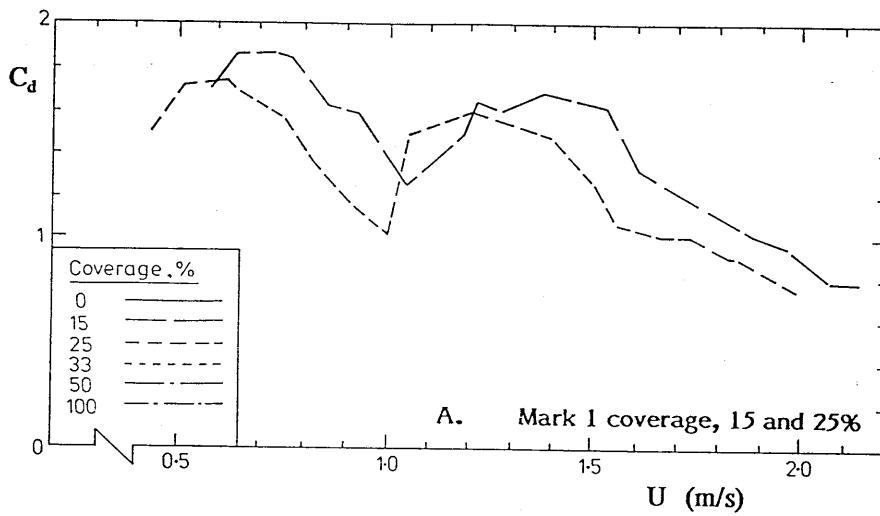


Fig. 6.7.14 Drag coefficient vs velocity.
(Uniform velocity profile, fairing fitted)

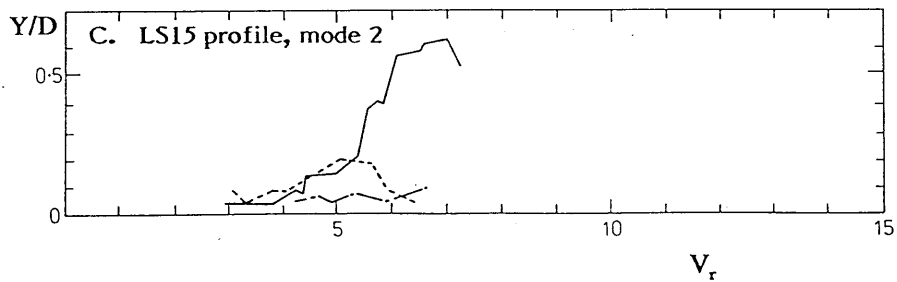
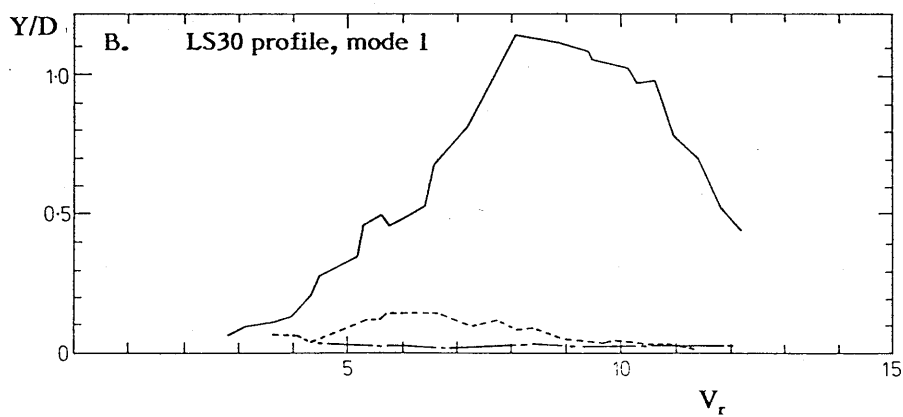
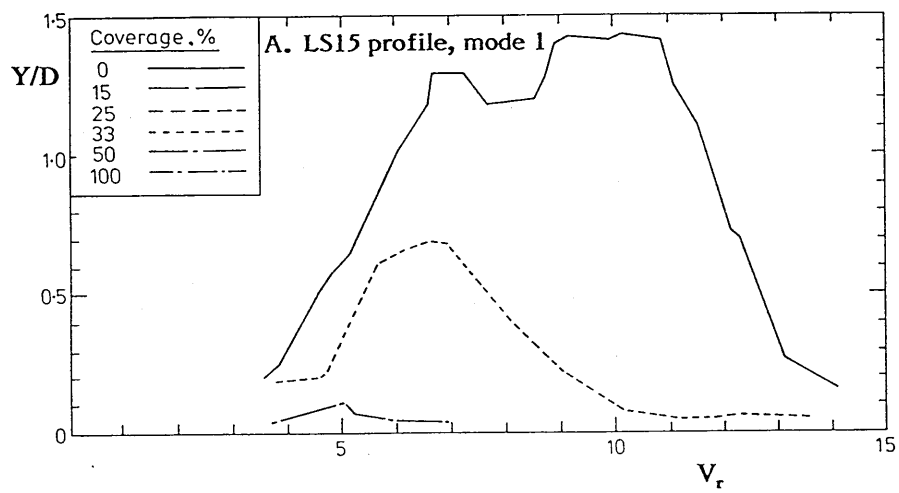


Fig. 6.7.15 Reduced Amplitude vs Reduced Velocity.
(Linear shear velocity profiles, fairing fitted)

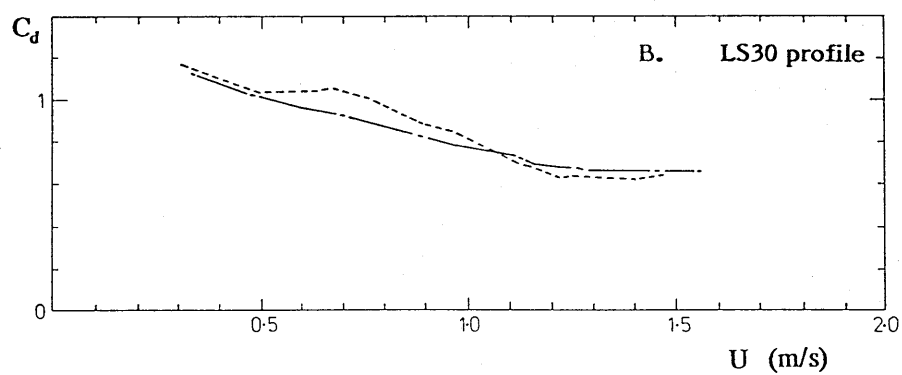
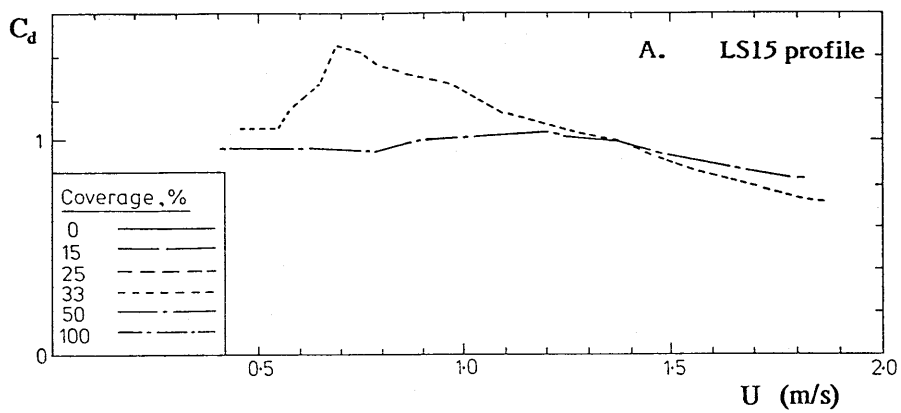


Fig. 6.7.16 Drag coefficient vs velocity.
(Linear shear velocity profiles, fairing fitted)

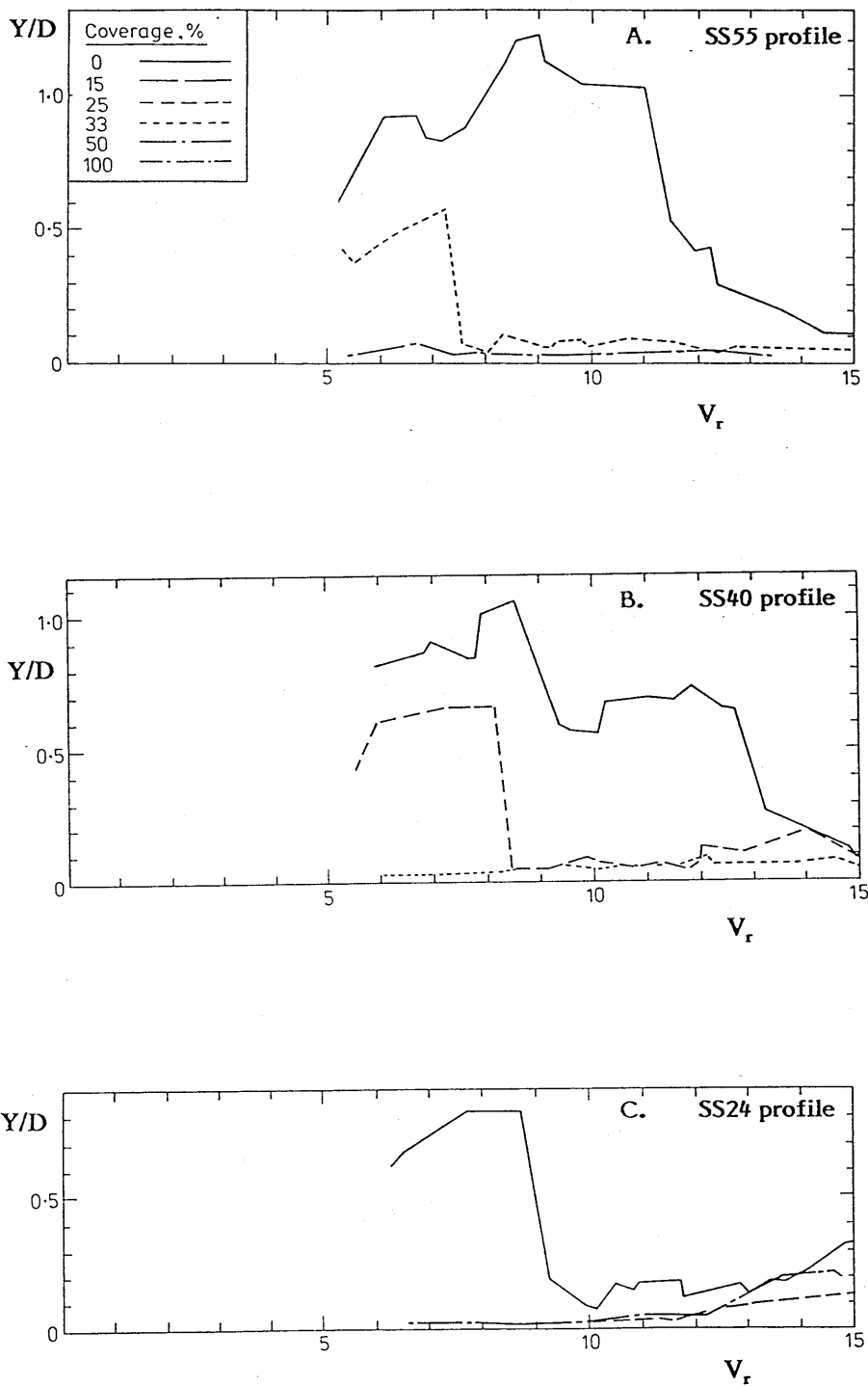


Fig. 6.7.17 Reduced Amplitude vs Reduced Velocity.
(Stepped shear velocity profiles, fairing fitted, Mode 1 only)

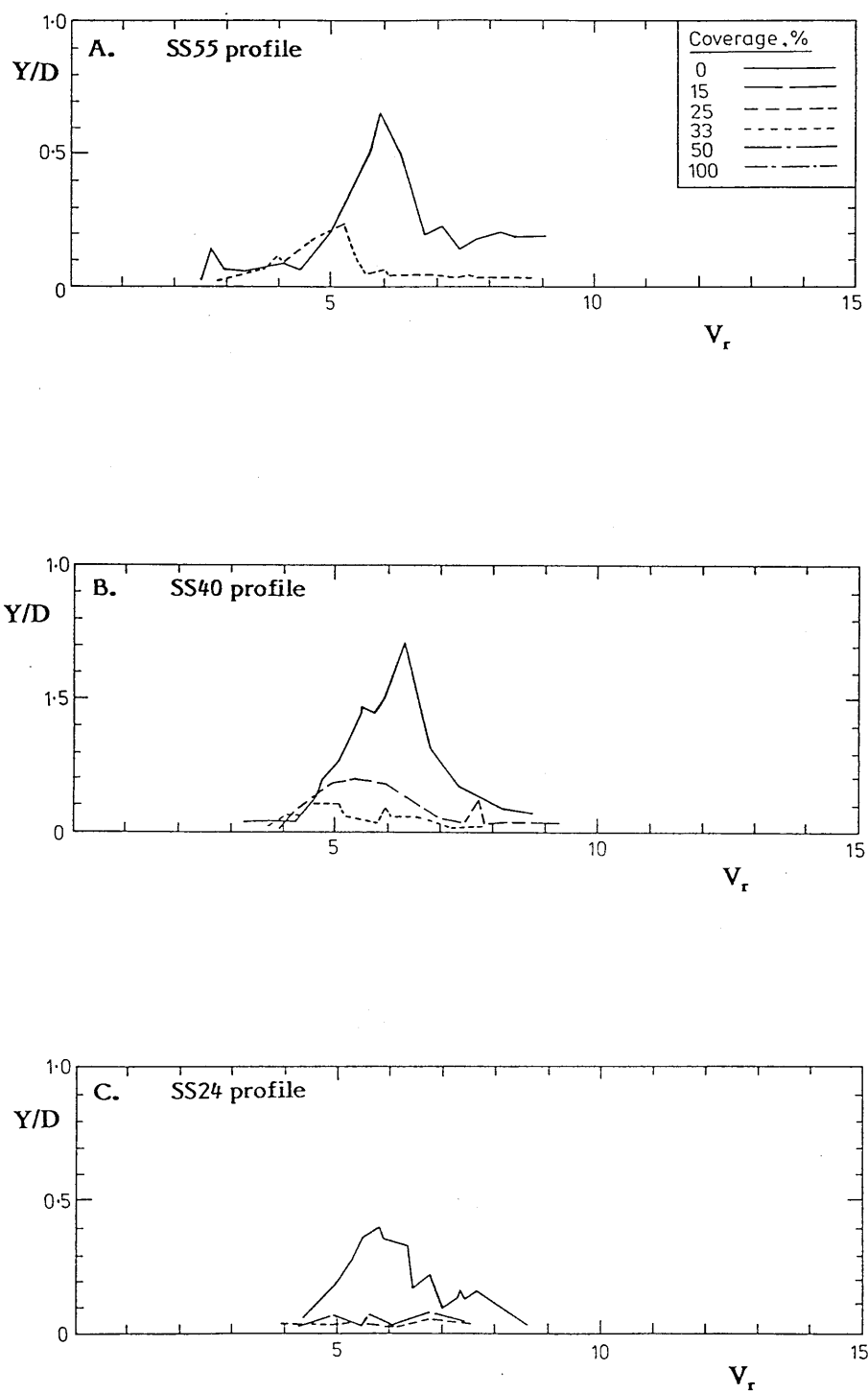


Fig. 6.7.18 Reduced Amplitude vs Reduced Velocity.
(Stepped shear velocity profiles, fairing fitted, Mode 2 only)

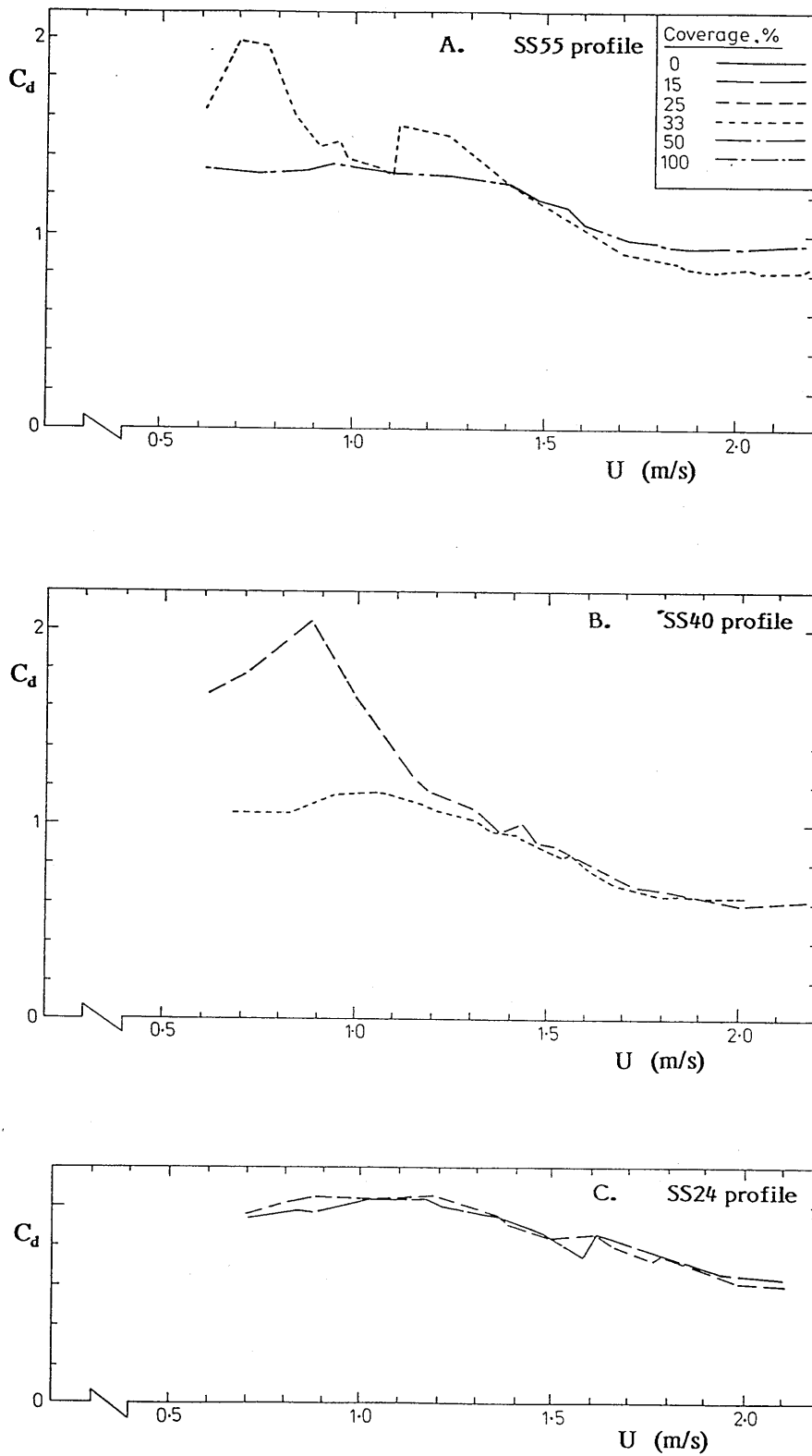


Fig. 6.7.19 Drag coefficient vs velocity.
(Stepped shear velocity profiles, fairing fitted)

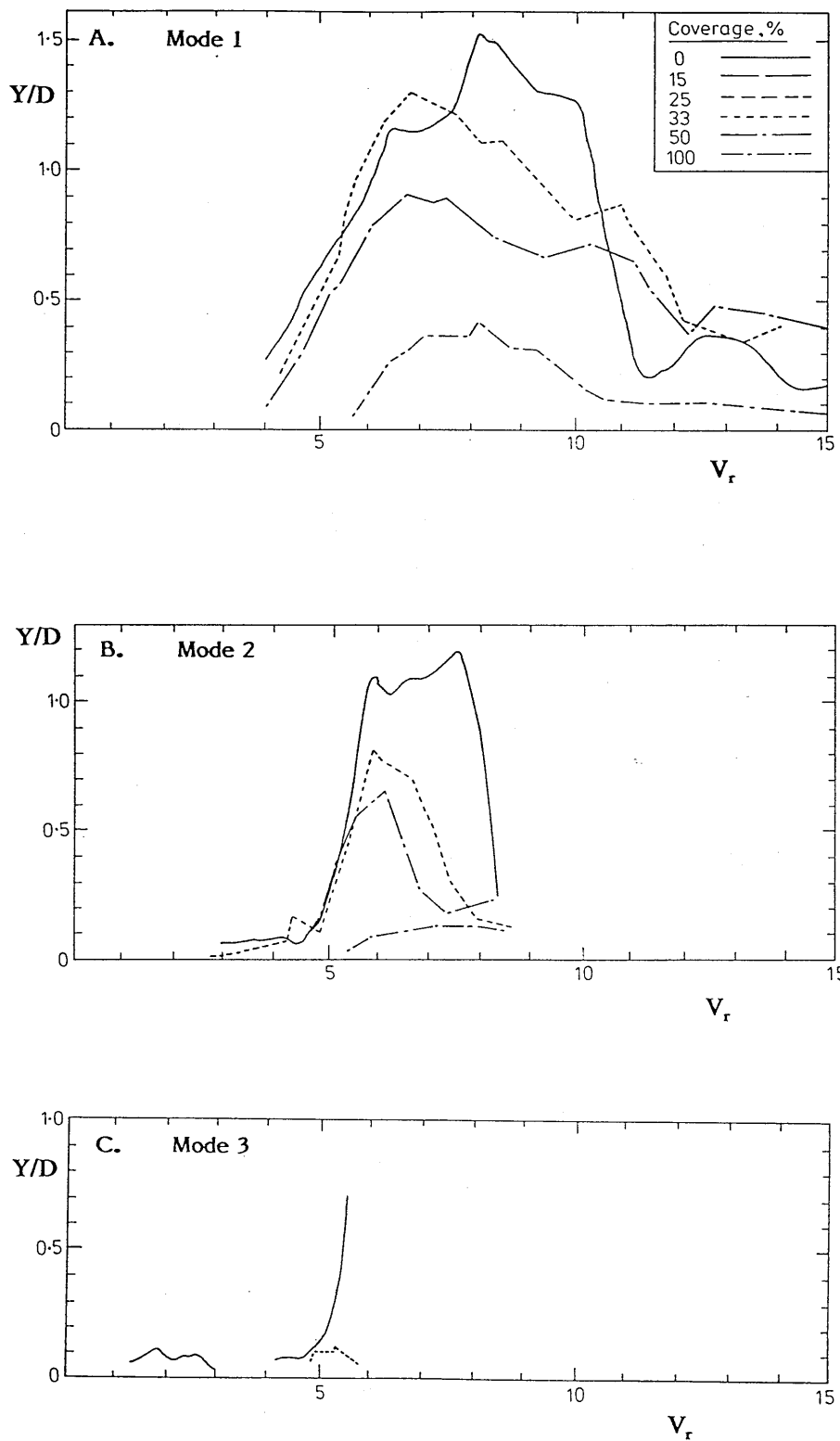


Fig. 6.7.20 Reduced Amplitude vs Reduced Velocity.
(Uniform velocity profile, strake fitted)

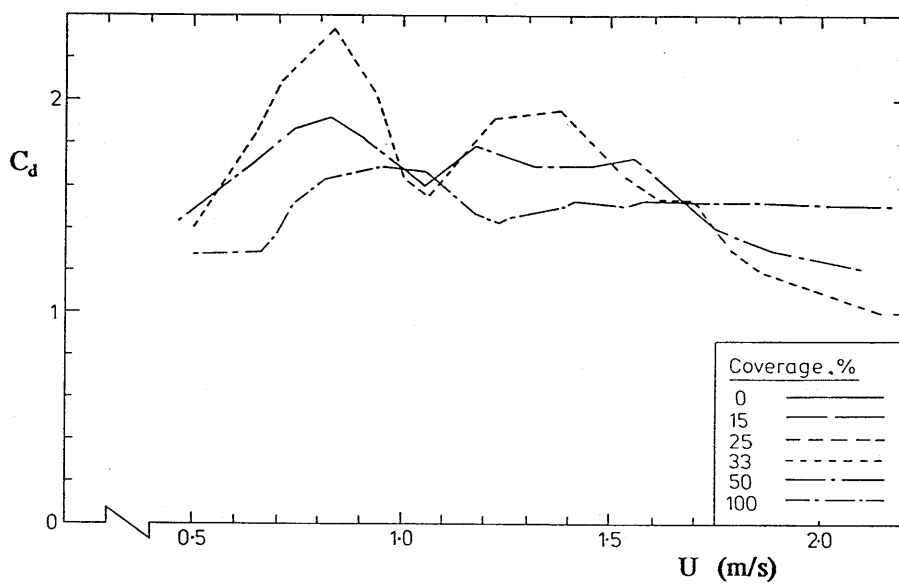


Fig. 6.7.21 Drag coefficient vs velocity.
(Uniform velocity profile, strake fitted)

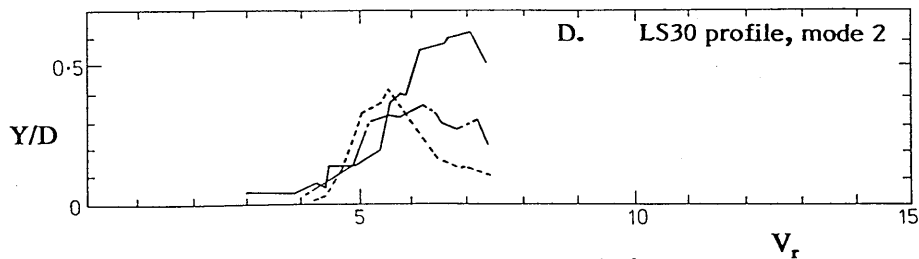
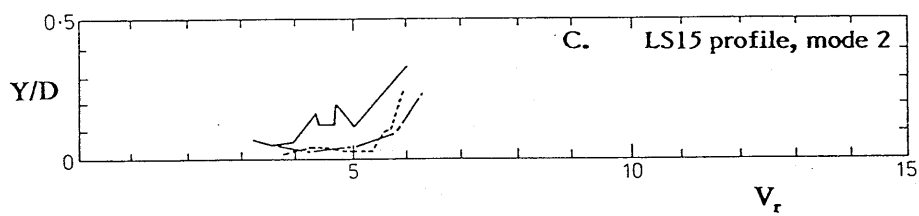
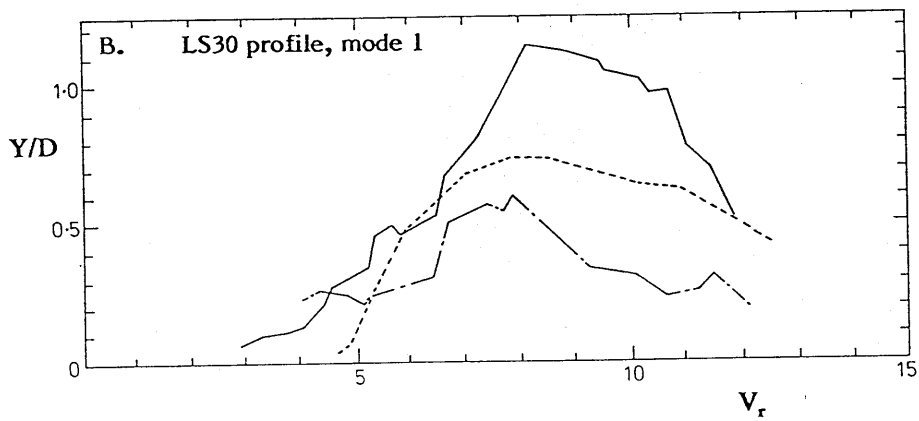
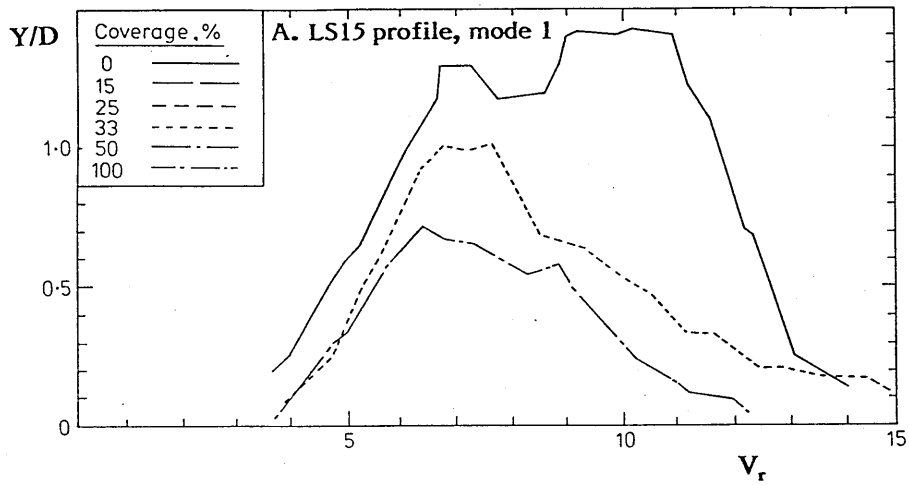


Fig. 6.7.22 Reduced Amplitude vs Reduced Velocity.
(Linear shear velocity profiles, strake fitted)

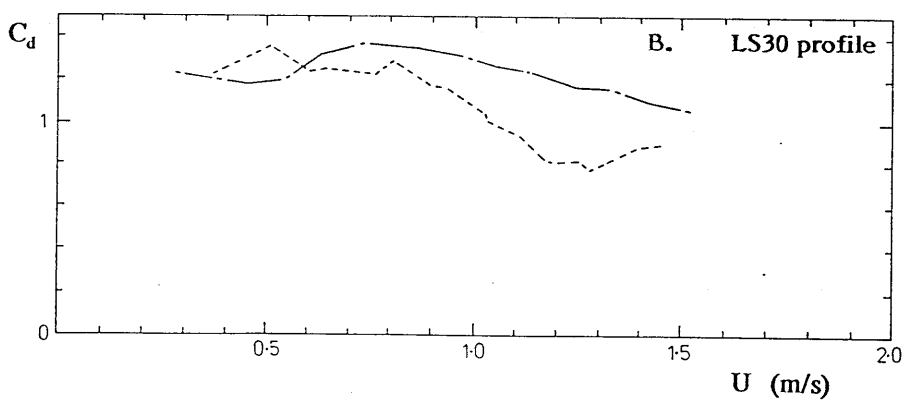
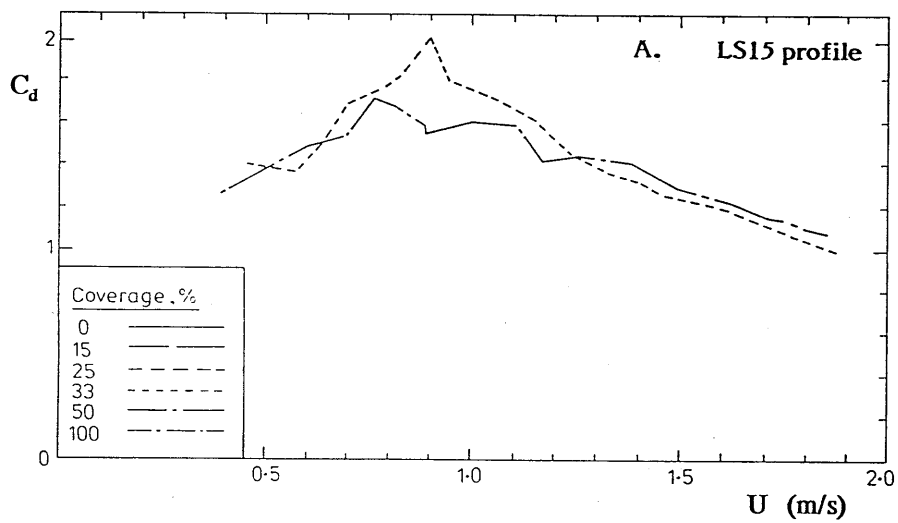


Fig. 6.7.23 Drag coefficient vs velocity.
(Linear shear velocity profiles, strake fitted)

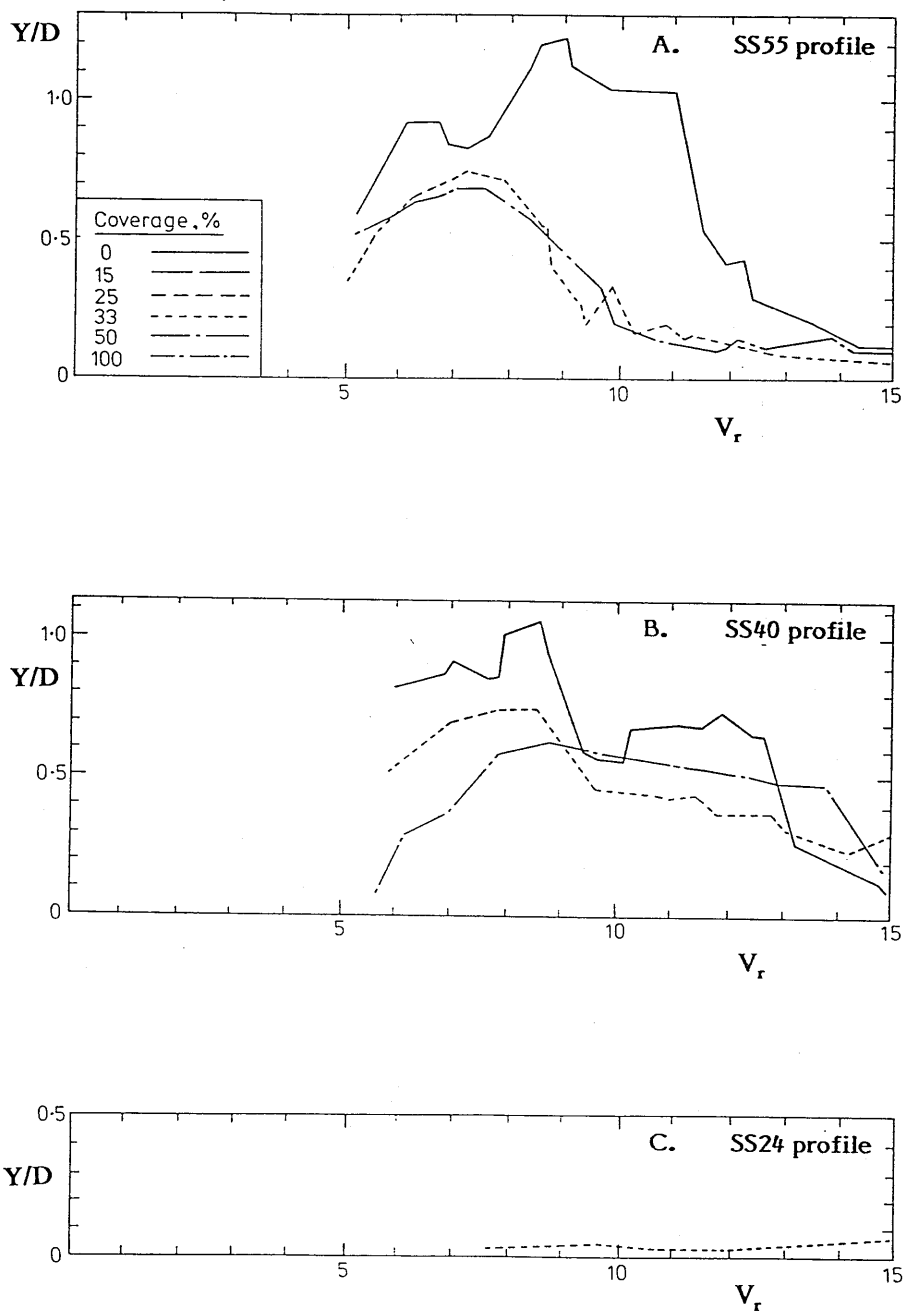


Fig. 6.7.24 Reduced Amplitude vs Reduced Velocity.
 (Stepped shear velocity profiles, strake fitted, Mode 1 only)

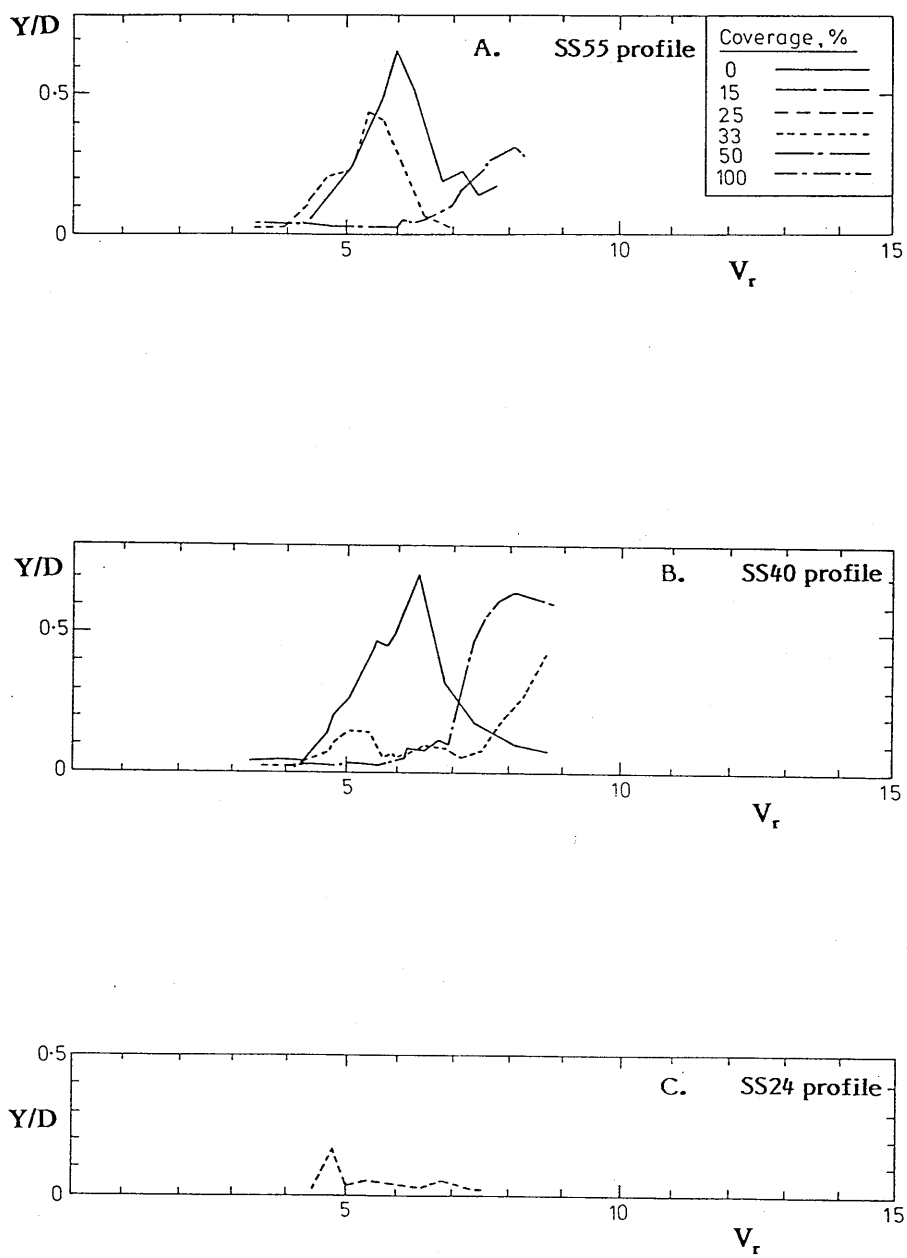


Fig. 6.7.25 Reduced Amplitude vs Reduced Velocity.
(Stepped shear velocity profiles, strake fitted, Mode 2 only)

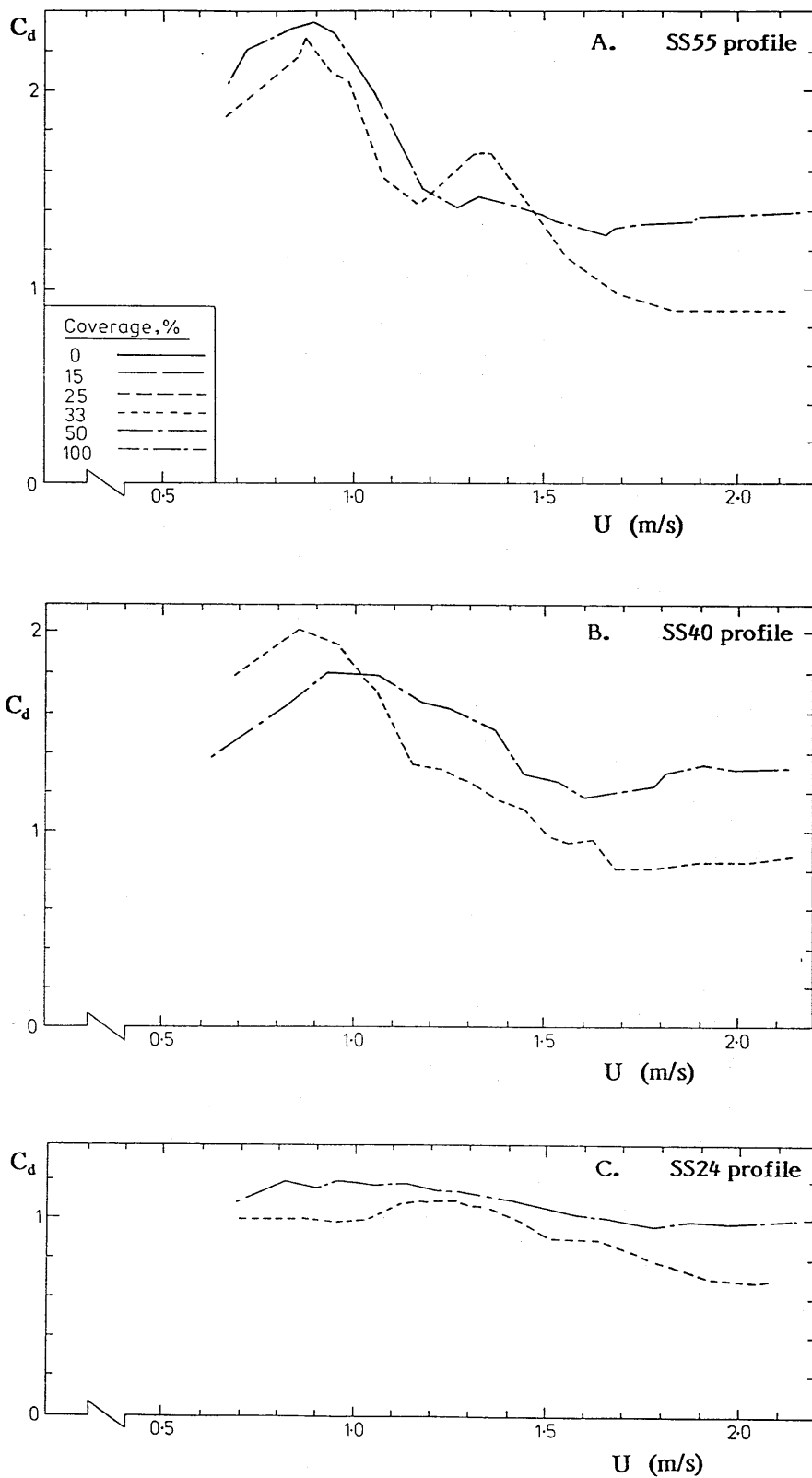


Fig. 6.7.26 Drag coefficient vs velocity.
(Stepped shear velocity profiles, strake fitted)

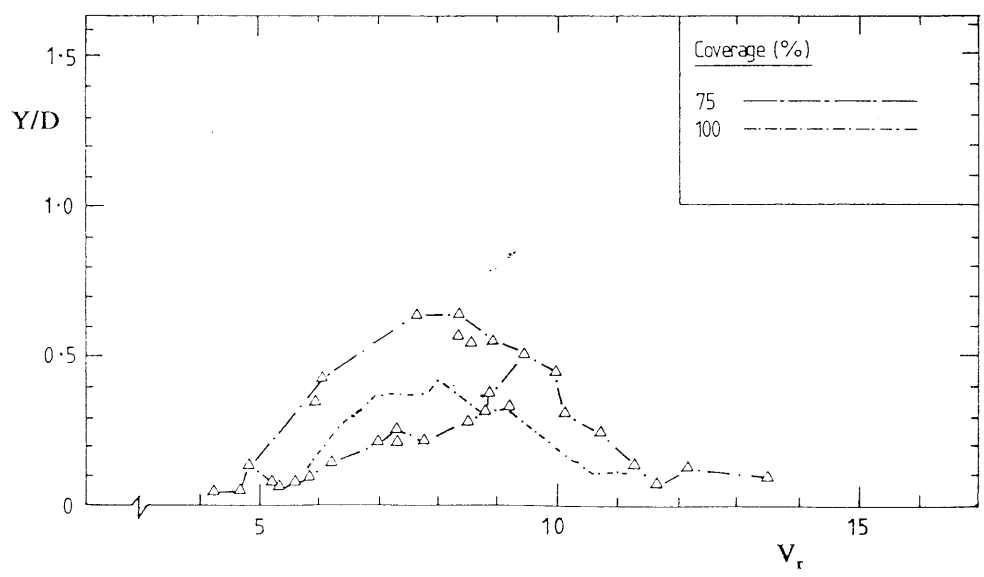
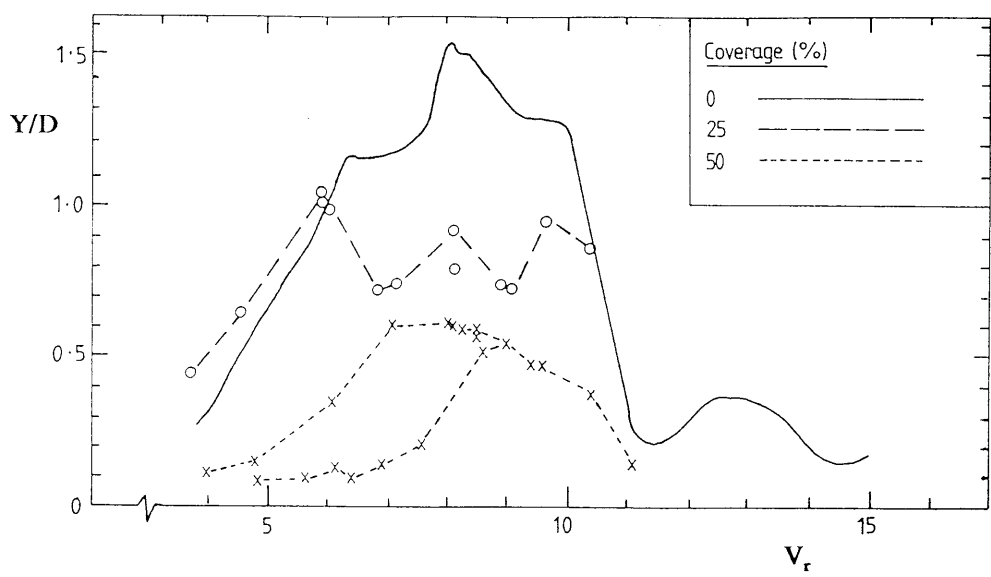


Fig. 6.7.27 Reduced Amplitude vs Reduced Velocity.
(Uniform velocity profile, stake fitted centrally)

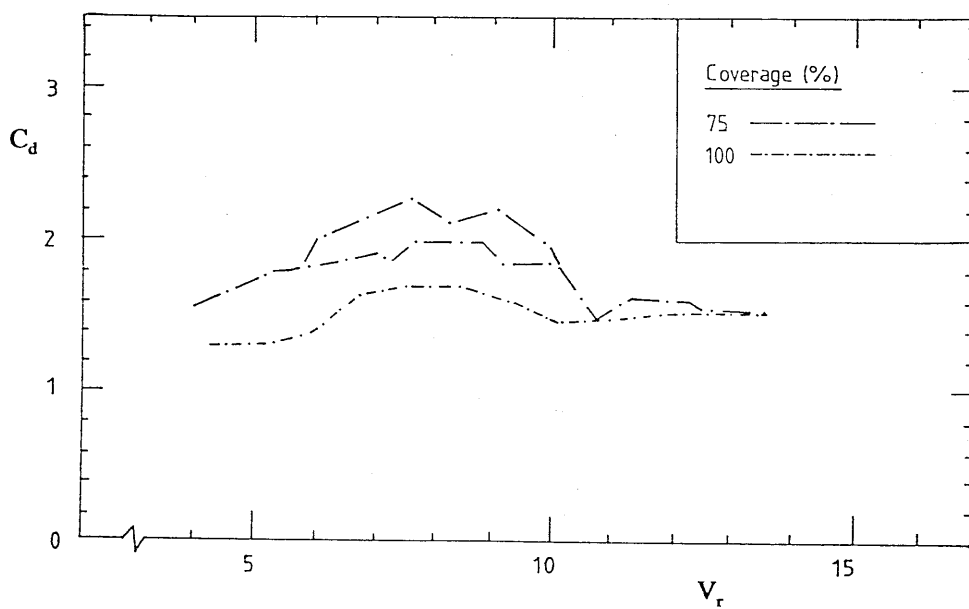
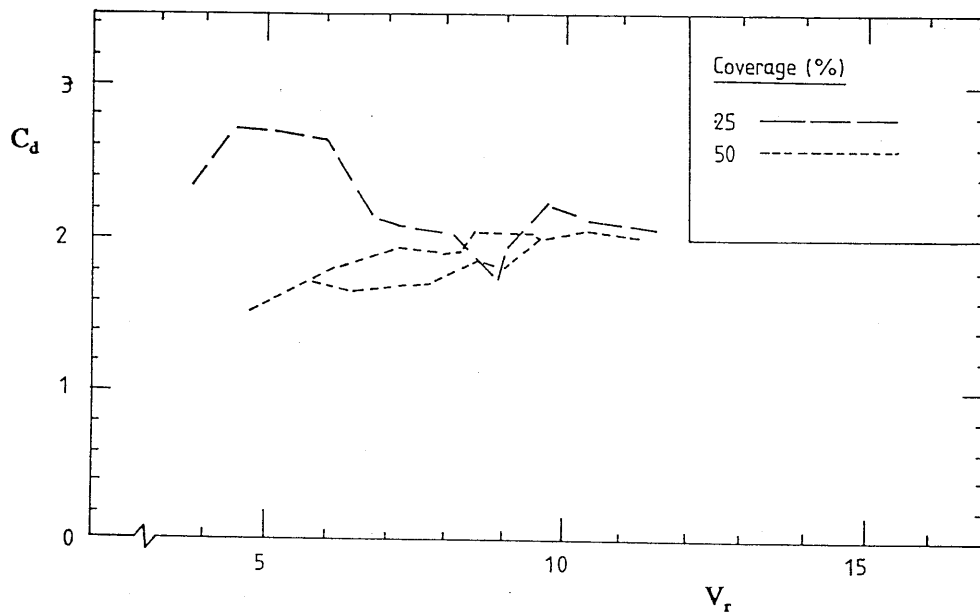


Fig. 6.7.28 Drag coefficient vs. Reduced Velocity.
(Uniform velocity profile, strake fitted centrally)

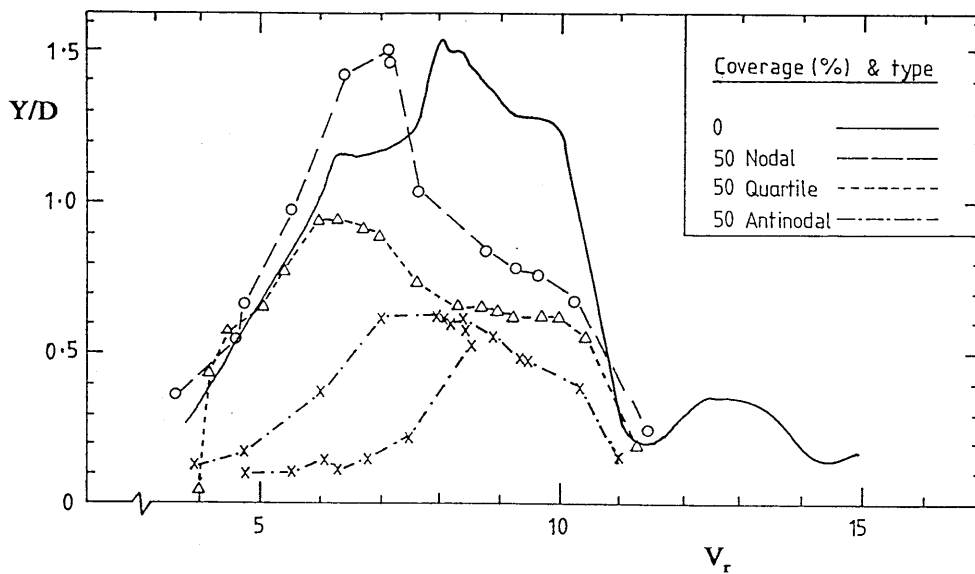


Fig. 6.7.29 Reduced Amplitude vs Reduced Velocity.
(Uniform velocity profile, 50% stake fitted in various positions)

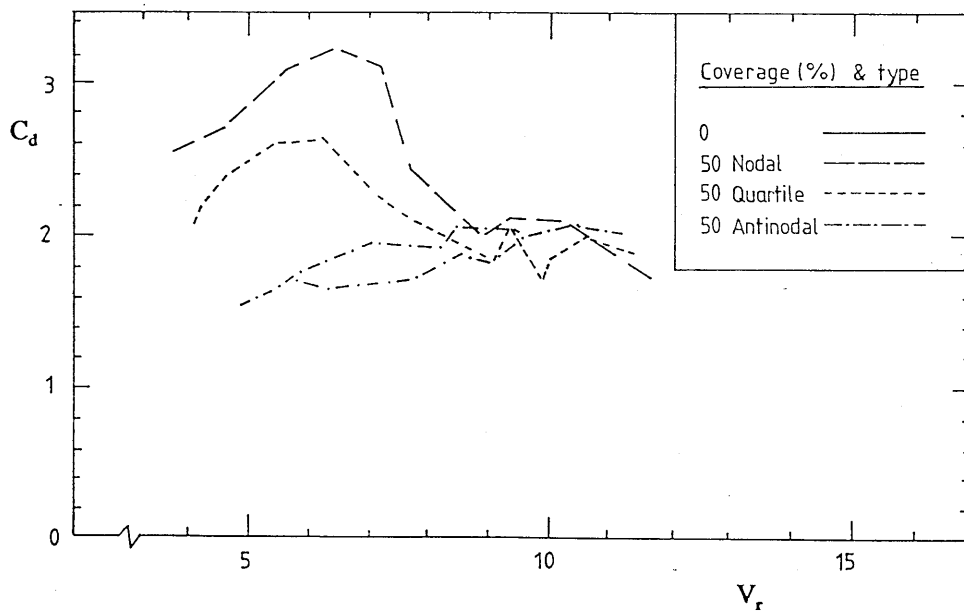


Fig. 6.7.30 Drag coefficient vs. Reduced Velocity.
(Uniform velocity profile, 50% stake fitted in various positions)

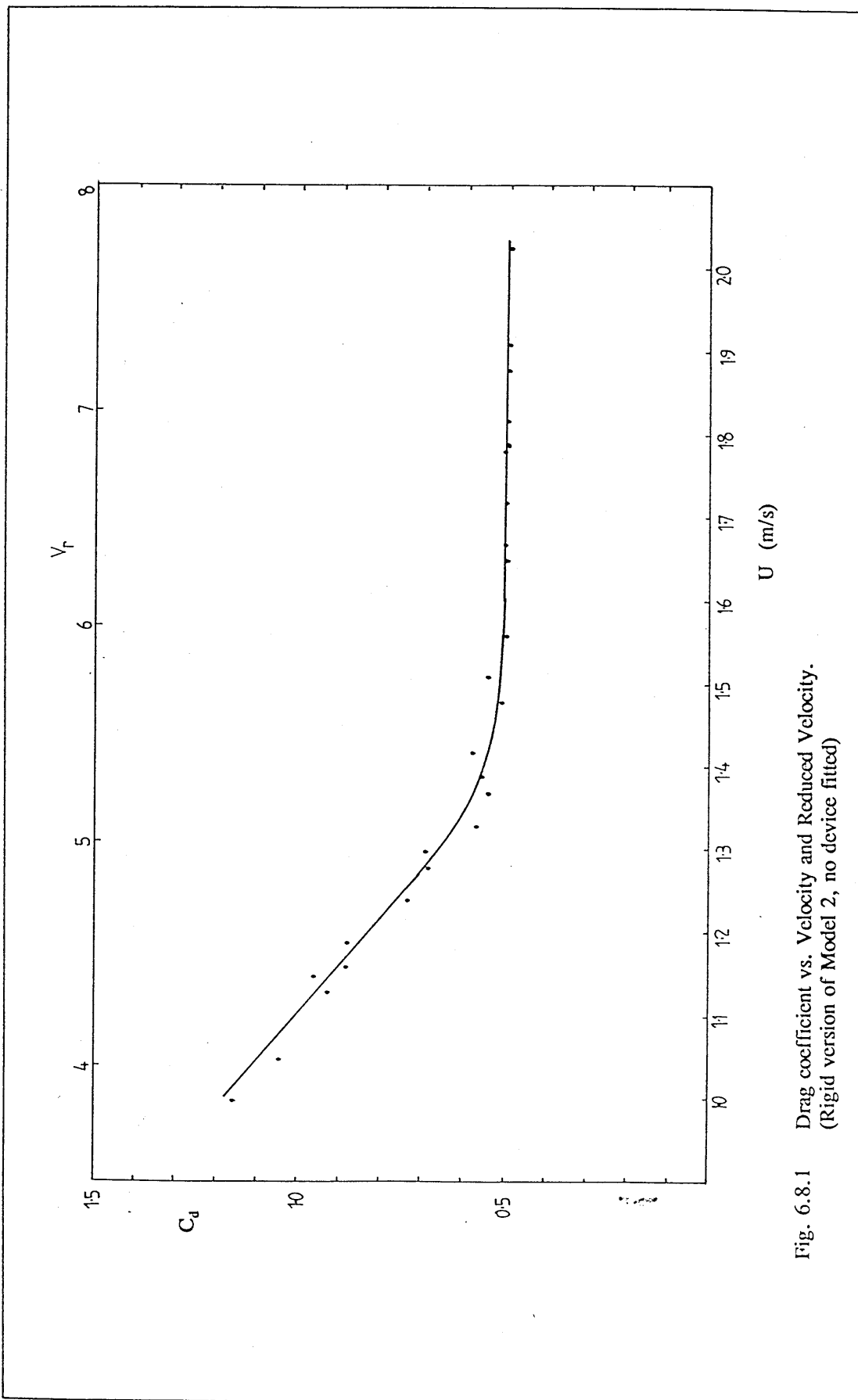


Fig. 6.8.1 Drag coefficient vs. Velocity and Reduced Velocity.
(Rigid version of Model 2, no device fitted)

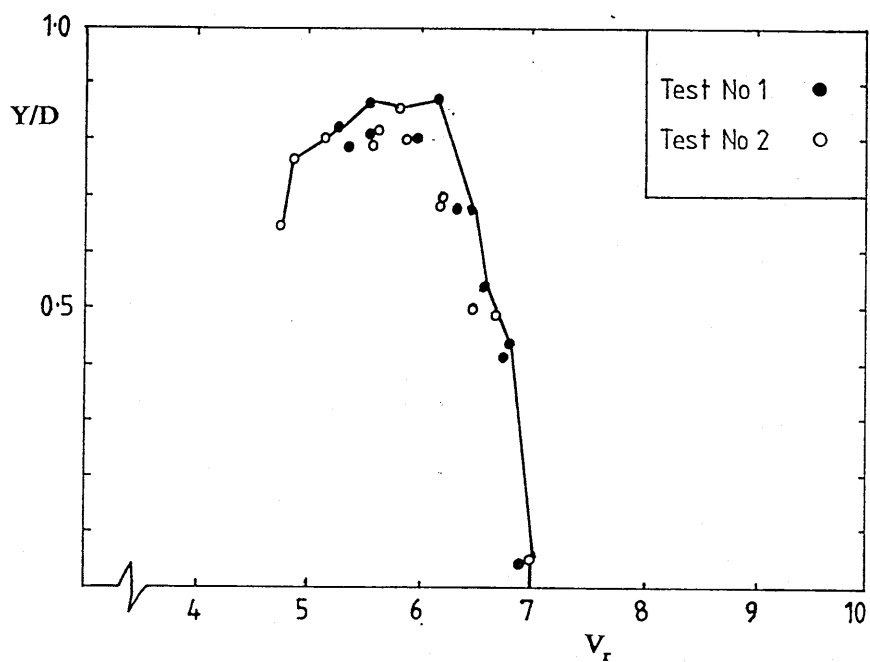


Fig. 6.8.2 Reduced Amplitude vs Reduced Velocity.
(Flexible version of Model 2, no device fitted)

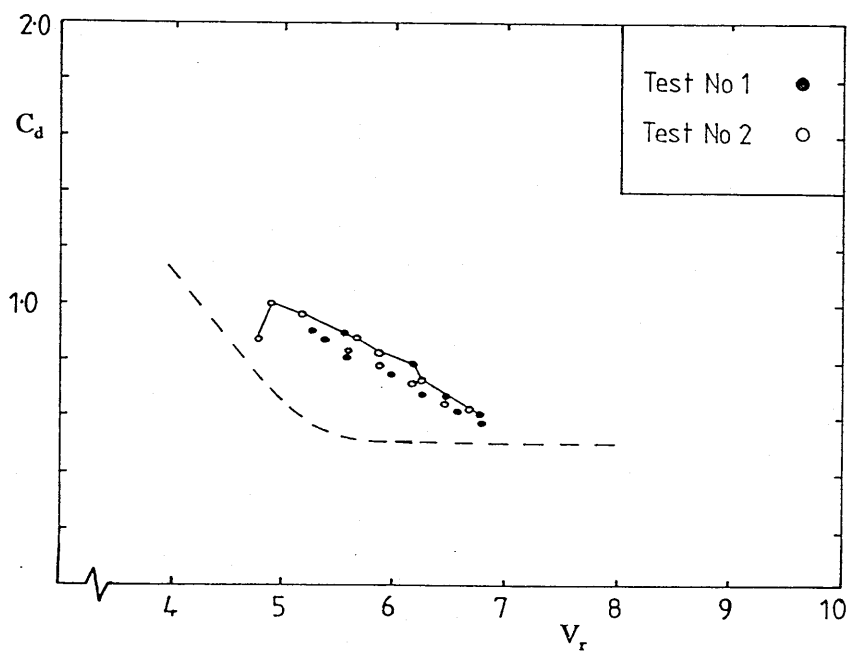


Fig. 6.8.3 Drag coefficient vs. Reduced Velocity.
(Flexible version of Model 2, no device fitted)

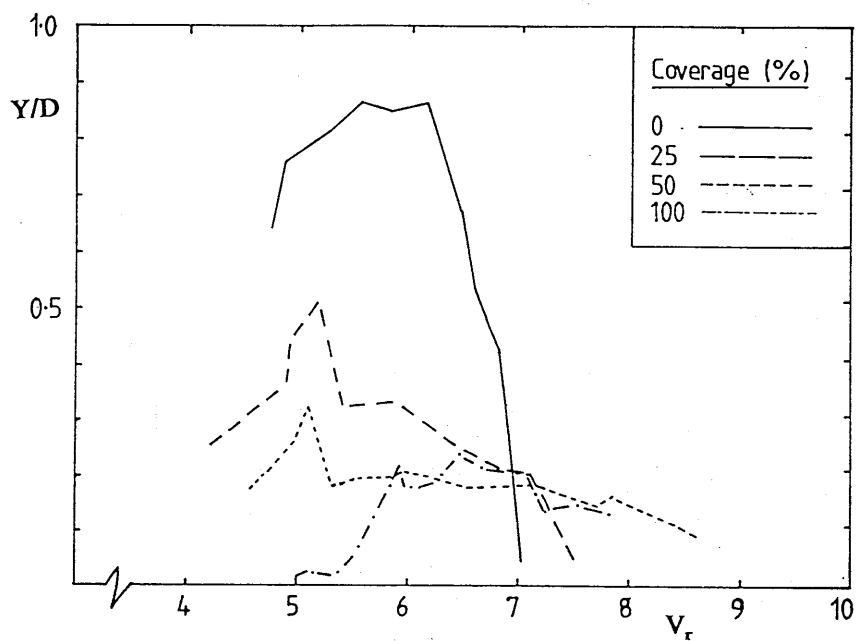


Fig. 6.8.4 Reduced Amplitude vs Reduced Velocity.
(Flexible version of Model 2, commercial strake with 4.5 D pitch fitted)

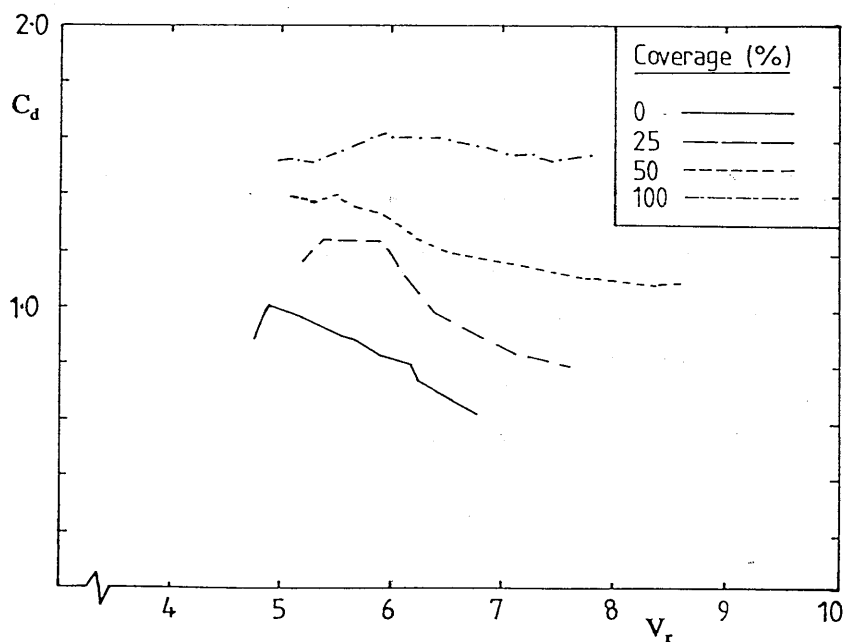


Fig. 6.8.5 Drag coefficient vs. Reduced Velocity.
(Flexible version of Model 2, commercial strake with 4.5 D pitch fitted)

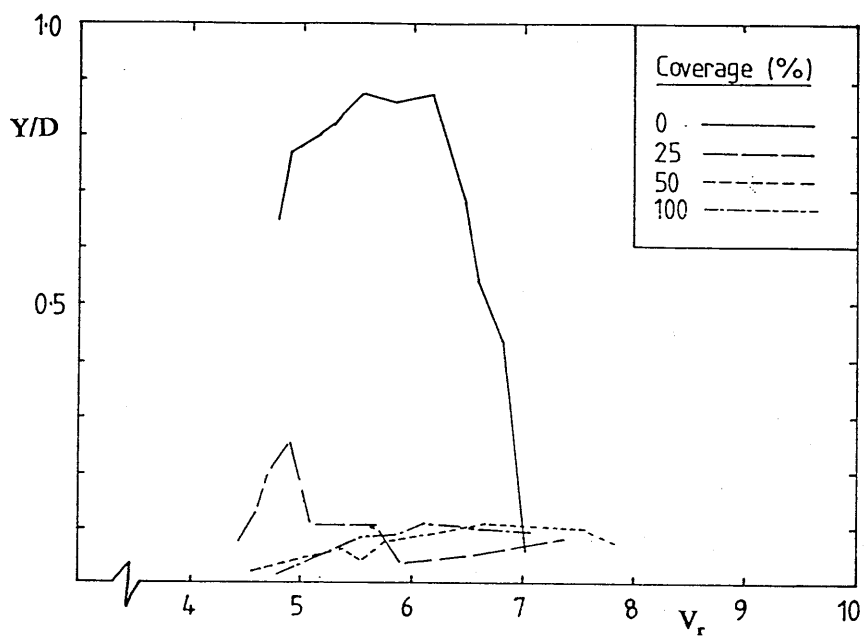


Fig. 6.8.6 Reduced Amplitude vs Reduced Velocity.
(Flexible version of Model 2, commercial strake with 15 D pitch fitted)

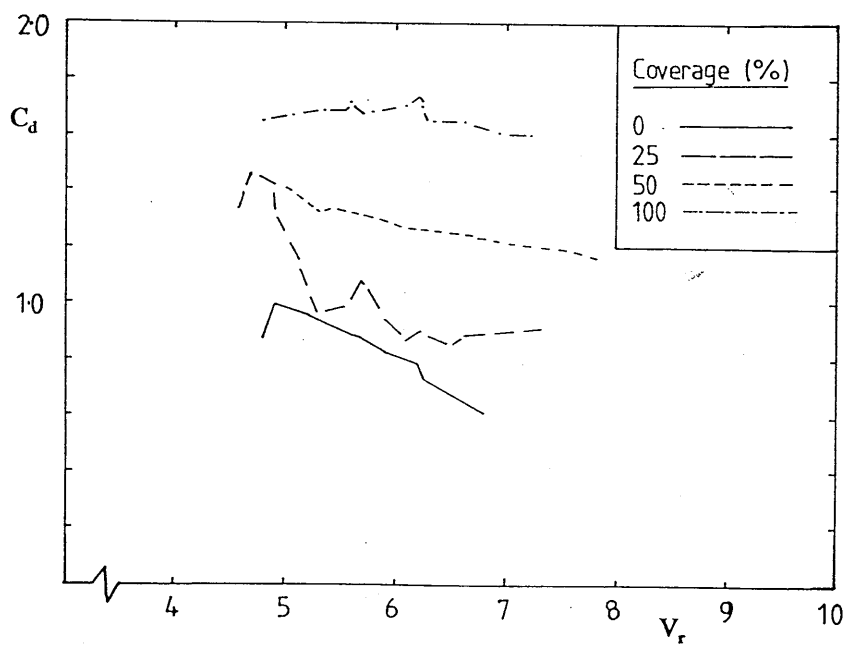


Fig. 6.8.7 Drag coefficient vs. Reduced Velocity.
(Flexible version of Model 2, commercial strake with 15 D pitch fitted)

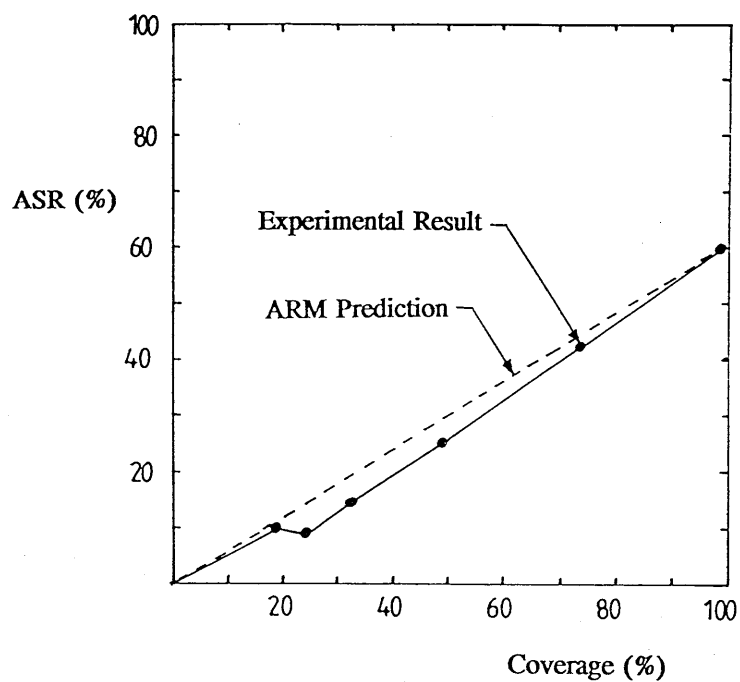


Fig. 7.3.1 Amplitude Suppression Ratio vs. Coverage.
(Uniform velocity profile, Mode 1, Shroud fitted)

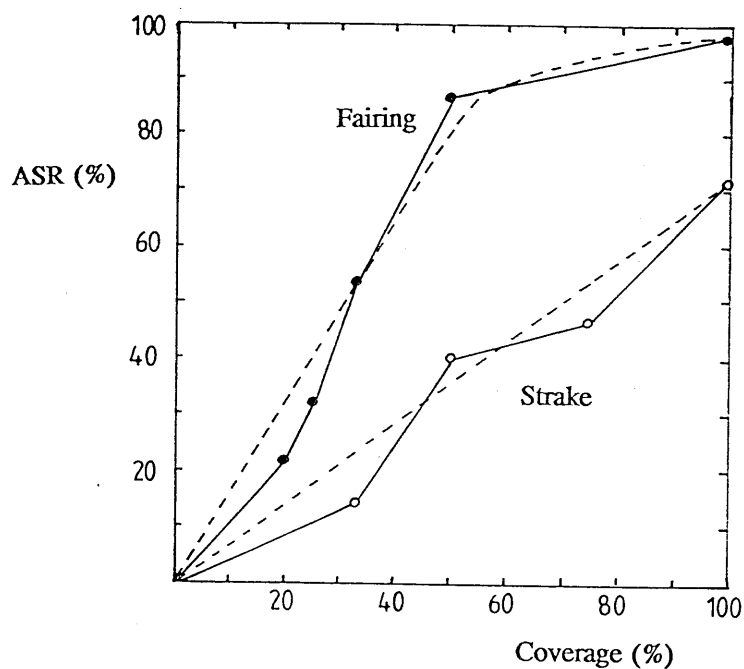


Fig. 7.3.2 Amplitude Suppression Ratio vs. Coverage.
(Uniform velocity profile, Mode 1, Fairing or Strake fitted)

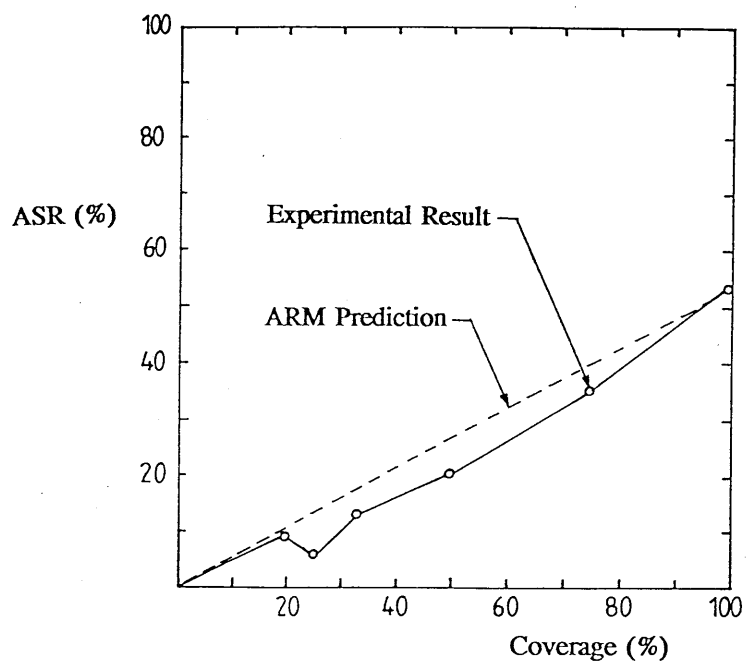


Fig. 7.3.3 Amplitude Suppression Ratio vs. Coverage.
(Uniform velocity profile, Mode 2, Shroud fitted)

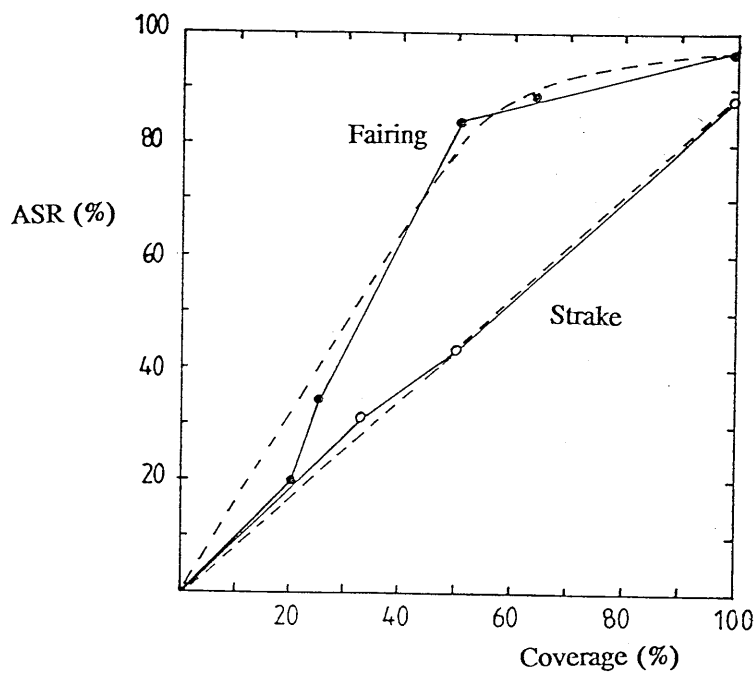


Fig. 7.3.4 Amplitude Suppression Ratio vs. Coverage.
(Uniform velocity profile, Mode 2, Fairing or Strake fitted)

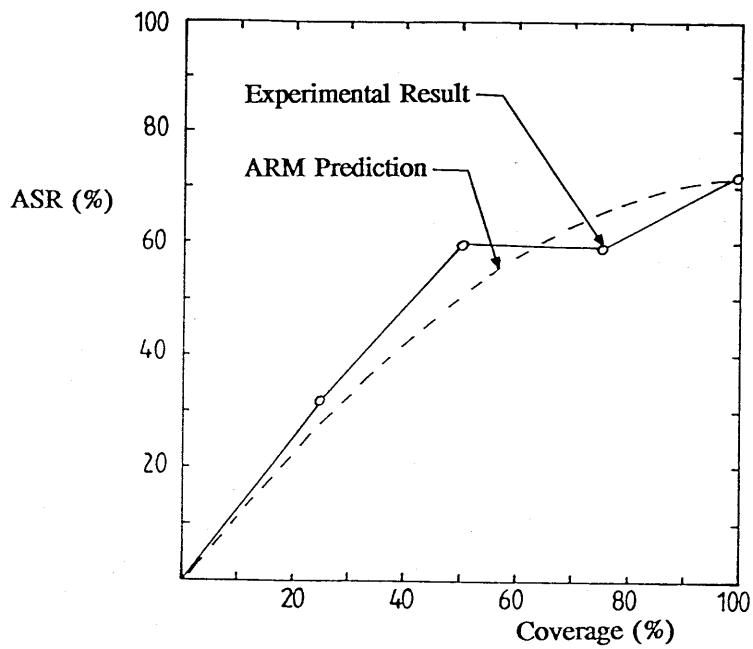


Fig. 7.3.5 Amplitude Suppression Ratio vs. Coverage.
(Uniform velocity profile, Mode 1, Strake fitted centrally)

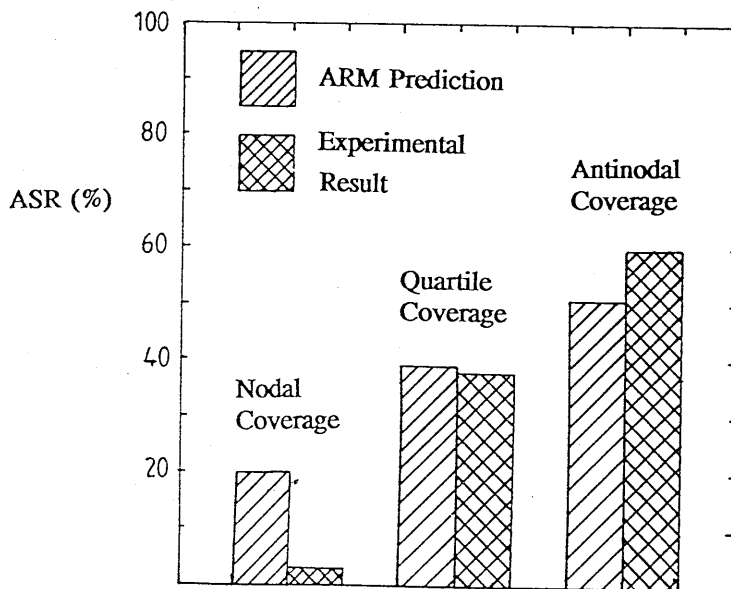


Fig. 7.3.6 Amplitude Suppression Ratio with the strake fitted in three different positions.
(Uniform velocity profile, 50% coverage)

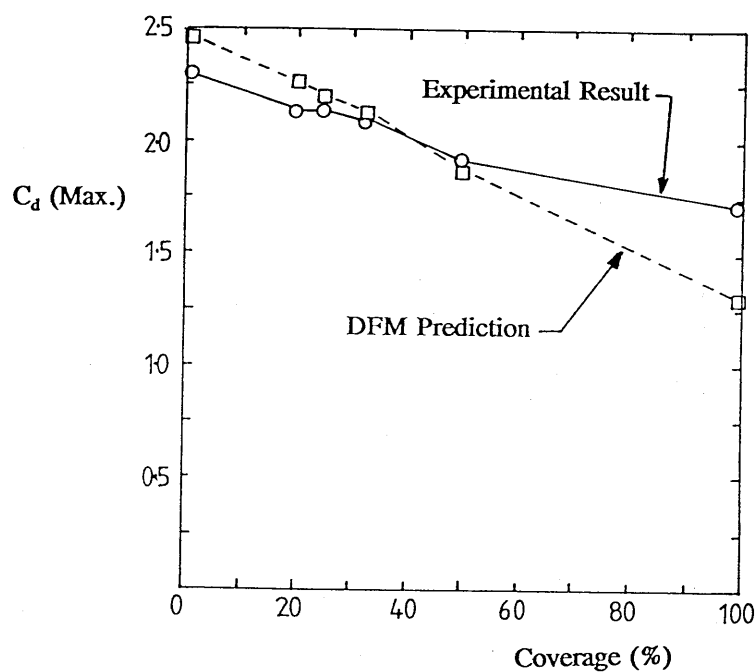


Fig. 7.3.7 Maximum drag coefficient vs. Coverage.
(Uniform velocity profile, Shroud fitted)

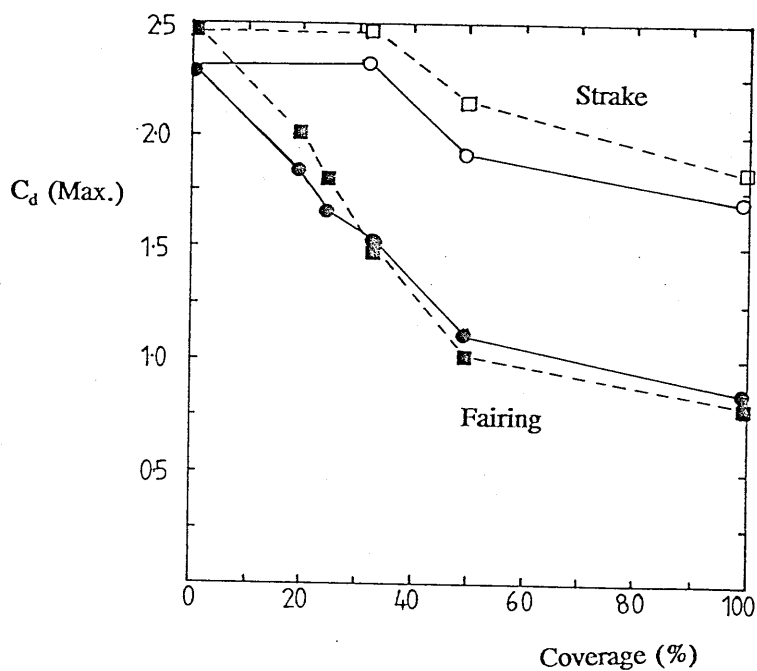


Fig. 7.3.8 Maximum drag coefficient vs. Coverage.
(Uniform velocity profile, Fairing or Strake fitted)

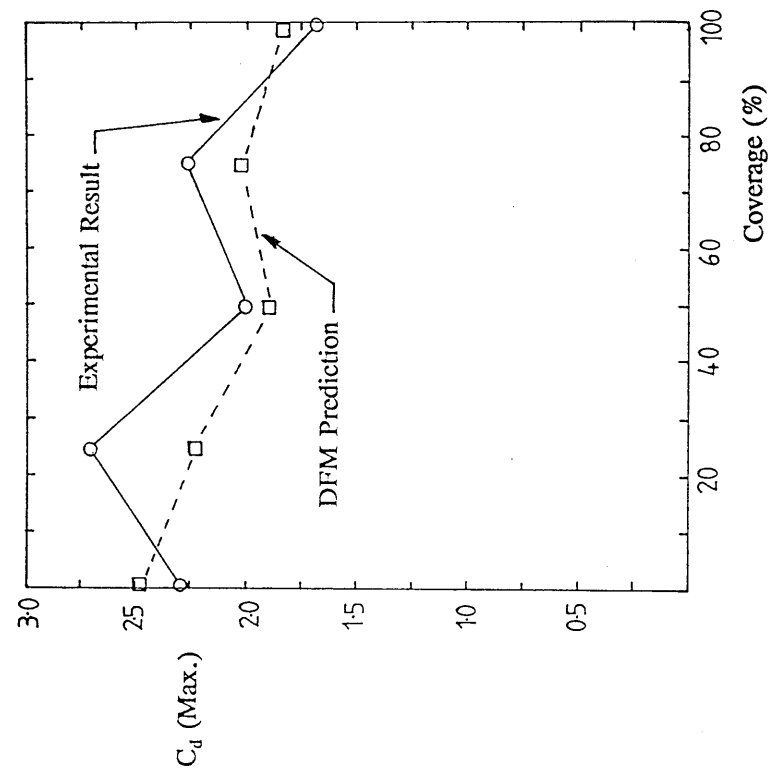


Fig. 7.3.9 Maximum drag coefficient vs. Coverage.
(Uniform velocity profile, Strake fitted centrally)

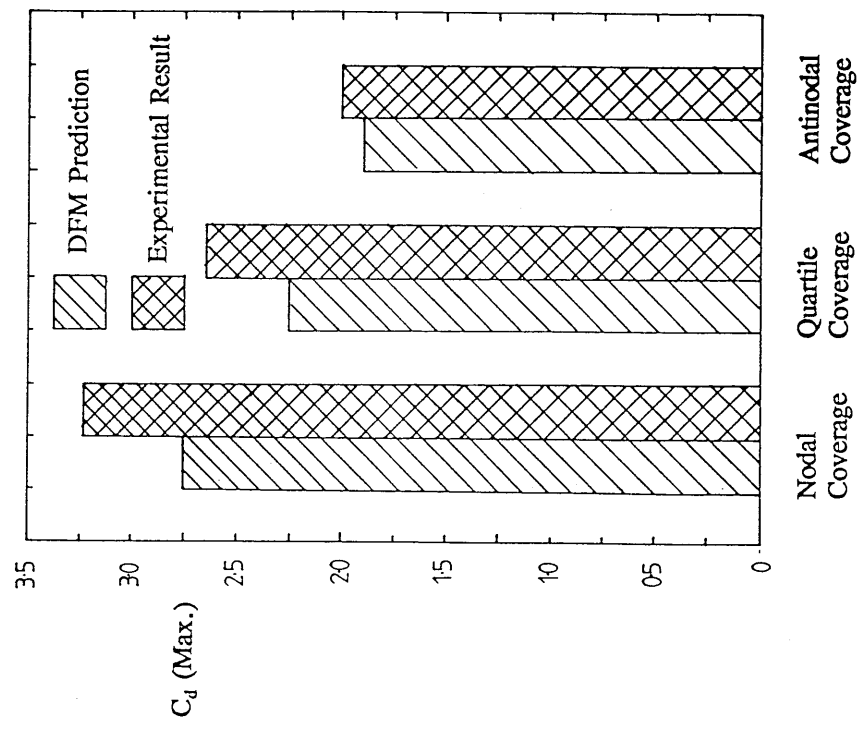


Fig. 7.3.10 Maximum drag coefficient with the strake fitted in three different positions.
(Uniform velocity profile, 50% coverage)

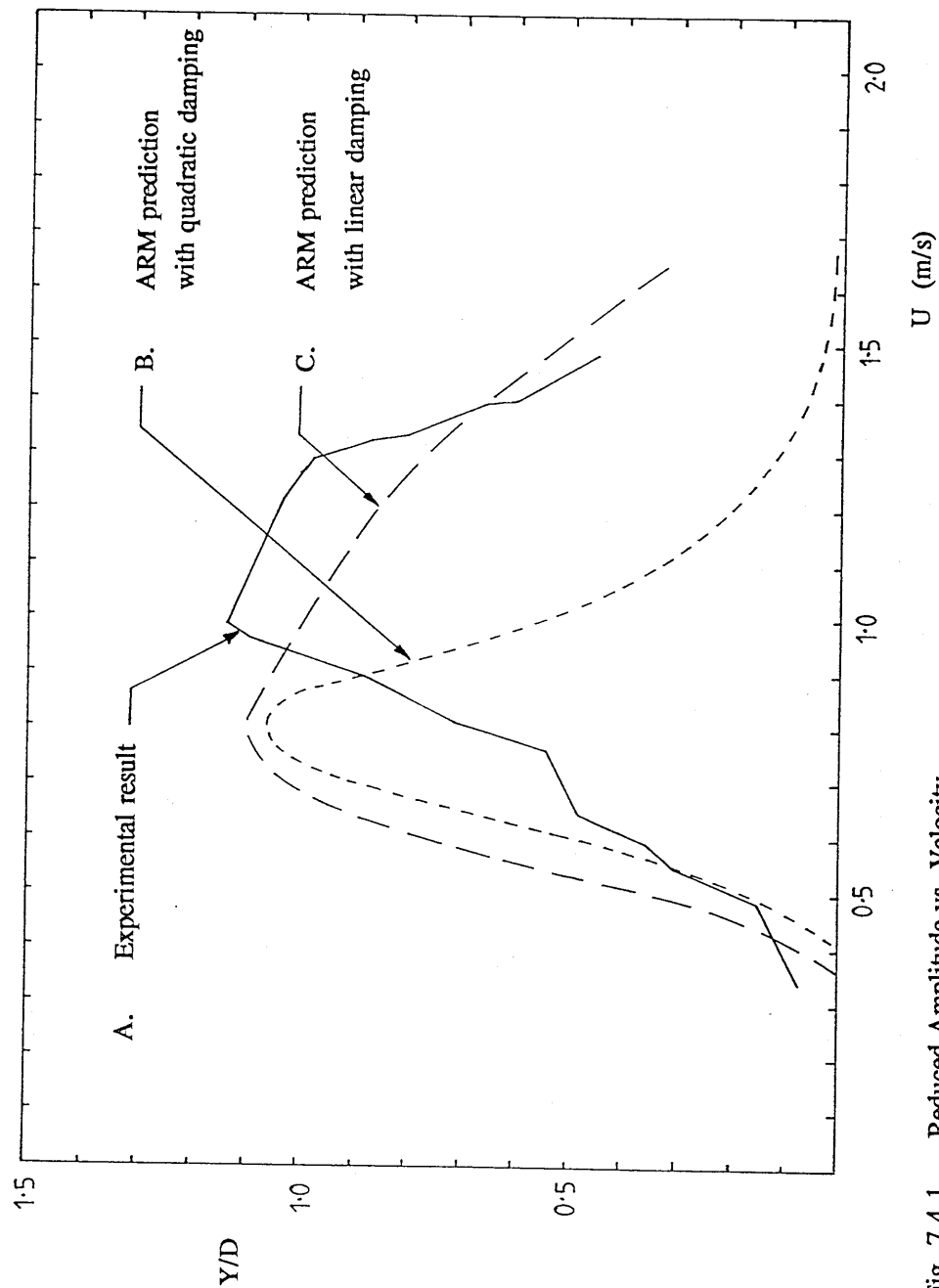


Fig. 7.4.1 Reduced Amplitude vs. Velocity.
(LS30 velocity profile, No device fitted, Mode 1)

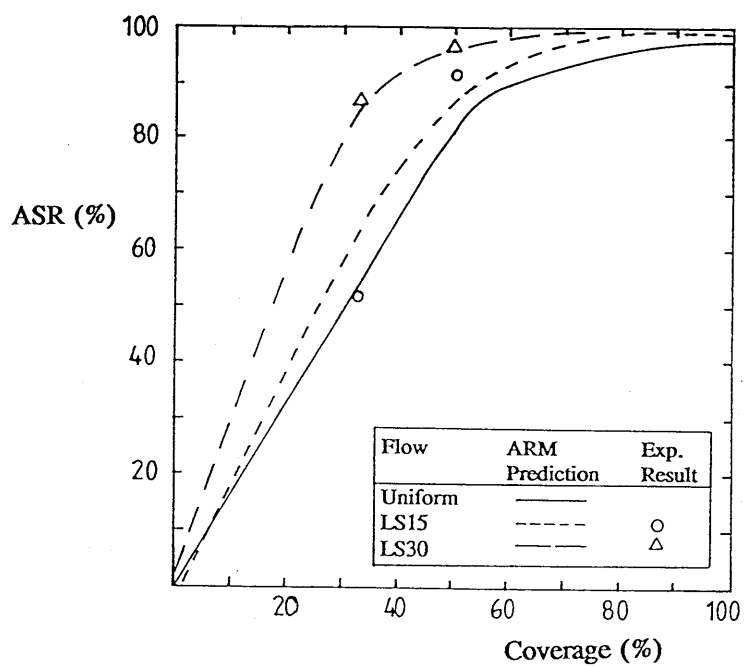


Fig. 7.4.2 Amplitude Suppression Ratio vs. Coverage.
(Linear shear profiles, Mode 1, Fairing fitted)

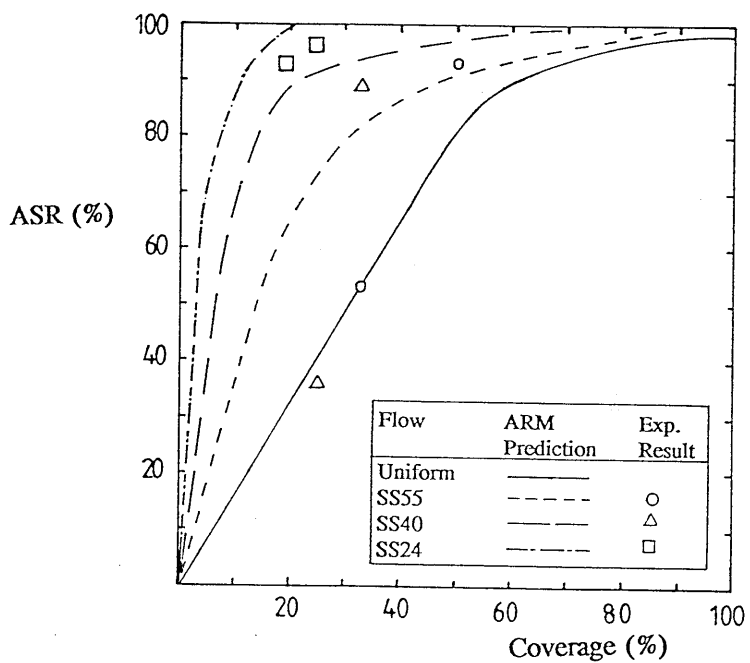


Fig. 7.4.3 Amplitude Suppression Ratio vs. Coverage.
(Stepped shear profiles, Mode 1, Fairing fitted)

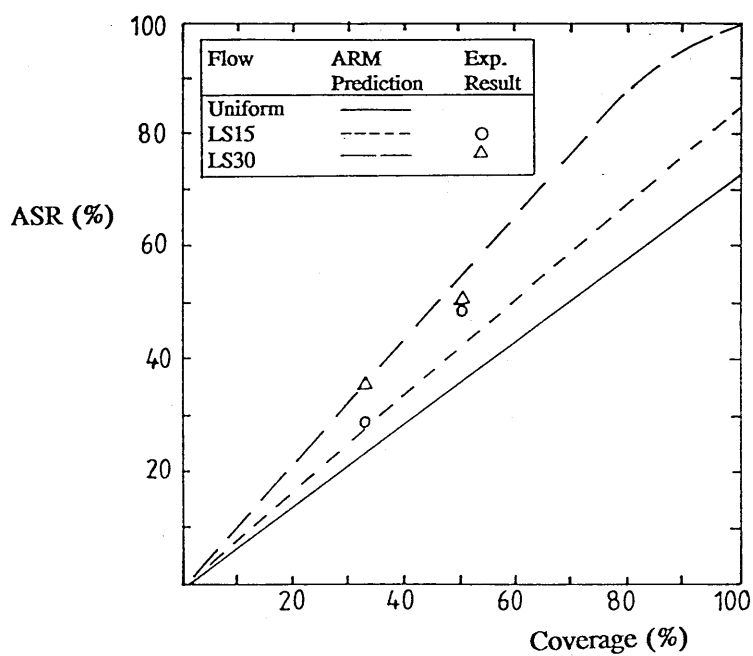


Fig. 7.4.4 Amplitude Suppression Ratio vs. Coverage.
(Linear shear profiles, Mode 1, Strake fitted)

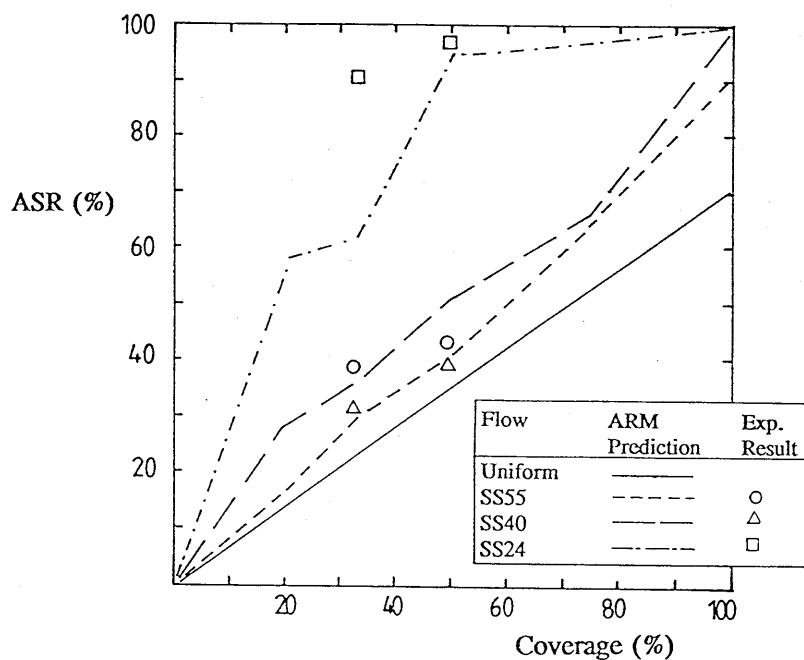


Fig. 7.4.5 Amplitude Suppression Ratio vs. Coverage.
(Stepped shear profiles, Mode 1, Strake fitted)

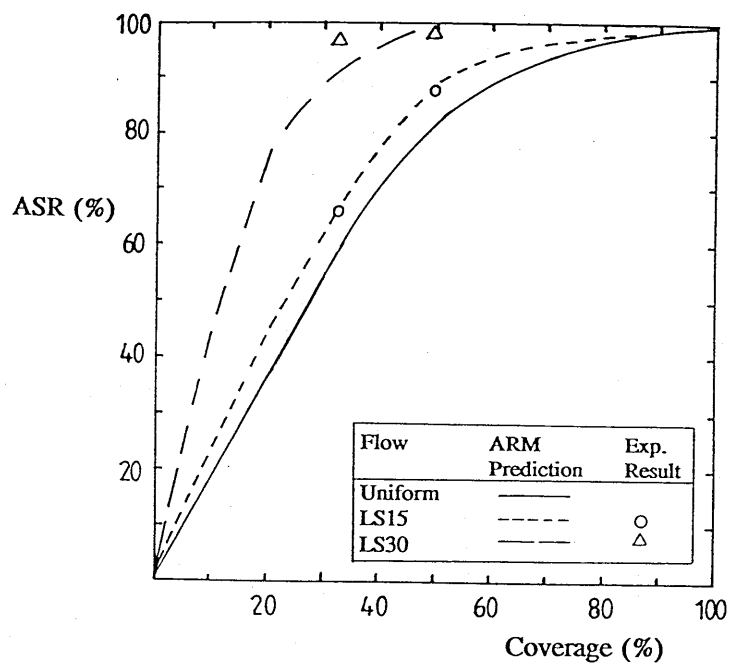


Fig. 7.4.6 Amplitude Suppression Ratio vs. Coverage.
(Linear shear profiles, Mode 2, Fairing fitted)

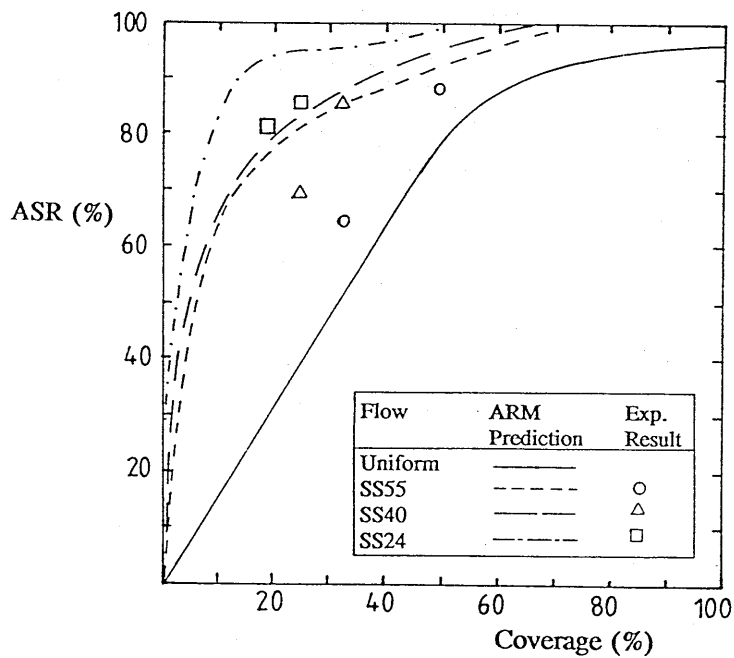


Fig. 7.4.7 Amplitude Suppression Ratio vs. Coverage.
(Stepped shear profiles, Mode 2, Fairing fitted)

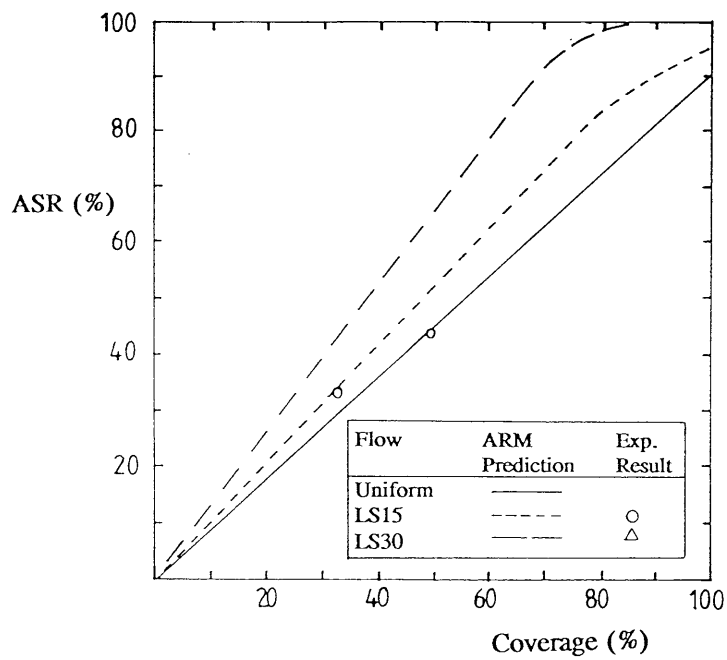


Fig. 7.4.8 Amplitude Suppression Ratio vs. Coverage.
(Linear shear profiles, Mode 2, Strake fitted)

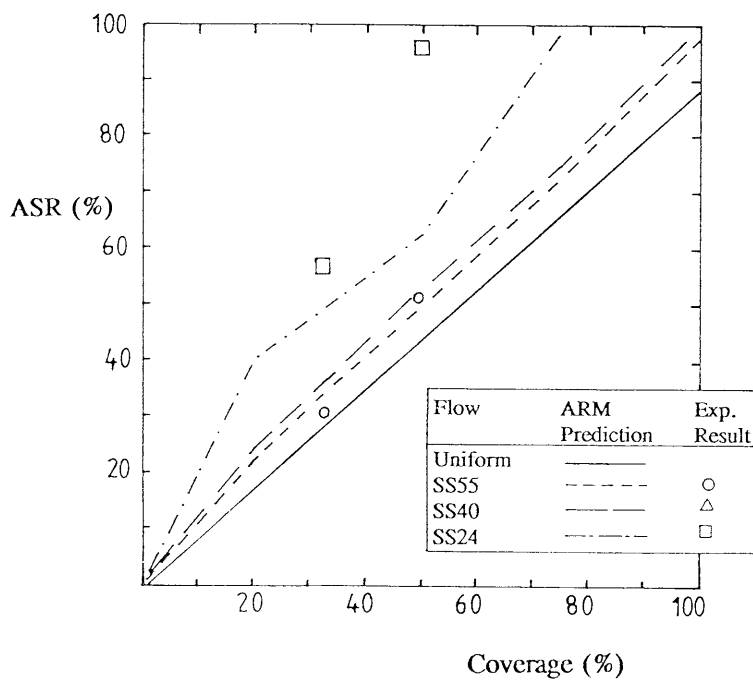


Fig. 7.4.9 Amplitude Suppression Ratio vs. Coverage.
(Stepped shear profiles, Mode 2, Strake fitted)

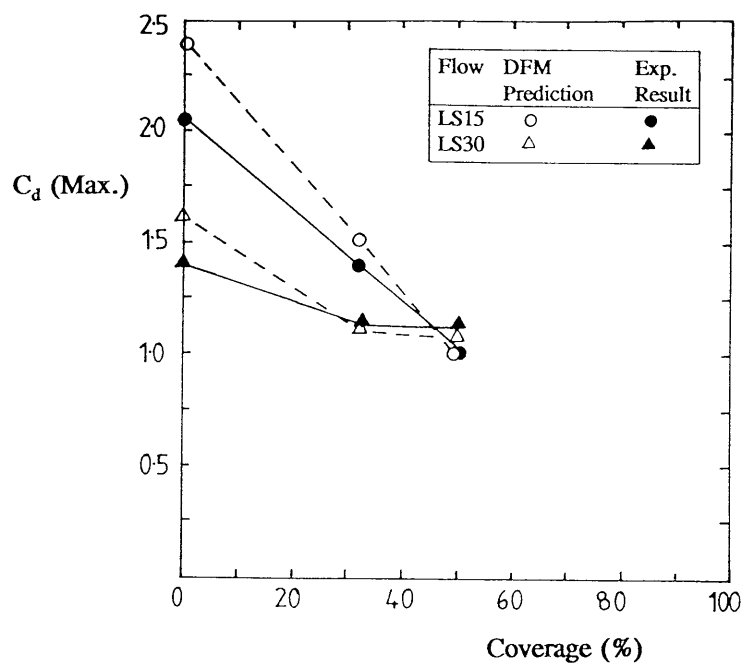


Fig. 7.4.10 Maximum drag coefficient vs. Coverage.
(Linear shear profiles, Fairing fitted)

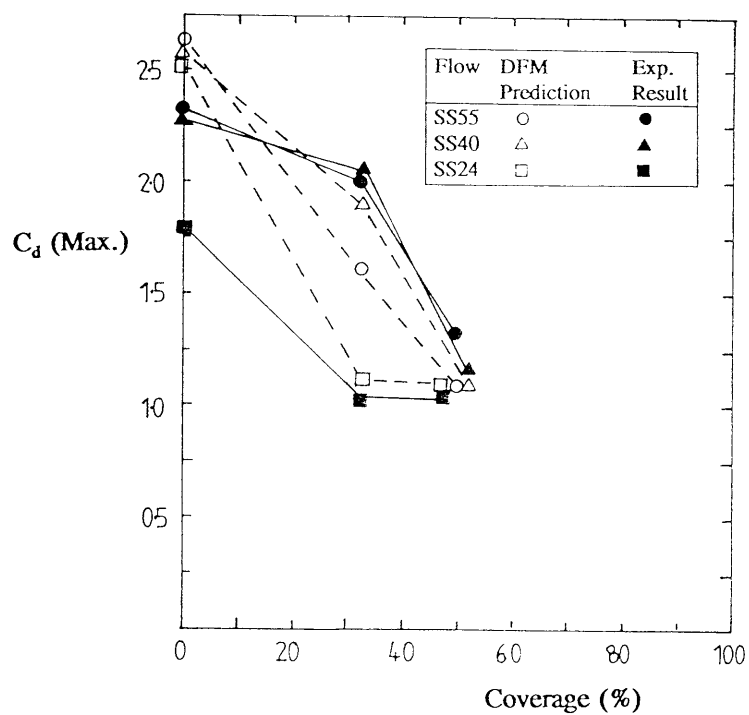


Fig. 7.4.11 Maximum drag coefficient vs. Coverage.
(Stepped shear profiles, Fairing fitted)

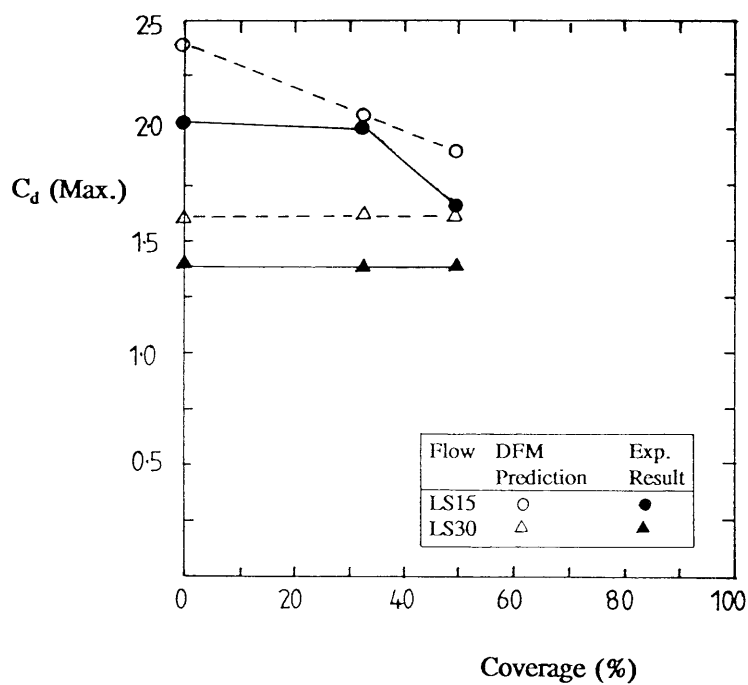


Fig. 7.4.12 Maximum drag coefficient vs. Coverage.
(Linear shear profiles, Strake fitted)

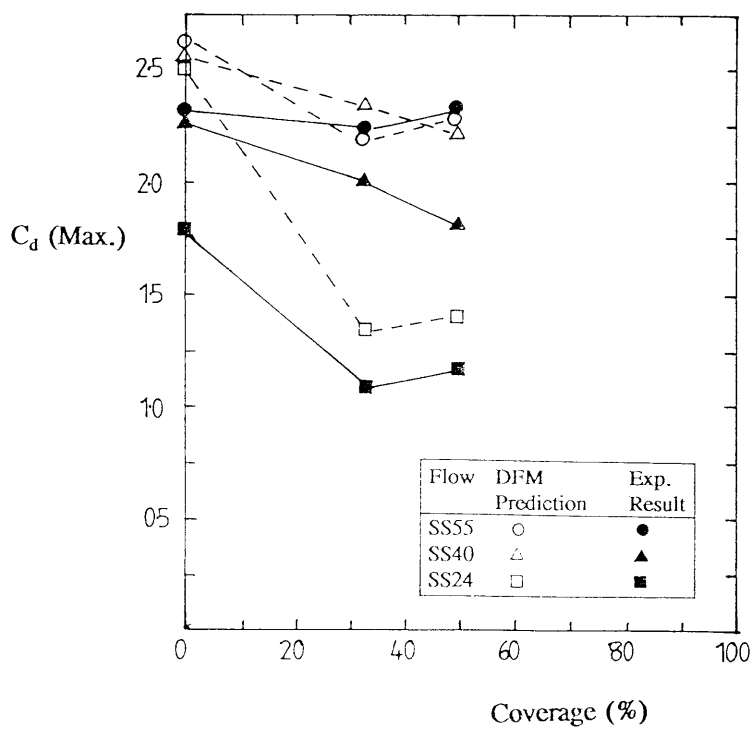


Fig. 7.4.13 Maximum drag coefficient vs. Coverage.
(Stepped shear profiles, Strake fitted)

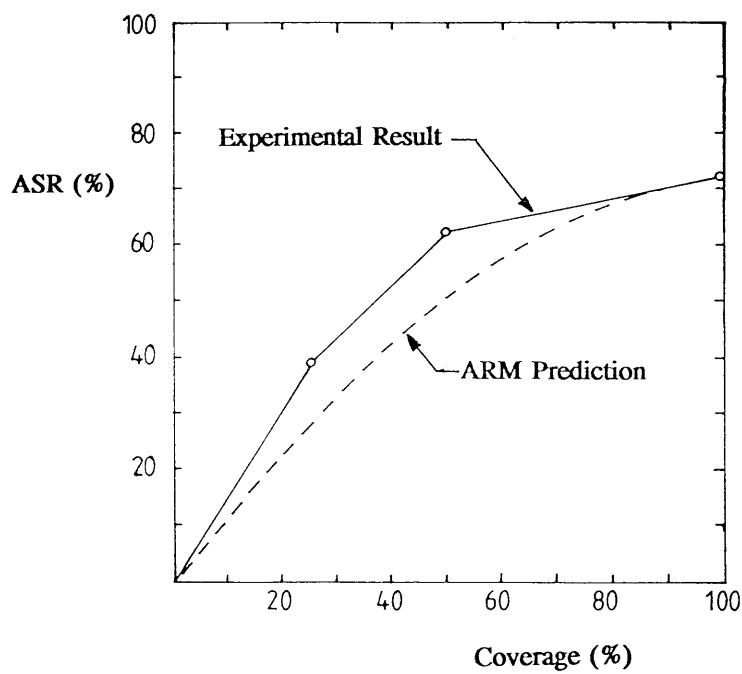


Fig. 7.5.1 Amplitude Suppression Ratio vs. Coverage.
(Model 2, 4.5D stake fitted)

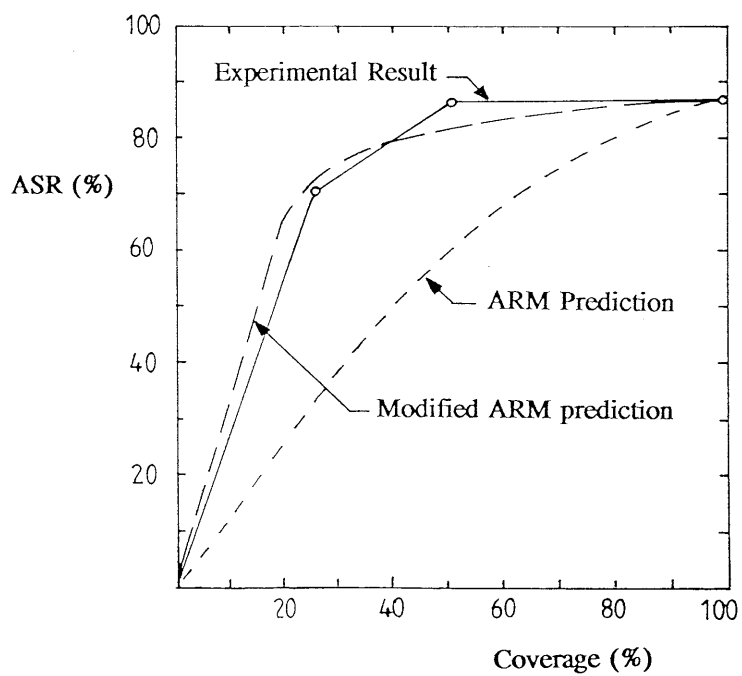


Fig. 7.5.2 Amplitude Suppression Ratio vs. Coverage.
(Model 2, 15D stake fitted)

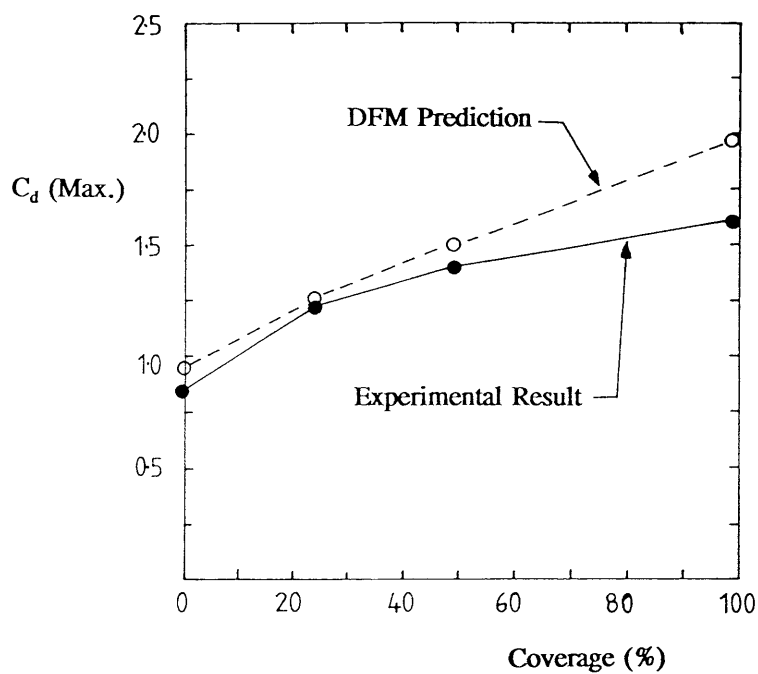


Fig. 7.5.3 Maximum drag coefficient vs. Coverage.
(Model 2, 4.5D stake fitted)

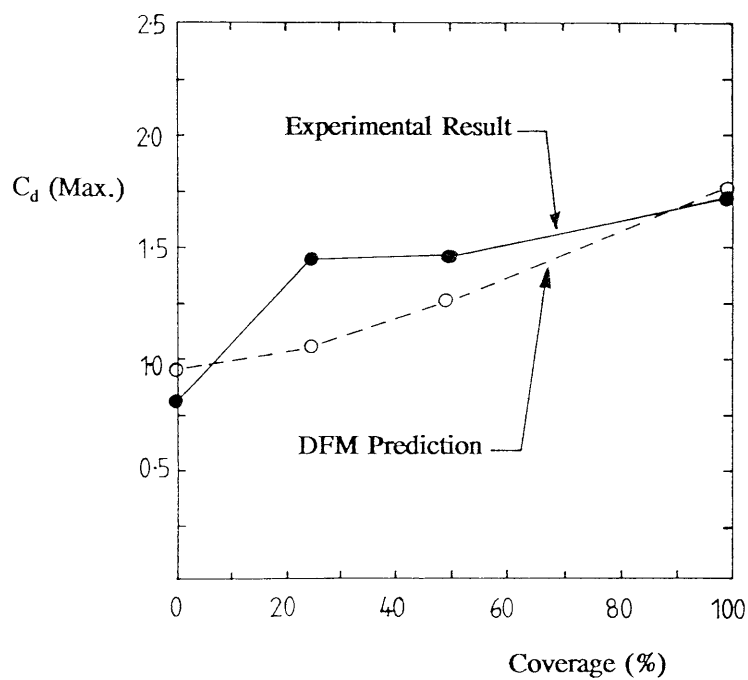


Fig. 7.5.4 Maximum drag coefficient vs. Coverage.
(Model 2, 15D stake fitted)

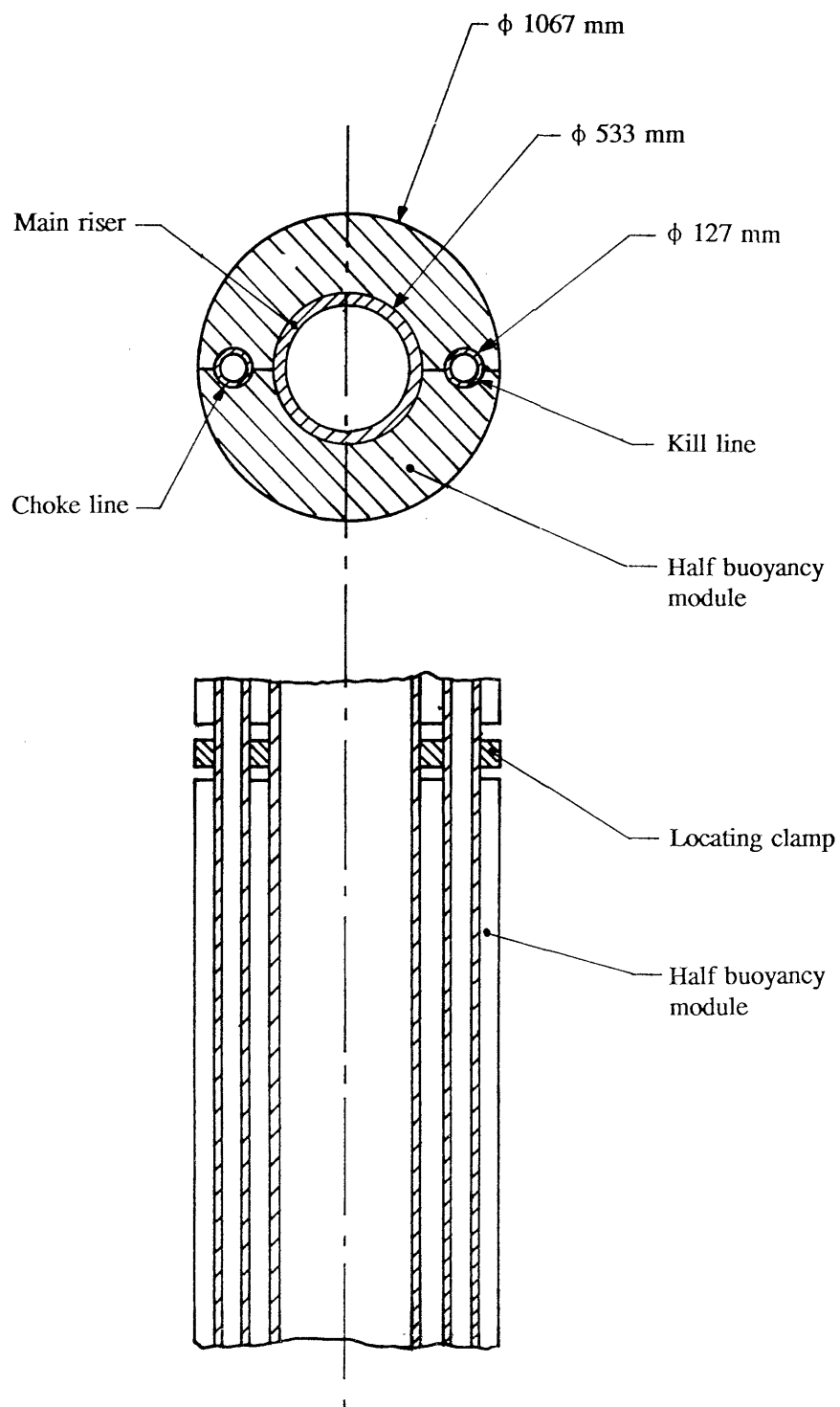


Fig. 8.1 Conventional whole buoyancy module fitted to riser with choke and kill lines.

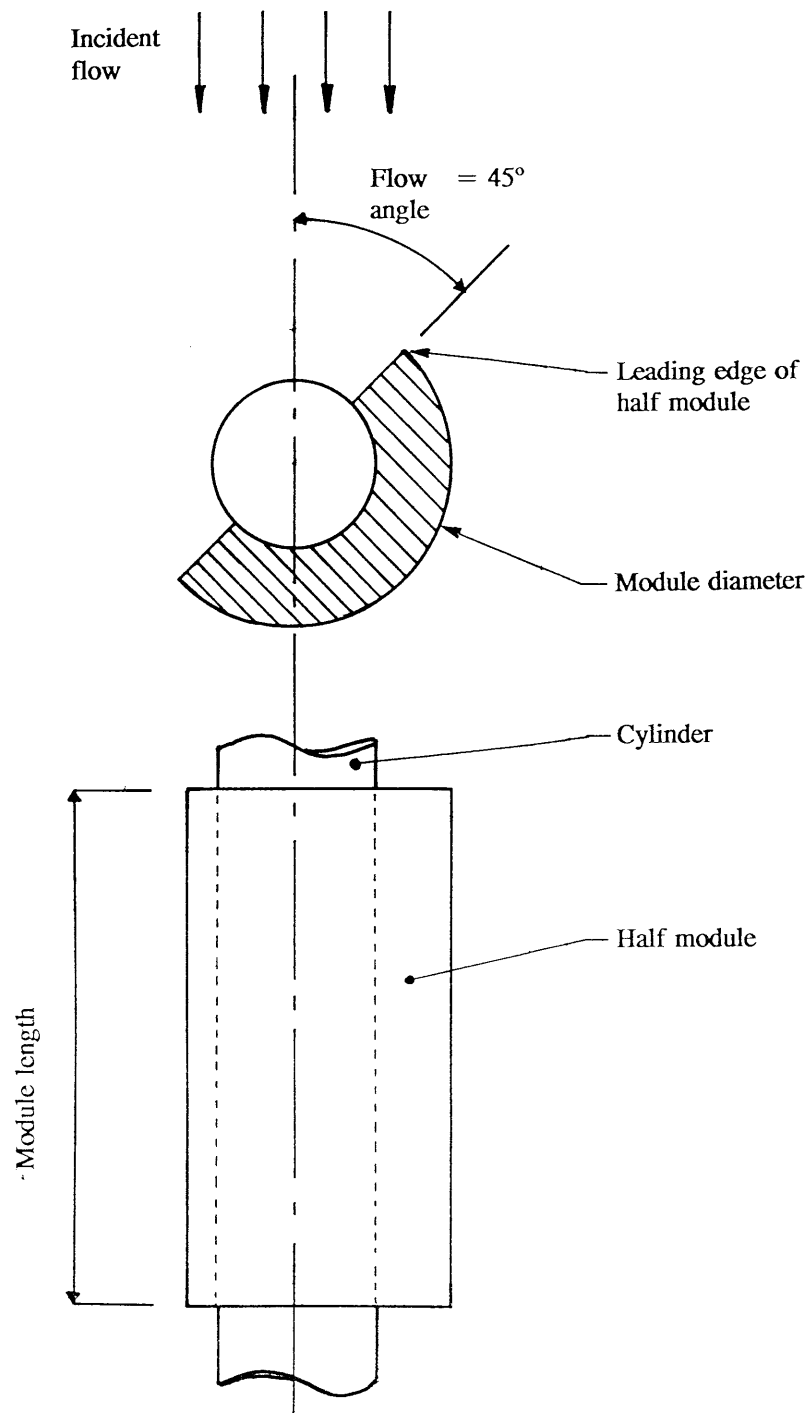


Fig. 8.2 Half buoyancy module fitted to cylinder.

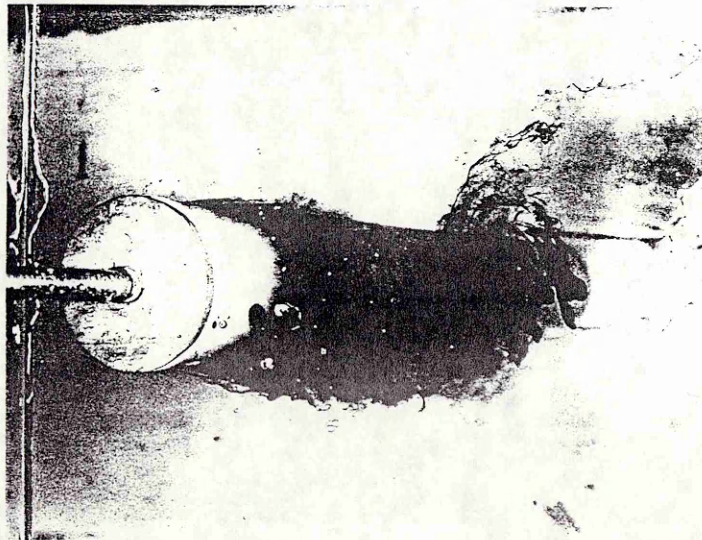


Fig. 8.3 Flow pattern around the bare cylinder.

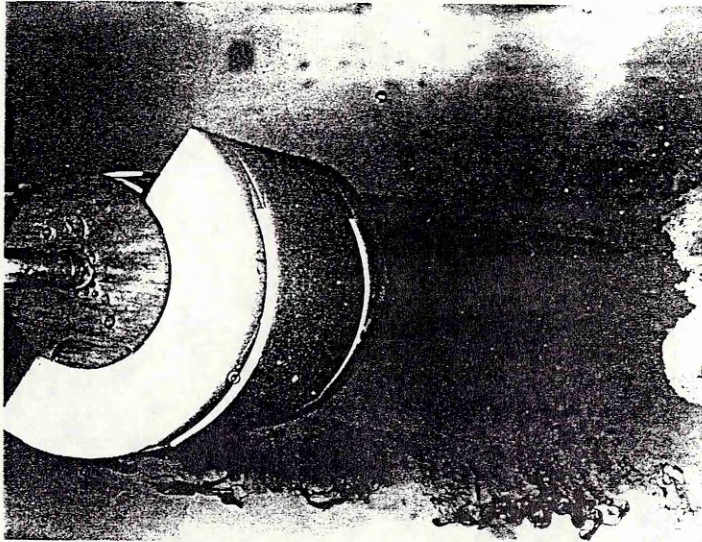


Fig. 8.4 Flow pattern with half buoyancy module fitted: (Flow angle = 45 degrees)

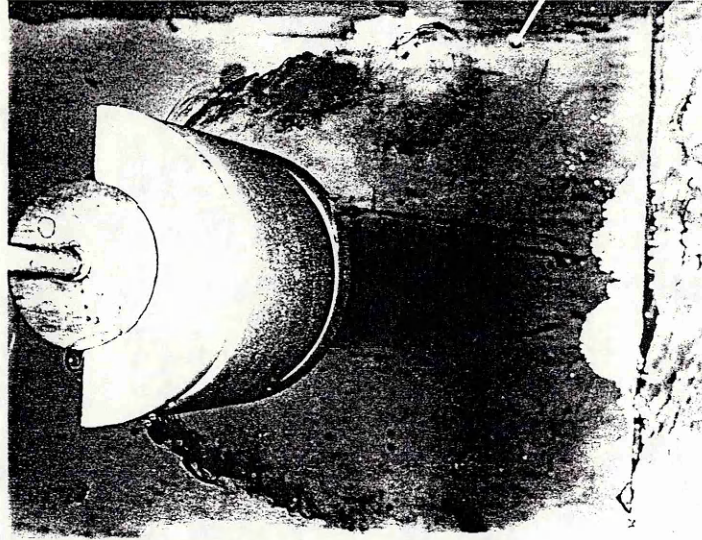


Fig. 8.5 Flow pattern with half buoyancy module fitted: (Flow angle = 90 degrees)

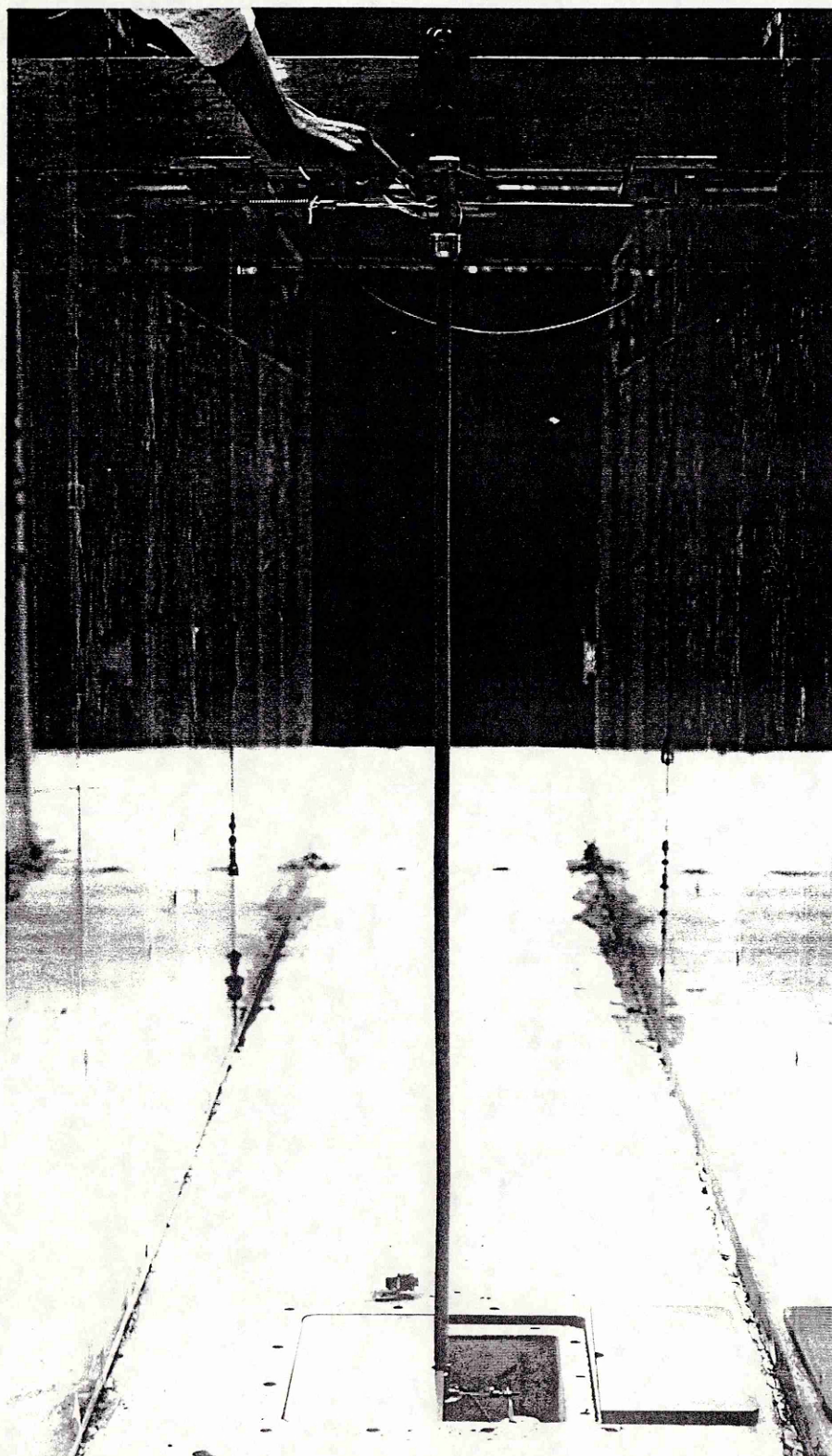


Fig. 8.6

View along the test section of the 1 m flume with the 0.014m diameter model installed.

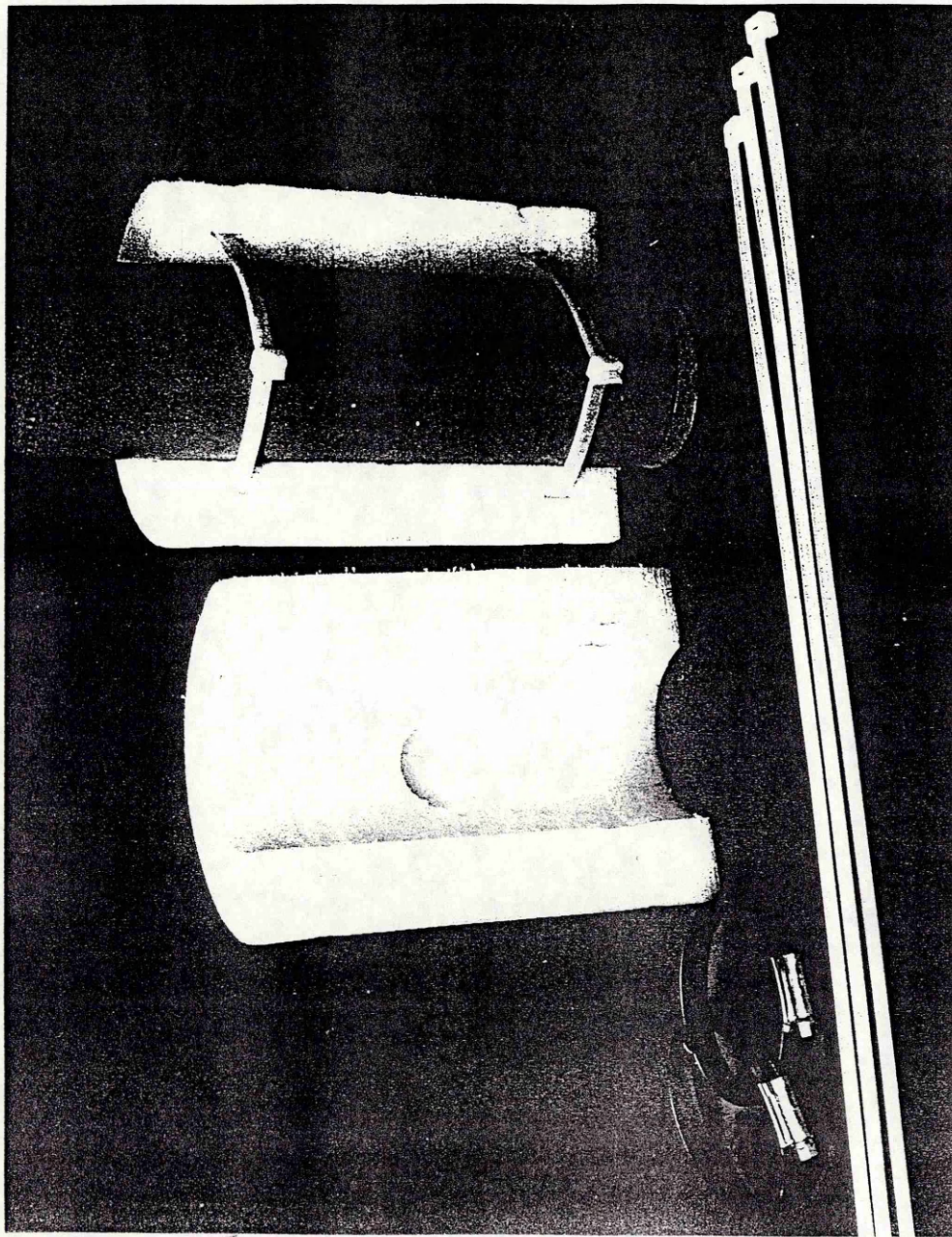


Fig. 8.7 Half buoyancy modules and fixing arrangement used for large scale testing of the BSD.

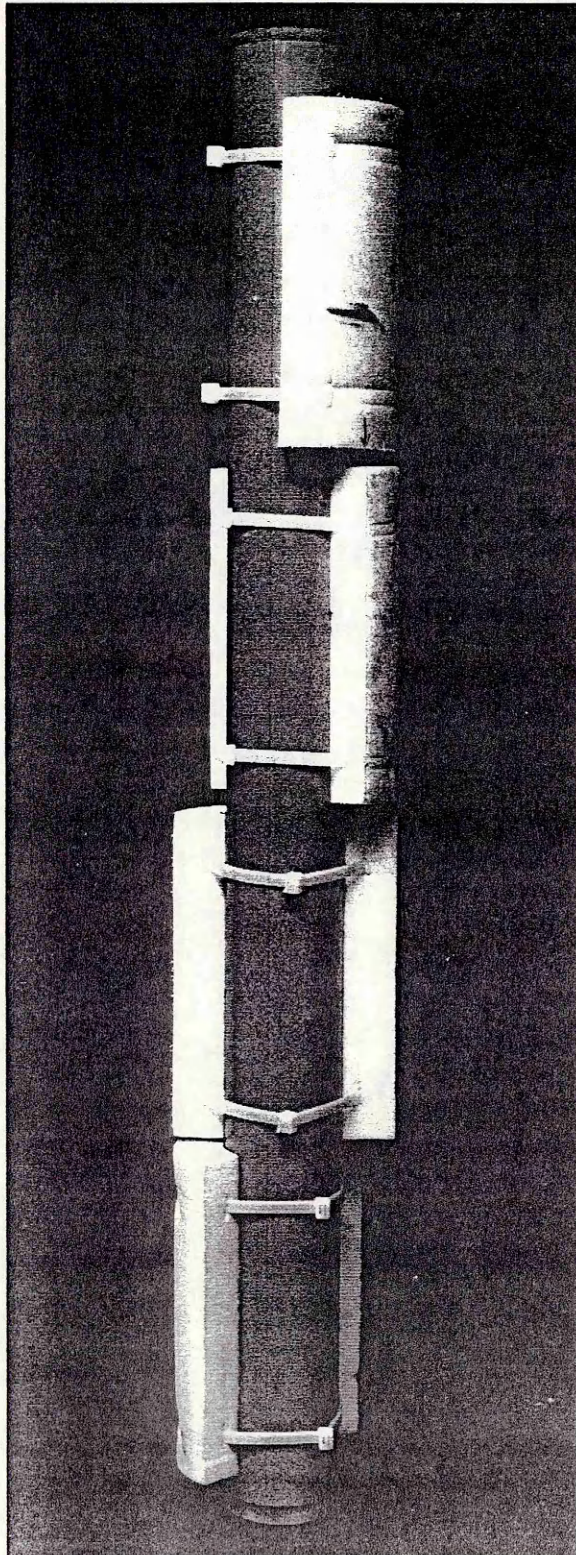


Fig. 8.8 Half buoyancy modules fitted to a cylinder with 45 degree stagger angle.

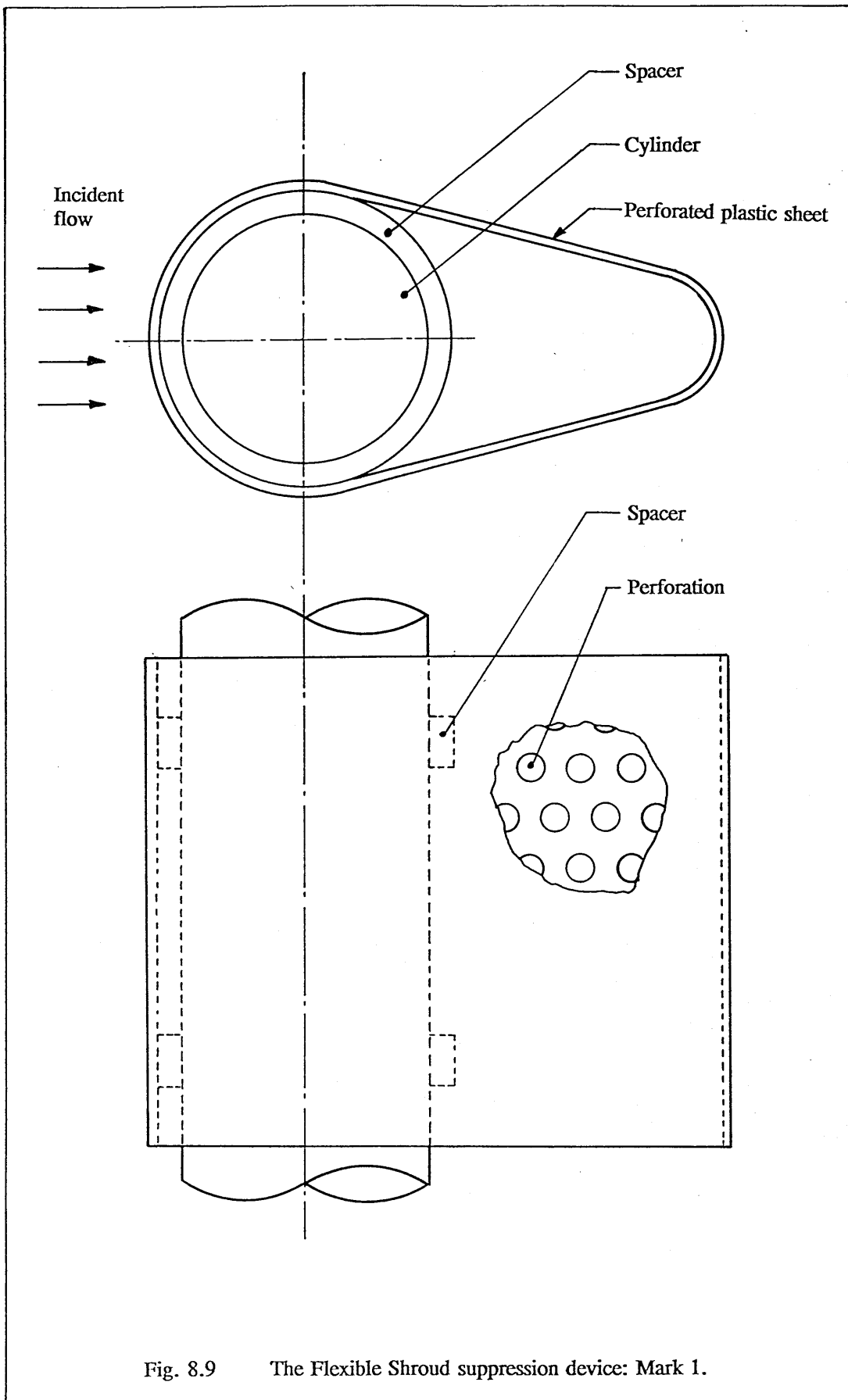


Fig. 8.9 The Flexible Shroud suppression device: Mark 1.

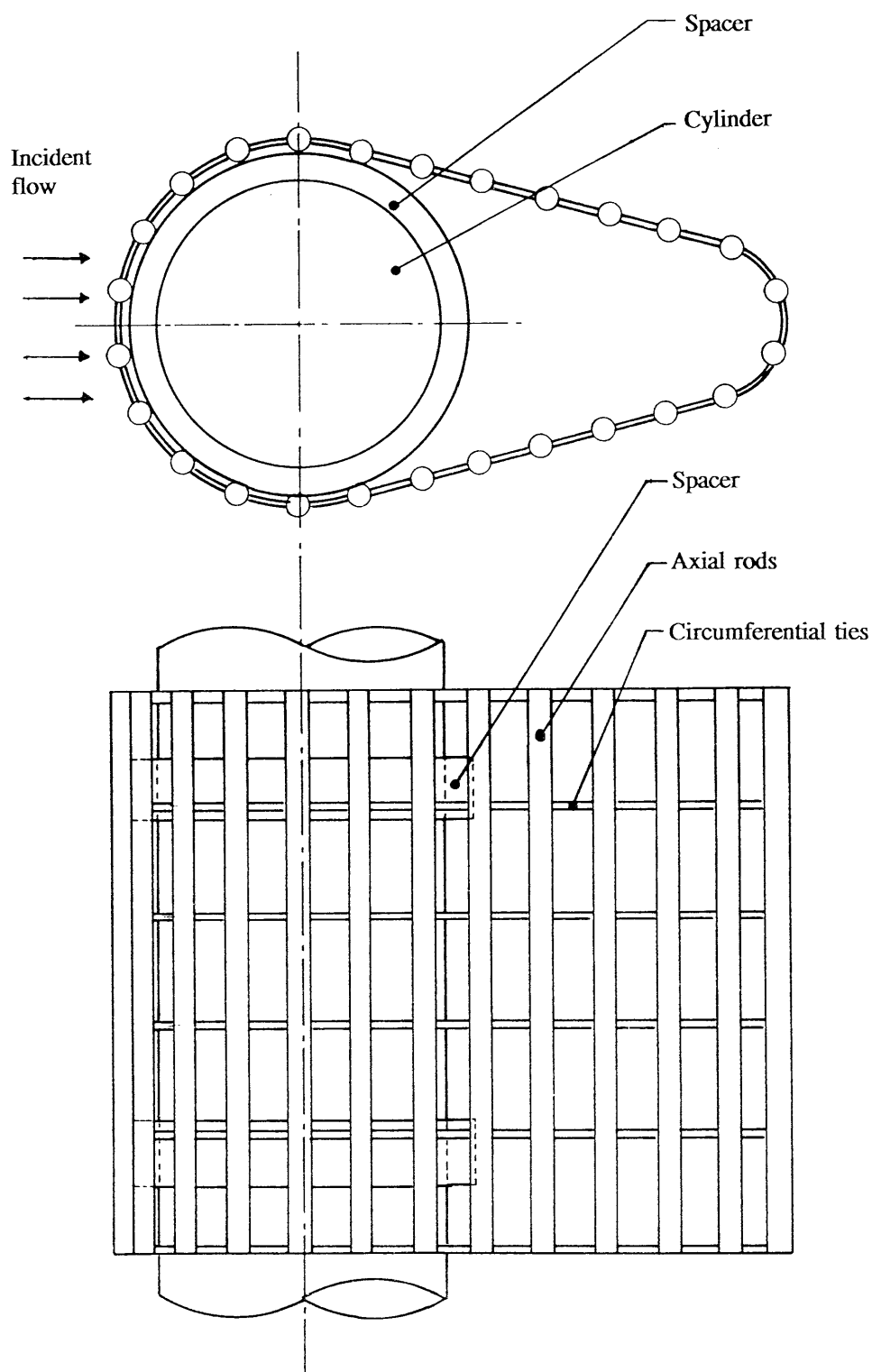


Fig. 8.10 The Flexible Shroud suppression device: Mark 2.