ADDED HYDRODYNAMIC LOADING DUE TO SACRIFICIAL ANODES

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There is a scarcity of data relating to the hydrodynamic loading due to sacrificial anodes A method of calculating factors to apply to circular cylinder force coefficients, using the discrete vortex method is described The results are presented for uni-directional and oscillatory flows Equivalent force coefficients are calculated and used in a Morison type analysis of a simple jacket structure in a current and in waves The overall in-line loading was found to be increased by as much as 9% in modest environmental conditions

INTRODUCTION

Steel offshore jacket structures are generally protected against corrosion by sacrificial anodes, an impressed current system or a hybrid of the two Sacrificial anodes are cast from reactive metals (normally zinc or aluminum alloys) which are more electro-negative than the structures they protect, and so corrode in preferance to the structural steel of the jacket Their size is dictated by the desired current and by manufacturing constraints They are long and slender, of trapezoidal or circular section (perhaps of dimension 2 00 × 0 25 × 0 25m) and weigh in the order of hundreds of kilograms The optimum stand-off distance between the underside of the anode and the structure to which it is attached has been found to be in the region of 0 25 to 0 35m Sacrificial anodes are distributed all over the wetted surface of jackets and can number many hundreds even for structures of modest dimensions It has been estimated that the weight of anodes attached to jackets destined for the North Sea comprises between 4 and 6% of the total weight of the structure when ready for launch[(1)]

Sacrificial anode systems are usually designed with reference to codes or standards issued by a certifying authority, a government agency, or the owner or operator of the platform A commonly used code is the DNV Recommended Practice RPB401 'Cathodic Protection Design'[(2)] Whilst codes and guides give detailed information and procedures concerning protection against corrosion, little is specified concerning

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the fluid loading relating to individual anodes or their effect on the fluid loading of the structure as a whole

The in-line force, F, experienced by an element of a jacket structure due to waves is calculated using Morison's equation which, in its simplest form, may be expressed as,

$$F = \frac{1}{2}\rho A C_D U|U| + \rho V C_M U, \tag{1}$$

where A and V are the projected area and the volume of the element, U and U are the local undisturbed velocity and acceleration of the fluid of density ρ, and C_D and C_M are the drag and inertia coefficients respectively The terms on the right hand side of the equation represent the drag force and the inertia force experienced by the member in the direction of the ambient flow Whilst a great deal of research has been carried out to establish values for the force coefficients (see Sarpkaya[3], for example) they continue to be a matter of debate This is because it is difficult to devise appropriate small scale experiments, interpret full scale data (which are in any case scarce) or develop realistic mathematical models for the extremely complex flows involved Furthermore, the coefficients have been shown to be highly dependent on roughness and protrusions produced by marine growth, or other causes It is evident that the local forces are likely to be significantly influenced by the presence of sacrificial anodes

Where the matter is discussed in codes and guidelines it is generally considered that the presence of anodes does not significantly increase volumes and inertia coefficients, but it does increase areas and drag forces It is therefore normal practice when assessing the drag on structural elements to allow for the increases resulting from the presence of anodes by making adjustments to drag coefficients This may involve the use of 'equivalent' force coefficients based on the projected areas of the members plus anodes, or it may simply involve the application of a global increase of the drag coefficients of anode bearing members by between 7 and 10%[1] When carrying out fatigue calculations for the anode itself, the fluid particle kinematics may be modified in accordance with simple potential flow theory in which, for example, the velocity at an anode located at the shoulder of a cylinder is taken as approximately twice that of the ambient flow Estimates of the increase in the net loading on the structure when the force coefficients are modified to account for the presence of the anodes vary from 5%[1] to values several times as much, according to the author of the calculation and the assumptions made

Such procedures may appear rather vague and simplistic, but the data and the more sophisticated models required for a better informed approach simply do not exist Although a considerable amount of work has been carried out on interference effects, the configurations considered are usually a small number of cylinders of comparable size, or multi-tube arrangements studied in connection with heat exchangers There has also been some work on risers and mult-tube risers[6], but for the most part work in this area relates to specific design projects A modest experimental programme carried out by Singh *et al* [7] into the flow past cylinders with various appurtenances in waves at low Keulegan Carpenter number suggested further work on the subject As far as is known no further work has been done apart from an experimental programme, which is still in progress, and some numerical work, which forms the subject of this paper, carried out by the present authors and associates

The experimental programme to date comprises an investigation of anode drag in uni-directional flow[8], and an investigation of the drag of 'blocked anodes', in which the gap between the anode and the member is blocked Both experiments were carried out on cylinders of diameter D in wind tunnels at Reynolds numbers $(U_m D/\nu)$ of approximately 10^5, where U_m is the magnitude of the velocity of the air of kinematic viscosity ν A further experiment is planned on anode bearing cylinders using a wave flume capable of generating waves with Reynolds numbers up to 2×10^5 and Keulegan Carpenter numbers up to 50 The objectives of the investigations are to establish the fluid phenomena involved in such flows and to devise a method of estimating the hydrodynamic forces experienced by anode bearing structural members, taking those phenomena into account

MODELLING THE FLOW

Uni-directional flow about circular cylinders involves boundary layer separation and the formation of a wake downstream from the body composed of regular, organised vortex structures which diffuse with, and are convected by, the ambient flow field The flow is Reynolds number dependent At low Reynolds numbers the boundary layers are relatively thick and viscous diffusion is important As the Reynolds number increases the boundary layers become progressively thinner (giving rise to increasingly high localised velocity gradients) and convection increasingly dominates the flow A drag crisis, in which the drag coefficient decreases dramatically, occurs at a Reynolds number of around 2×10^5 when the boundary layers undergo transition from laminar to turbulent flow At still higher Reynolds numbers the flow regains some of the structure observed before transition in the sense that a regular vortex shedding process may once again be detected The mean location of the time dependent boundary layer separation points vary with Reynolds number and are of great importance in relation to the Reynolds number dependency of the drag

If uni-directional flow about cylinders is difficult to model, wave flows are even more so The particle kinematics, which are essentially orbital in nature, can be represented realistically for many applications by planar oscillatory flow Even so, vorticity generated by the cylinder on one half cycle of the flow may be swept back over the cylinder to interact with the vorticity being generated on the subsequent half cycle The complex vortex shedding pattern that ensues is critically dependent not only on the Reynolds number, but also on the Keulegan Carpenter number of the flow, which represents the magnitude of the fluid particle orbits relative to the cylinder diameter If the Keulegan Carpenter number is sufficiently low the flow, to all intents and purposes, remains attached If it is sufficiently high, a pseudo Karman vortex street forms on each half cycle of the flow Regimes of interest offshore tend to be characterised by high Reynolds numbers and high Keulegan Carpenter numbers

Adding anodes to the configuration has both advantages and disadvantages as far as modelling the flow is concerned Whilst there is inevitably an interaction between the anode and cylinder flows, in certain arrangements the separation points tend to become fixed On these grounds, as for flat plates and square cylinders, it can be argued that the Reynolds number dependency of the anode bearing cylinder is likely to be less than that for the bare cylinder It has in any case become customary, because there is no other choice, to endeavour to predict flow at high Reynolds numbers from small scale experiments and low Reynolds number computations The same approach has been taken in the work to be described

Since it is not possible to model the flow rigorously using even the most sophisticated of numerical techniques and the most powerful computers available, and since for various reasons comprehensive large scale experiments are prohibitive, the approach adopted uses as simple a numerical model as possible and available experimental data The easiest approach is to use force coefficients for cylinders in unbounded flows (which are readily available) and to calculate an equivalent coefficient for the cylinder plus anodes based on the total projected area The approach could be made more sophisticated by using the force coefficients in conjunction with velocities calculated using simple potential theory It could be further refined by using instead drag coefficients for cylinders in close proximity to a wall, for which results are also available However, it has been shown(9) using experimental results from experiments on anodes in unbounded flows, ground flows and attached to cylinders(8), that these overly simplistic approaches do not give satisfactory results This is not surprising Potential flow theory can only give the crudest approximation to the kinematics of such complex separated flows, and in any case the flow about anode bearing cylinders is quite different from unbounded cylinders or cylinders in a ground flow For example, although anodes attached transverse to the flow increase the drag considerably, those attached in-line with the flow have been found to decrease it in relation to that of a bare cylinder(8)

The next simplest class of models available, which have been used with some success with separated flows, are the discrete vortex models and those that have evolved from them This approach is used to model the flow around a circular cylinder with a protuberance with a view to producing factors which may be applied to circular cylinder force coefficients so as to give an estimate of the forces experienced by cylinders bearing anodes

THE NUMERICAL MODEL

The flow about a structural member with anodes was modelled by a two-dimensional circular cylinder with protrusions in an incompressible time-dependent flow This was on the basis that in reality the gap between the anode and the member becomes effectively 'blocked' with encrustations and weed A two-dimensional model was chosen because a three-dimensional one is not viable Also, in two dimensions, it is possible to treat the flow about the deformed cylinder through calculation of the flow about a circular cylinder, which is relatively simple, using a conformal transformation technique The transformation which takes a circle of radius a (in the ζ-plane) into a deformed circle (in the z-plane) is based on a Laurent series and is given by,

$$z = e^{i\alpha}(\zeta + \sum_{k=1}^{N_t} \frac{a_k}{\zeta^k}) \tag{2}$$

All calculation is carried out in the circle plane, although the usual equations for flow about a circular cylinder have to be duly modified to take account of the flow in the real (transformed) plane

The method used is based on the discrete vortex method which, along with other allied methods, has been thoroughly reviewed by Sarpkaya[13] The flow is described by the vorticity convection equation,

$$\frac{\partial \omega}{\partial t} + u \cdot \nabla \omega = \nu \nabla^2 \omega, \tag{3}$$

which is solved by an operator splitting technique involving a diffusion step and a convection step over each timestep of a continually evolving flow

The vorticity in the flow, which is largely confined to subdomains in an otherwise irrotational flow, is modelled by arrays of point singularities, or point vortices At each timestep discrete vortices are introduced around the cylinder so as to satisfy appropriate boundary conditions The vortex circulations are then distributed over the nodes of a grid covering the computational domain, where they are allowed to diffuse in accordance with the diffusion equation,

$$\frac{\partial \omega}{\partial t} = \nu \nabla^2 \omega, \tag{4}$$

using a finite difference scheme and properly accounting for the diffusion in the real plane[10]

With the updated distribution of vorticity, the convection step is carried out in accordance with the equation,

$$\frac{\partial \omega}{\partial t} + u \quad \nabla\omega = 0, \tag{5}$$

using the basic discrete vortex method, a Lagrangian approach in which the vorticity, $\omega = \nabla \times u$, is represented by discrete vortices which are convected with the flow

The nodal velocities are calculated[10] via the complex potential,

$$W = \phi + i\psi = (\zeta - \frac{a^2}{\zeta})U + \frac{i}{2\pi}\sum_{j=1}^{N_v} \Gamma_j(\mathrm{Ln}(\zeta - \zeta_j) - \mathrm{Ln}(\zeta - \zeta_j^*)), \tag{6}$$

where ϕ is the velocity potential, ψ is the stream function, U is the free stream velocity, Γ_j is the strength of the j^{th} nodal vortex and ζ_j^* is the location of its image vortex inside the circle required to maintain the boundary condition, $\partial\phi/\partial n = 0$, on the cylinder surface The velocities of the discrete vortices, located at ζ_p, are calculated by interpolation and moved using a simple time integration scheme,

$$\zeta_p(t + \Delta t) = \zeta_p(t) + \frac{\partial \zeta_p}{\partial t}\Delta t, \tag{7}$$

in which the $\frac{\partial \zeta_p}{\partial t}$ term correctly accounts for the kinematics in the real plane

The complex force ($Z = X + iY$) on the cylinder plus anode may then be calculated from the generalised Blasius equation which can be shown to reduce to[10],

$$Z = 2\pi\rho a^2(1 - \frac{a_1}{a^2}e^{2i\alpha})\frac{\partial U}{\partial t} - i\rho\sum_{p=1}^{N_v}\left((u_p - u_p^*)\Gamma_p - (a + \zeta_p^*)\frac{\partial \Gamma_p}{\partial t}\right), \tag{8}$$

where u_p and u_p^* are the velocities of the p^{th} vortex and its image respectively Force coefficients may be obtained by non-dimensionalising the complex force, with respect to $\frac{1}{2}\rho U_m^2 D$, and decomposed into Fourier averaged drag and inertia coefficients for unsteady flows Further details of the model may be found in reference 10

NUMERICAL RESULTS

The results for the variation of the drag coefficient with Reynolds number for the bare cylinder in uniform uni-directional flow are shown in Figure 1, and can be seen to be in reasonable agreement with experimental results[3] over the sub-critical range The alternate shedding pattern causes an asymmetric time dependent pressure distribution over the cylinder which leads to a zero mean transverse

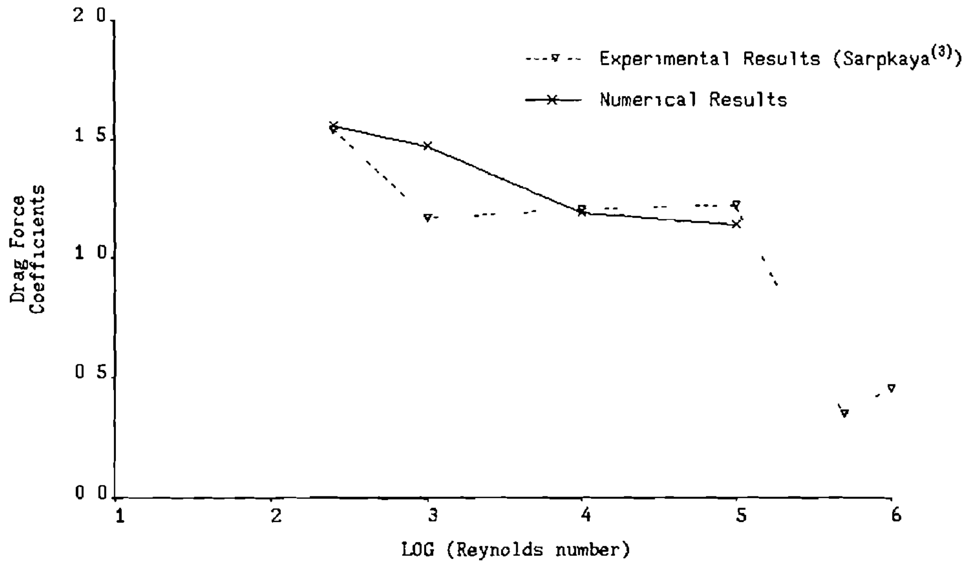
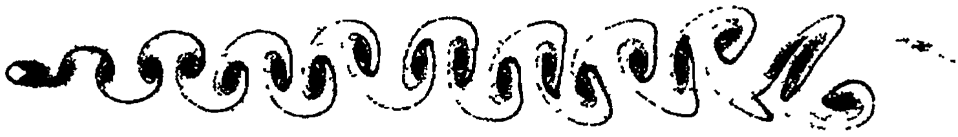


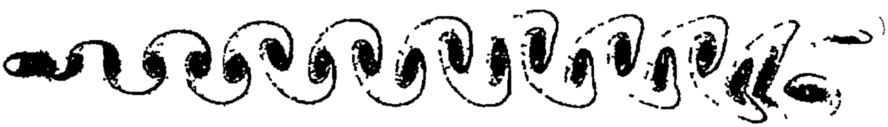
Figure 1 Drag Force Coefficents for Uni-directional Flow Around a Circular Cylinder

(lift) force The root mean square lift coefficients, $C_{L_{rms}}$, range from 0 57 to 0 73 which, again, are in good agreement with experimental results[3]

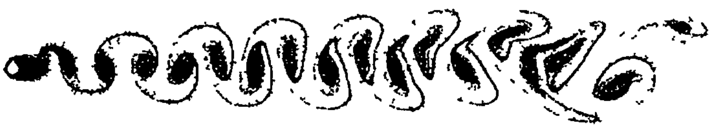
The addition of a single (blocked) anode has a significant influence on the flow, which depends on the angle of inclination, α, of the anode in relation to the flow direction The flow patterns for the anode at a variety of angles of incidence at a Reynolds number of 250 are shown in Figure 2 The presence of the anode has at least two fundemental effects on the flow, It allows the possibility of an asymmetric body with respect to the direction of flow, and over a range of angles it fixes the separation point on one side of the body, which also introduces asymmetries into the flow If the anode is in-line with the flow it behaves rather like a cylinder with a splitter plate ($\alpha = 0°$) or a reversed thick aerofoil ($\alpha = 180°$), leading to a reduction of the drag coefficient in relation to the bare cylinder If the anode is inclined, whether it is leading or trailing the cylinder, it tends to induce separation in such a manner that the vortices roll up closest to the 'lowest' side of the cylinder, leading to a fluctuating lift force with a steady negative non-zero mean The drag increases as the location of the anode approaches the shoulder of the cylinder A typical force trace is shown in Figure 3 and the results for the drag coefficient, expressed as multiples of the bare cylinder drag coefficient, are presented in Table 1 The fluctuations of the transverse force coefficients are of a similar magnitude to those of the bare cylinder, and the magnitude of the steady component of the transverse force coefficients can equal their root mean square value[10] The largest



Anode at 0° Inclination

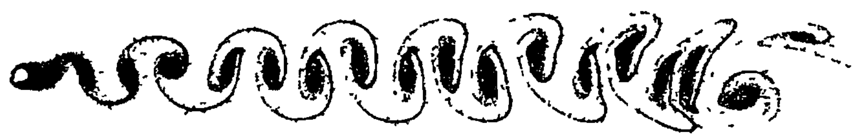


Anode at 45° Inclination



Anode at 90° Inclination

Anode at 180° Inclination



Anode at 225° Inclination

Figure 2 Flow Visualisations at $R_e = 250$

lift forces occur when the anode is located on the shoulder of the cylinder ($\alpha = 90°$ or 180°)

The results for a circular cylinder in planar oscillatory flow are dependent upon both the Reynolds number and the Keulegan Carpenter number The vortex shedding process is very complex, as may be appreciated from Figure 4, which shows the flow pattern when time $t = 75D/U_m$, the Reynolds number is 10^5 and the Keulegan Carpenter number is 15 The figure also shows the variation of the in-line force with time and gives the drag and inertia coefficients The variation of the root mean square (rms) in-line force coefficient, $C_{F_{rms}}$, with Keulegan Carpenter number is given in Figure 5 It may be seen that the numerical model tends to overpredict the in-line force, and that the coefficients are more appropriate to Reynolds numbers of the order of 10^4 There is considerable scatter in measured lift coefficients, but the numerical results for the lift coefficient are bounded on ei-

	Re	Angle of Inclination				
		0°	45°	90°	180°	225°
Single Anode (I)	250	1 451	1 137	1 376	0 897	1 158
	10^3	0 898	0 946	1 451	0 928	1 209
	10^4	0 882	0 899	1 621	0 884	1 236
	10^5	0 899	0 884	1 616	0 954	1 263
Single Anode (II)	10^5	0 706	0 998	1 866	0 606	1 421

Table 1 Drag Coefficient factors for Uni-directional Flow with Anodes Attached

	Angle of Inclination	Keulegan-Carpenter Number					
		5	10	15	20	25	30
$\frac{\bar{C}_D(\text{body})}{\bar{C}_D(\text{bare cylinder})}$	0°	0 950	0 890	0 957	0 947	0 967	0 809
	45°	1 107	0 972	0 974	1 170	1 219	0 936
	90°	2 186	1 788	1 675	1 520	1 613	1 338
$\frac{\bar{C}_M(\text{body})}{\bar{C}_M(\text{bare cylinder})}$	0°	0 869	0 758	0 904	1 029	0 698	1 007
	45°	1 077	1 054	1 115	0 944	1 117	1 028
	90°	1 075	0 950	1 089	1 496	0 971	1 515
$\frac{C_{F_{rms}}(\text{body})}{C_{F_{rms}}(\text{bare cylinder})}$	0°	0 870	0 813	0 926	0 943	0 896	0 817
	45°	1 076	1 016	1 021	1 116	1 169	0 930
	90°	1 235	1 373	1 467	1 481	1 458	1 337

Table 2 Force coefficient Factors for Oscillatory Flow with Anodes Attached

ther side by experimental results for Reynolds numbers of the same order[7][11][12]

The rms in-line force coefficients for a cylinder bearing a single anode at a number of angles of inclination, α, at different Keulegan Carpenter numbers and at a Reynolds number of 10^5, are shown in Figure 6 The force coefficients are non-dimensionalised with respect to the cylinder diameter The configuration is based on working drawings of a real anode of sectional area of $0\ 17 \times 0\ 17m$ attached to a cylinder of diameter $0\ 5m$ The flow mechanisms involved are difficult to decipher,

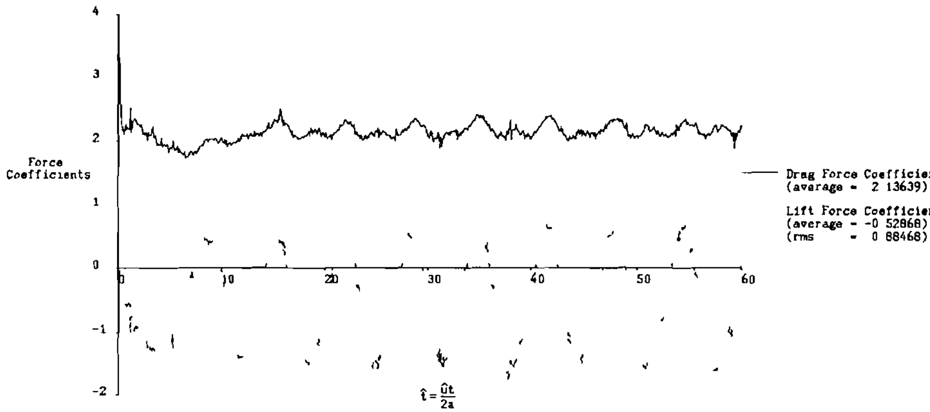


Figure 3 Force Coefficients for Anode at 90° Inclination (R_e = 250)

other than that they tend to follow similar trends to those shown by the anode bearing cylinders in uni-directional flows The largest in-line forces are obtained when the anodes are oriented at right angles to the flow, and the in-line force tends to decrease relative to the bare cylinder when the anodes are in-line with the flow

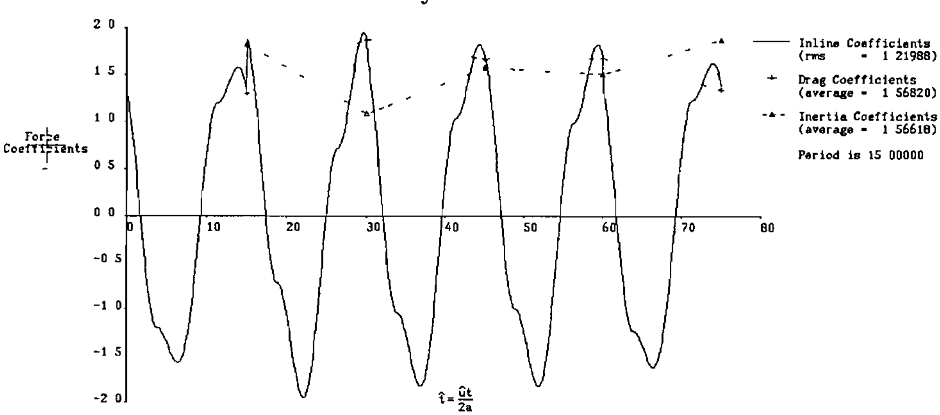
The same behaviour was obtained with a cylinder with two anodes attached to it at 180° to one another, as shown in Figure 7 As might be expected, the rms in-line force coefficient increases with the addition of another anode by around 18% when $\alpha = 45°$ to around 50% when $\alpha = 90°$ There are a few exceptions to these general trends but there appears to be no obvious reason for these

APPLICATION OF RESULTS

As has been mentioned, in assessing the fluid loading on a jacket structure, the Morison equation is used Global values for the force coefficients may well be taken, but a more refined solution may be obtained by choosing force coefficients associated with a characteristic Reynolds number and Keulegan Carpenter number for each member When anodes are fixed to the members such a calculation has to be refined still further to account for the flow asymmetries induced by their presence Each anode will contribute to the in-line force on the member to which it is attached, but it will increase or decrease it depending on whether it is in-line or transverse to the flow

Whilst an anode bearing member could be further sub-divided for the Morison analysis into elements that contain anodes and those that do not, the simplest approach to modifying the force coefficients is to calculate 'equivalent' force coefficients for each complete member In the method adopted the input data file to the Morison analysis software was adapted so as to signify whether, or not, members carried anodes, and what the anode orientation was with respect to a local axis

Oscillatory Flow Visualisation



Oscillatory Flow Force Coefficients

Figure 4 Oscillatory Flow at a Keulegan Carpenter number of 15

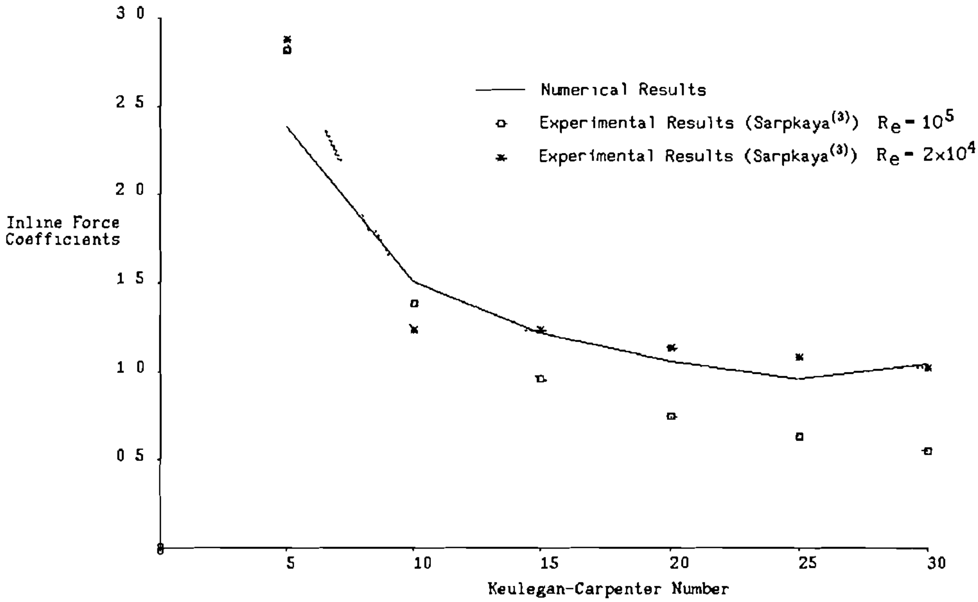


Figure 5 $C_{F_{rms}}$ against K_c for a Circular Cylinder

system The data file was then pre-processed to determine equivalent force coefficients along the two local axes normal to the members using factored force coefficients taken from the numerical study

The total in-line force on each member was apportioned according to the relative lengths of bare and anode bearing member, and to the anode orientation The in-line anodes were treated as a generalised geometry with fixed force coefficients taken straight from the numerical results The numerical transverse drag coefficients were modified so that the relative contributions made by the portion of the cylinder carrying the anode, and the anode itself, were proportioned to their respective projected areas

The equivalent drag coefficient, for example, of a member of length L_m and diameter D_m bearing N_I different sets of n_j in-line anodes and N_T different sets of n_j transverse anodes is given by[10],

$$C_{D\,\mathrm{equiv}} = C_D \Big(1 + \frac{1}{L_m D_m} \sum_{j=1}^{N_I+N_T} n_j L_j \beta_j \big(\frac{C^*_{D_j}}{C_D} - 1\big)\Big), \tag{9}$$

where the alignment factor, $\beta_j = D_m$ for an in-line anode, $\beta_j = D_j/\delta^*_j$ for a transverse anode and C_D is the drag coefficient for a bare circular cylinder The drag coefficient calculated numerically for the j^{th} configuration with the anode to cylin-

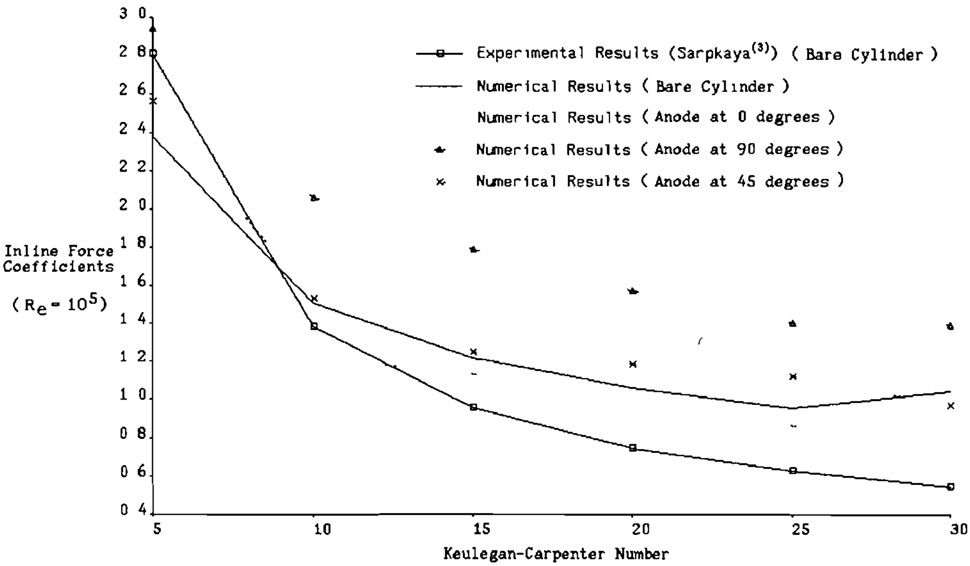


Figure 6 $C_{F_{rms}}$ against K_c with One Anode Attached

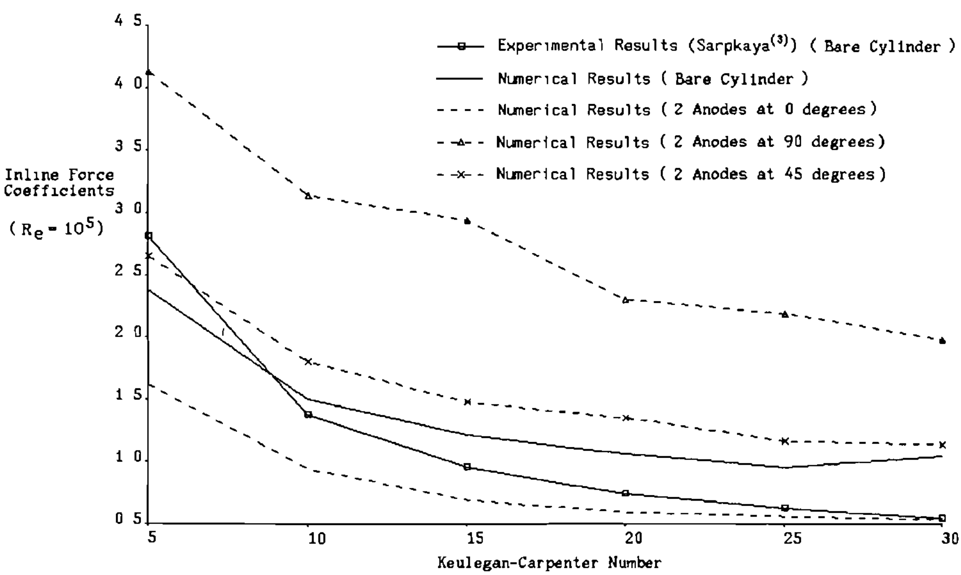


Figure 7 $C_{F_{rms}}$ against K_c with Two Anodes Attached der diameter ratio of δ_j^* is $C_{D_j}^*$ The approach is equally valid with experimental data, and the terms in parentheses could be treated as a multiplication factor to be used with any chosen circular cylinder drag coefficient

Every member in the data file, having been processed, has associated with it two sets of force coefficients appropriate to its local axes The calculation of the local forces F then takes the form

$$\begin{pmatrix} F_2 \\ F_3 \end{pmatrix} = \frac{1}{2}\rho D U_N M_D M \begin{pmatrix} U_1 \\ U_2 \\ U_3 \end{pmatrix} + \frac{1}{4}\rho\pi D^2 M_M M \begin{pmatrix} \dot{U}_1 \\ \dot{U}_2 \\ \dot{U}_3 \end{pmatrix}, \tag{10}$$

where,

$$M_D = \begin{pmatrix} 0 & C_{D_2} & 0 \\ 0 & 0 & C_{D_3} \end{pmatrix}, \quad M_M = \begin{pmatrix} 0 & C_{M_2} & 0 \\ 0 & 0 & C_{M_3} \end{pmatrix}$$

and U_N is the magnitude of the fluid velocity normal to the member and U_i and $\dot{U}_i$ are the local fluid velocities and accelerations in relation to the global axes calculated from linear wave theory The global fluid loading acting on every node of the jacket can be calculated in a straightforward manner from knowledge of the local fluid loading at any point on the structure

CASE STUDIES

Four case studies were carried out on the simple jacket structure shown in Figure 8 carrying a realistic distribution of anodes The structure has 44 nodes, 96 members and 140 anodes and stands approximately $50m$ above the sea bed In all the case studies the anodes were single and oriented in the global vertical plane The case studies were all chosen so that the force coefficients used lay within the data set of the numerical calculations

In the first case study the structure was subjected to a uniform uni-directional current of $1ms^{-1}$ and in the second, to a uni-directional current varying with water depth with a velocity profile given by $u = 1\ 06 - 0\ 24d$, where d varies from 0 at the mean water to 1 at the sea bed The profile is appropriate for parts of the southern North Sea The total base shear was calculated for the structure with and without anodes in both cases For the uniform current, the base shear of the anode bearing structure was 8 8% larger than that of the bare structure The base shear was increased by 8 5% with the current with a velocity profile

The calculation of the hydrodynamic forces on the jacket is more complicated for oscillatory flows, since the force coefficients are functions of the Keulegan Carpen-

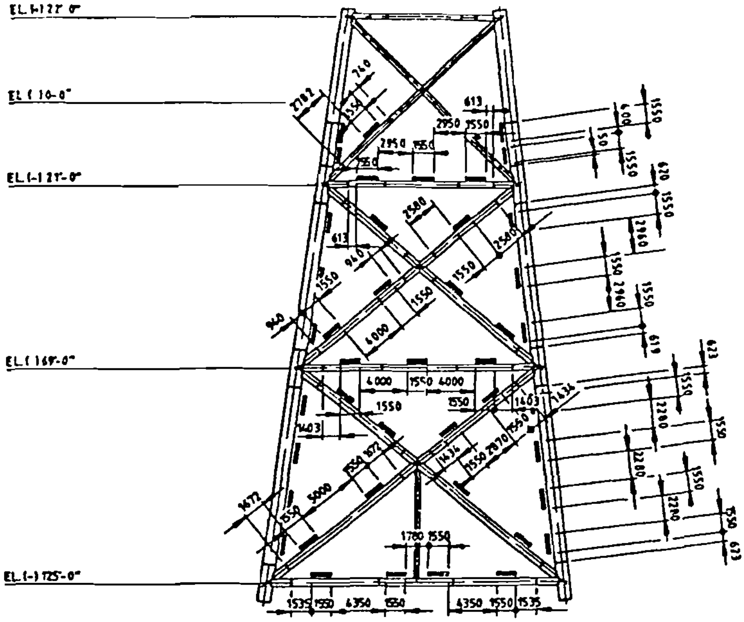


Figure 8 Jacket Structure

ter number, which in turn depends on the local fluid velocity amplitude and the diameter of the member. In order to keep the calculations as simple as possible, flow conditions were chosen so that, for a given wave, the Keulegan Carpenter number remained constant for all members of any given diameter. These conditions occur for a shallow water wave of a period of $45s$ for which the horizontal water particle velocity is independent of elevation and whose amplitude, $\frac{H}{2}\sqrt{\frac{g}{d}}$, depends only on the wave height. By selecting appropriate wave heights, calculations covering the Keulegan Carpenter range from 5 to 30 were carried out. Since a small vertical component of fluid loading is introduced by the wave flow, the comparison between the bare and anode bearing structure was made in terms of the average amplitude of the in-line force over the cycle. The results are given in Table 3.

H (m)	K_c	increase in in-line force
0.5	5 and 10	0.1%
1.0	10 and 20	5.4%
1.5	15 and 30	8.2%

Table 3 Structure Loading in Shallow Water Waves

The final case study was carried out for more realistic conditions, namely for a wave period of $10s$ and wave height $10m$ In this case the horizontal water particle velocity amplitude ranges from about 0 $32Hm/s$ at mean water level to 0 $08Hm/s$ at the sea bed, with an average value of 0 $15H$ The Keulegan Carpenter number used for all members was based on the average value of the velocity, which yielded values of 15 and 30, depending on member diameter The average amplitude of the in-line force over the cycle was found to be 9 2% higher for the anode bearing structure than for the bare structure

CONCLUSIONS

Experimental data on the hydrodynamic loading due to the presence of sacrificial anodes is scarce, a rigorous method of computing it, for all practical purposes, does not exist, and very simple models, such as potential flow models or the use of drag coefficients in unbounded flows, do not give good results The objective of the present study was to gain an understanding of the fluid phenomena and to model the flows as simply as possible in order to determine force coefficients for a variety of flow regimes that could be used to estimate the hydrodynamic loading on a jacket structure The discrete vortex method was used to model the separated flows, which cannot be handled by potential flow models A number of simplifying assumptions were made which allowed the calculation of factors to be applied to force coefficients for circular cylinders leading to coefficients for cylinders with protrusions in uni-directional flow and oscillatory flow for Keulegan Carpenter numbers from 5 to 30

The results for the uni-directional flow showed the same trends as obtained by experiment, in that in-line anodes decreased the force coefficients and transverse ones increased them Similar trends were shown by anodes in oscillatory flow The results for oscillatory flows also showed that as the Keulegan Carpenter number increases, and the drag increasingly dominates the flow, the influence of the anodes becomes increasingly important as shown in Table 3 Finally, it was shown, for the simple structure chosen, and in the context of assumptions made, even fairly modest environmental conditions can lead to a hydrodynamic loading on the structure with anodes that is 9% greater than the structure without anodes It may be assumed that this increase will be significantly larger in the harsher environment to which jacket structures are often exposed

The study indicates that the hydrodynamic loading due to sacrificial anodes is considerably higher than generally estimated from the simple approaches described in the introduction The method of obtaining a more accurate estimate developed in this study could equally well be used with experimental data rather than the numerical results calculated using the discrete vortex method The results obtained in this study justify further work of this nature

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