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VORTEX INDUCED VIBRATIONS IN WIND - DESIGN CRITERIA

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ABSTRACT

A design procedure with criteria for vortex induced vibrations due to wind, based on DIN 4133/Eurocode-1 and adopted for offshore conditions, is introduced. The proposal contains criteria for avoiding vortex induced vibrations and a procedure for controlling fatigue damage. A comparison with some existing design codes is given.

Parameters utilised in current design practice in Norway are reviewed and some recent developments discussed. Emphasis is put on investigating the interrelation between parameters and identifying the key parameters and criteria representing boundaries for avoiding vortex induced vibrations. Special design considerations, like vortex initiated global vibrations (frame vibrations) and wake induced vibrations, are discussed.

This paper may be seen as a follow-up of the paper: "Wind Induced Resonant Cross-Flow Vibrations on Norwegian Offshore Flare Booms", OMAE-95, by the same authors, in which the experienced problem area and a tentative solution strategy were presented. Recent experience with vortex mitigation devices applied on flare booms is included. As an introduction, a short description of the vortex shedding phenomena is given.

PHENOMENOLOGICAL DESCRIPTION

When a fluid, or air, is flowing around a stationary cylinder the flow will separate and vortices are periodically shed in the downstream flow. When vortices are shed from the surface of the cylinder, the local pressure field changes and this results in a force resultant acting on the cylinder varying with the same frequency as the vortex shedding frequency. The process of vortex shedding, with changes in the pressure fields as a function of time, is shown in Fig 1. Mujamdar and Douglas (1973).

At a given critical velocity the frequency of vortex shedding is equal to a multiple of a natural frequency of the cylinder implying resonant vibrations. The vibrations will force the vortex shedding frequency to "lock-in" to the natural frequency for a range of velocities leading to a bandwidth of quasi-resonant vibrations around the critical velocity. The cylinder may be excited resulting in both cross-flow and in-line flow vibrations.

DESIGN PRACTICE REVIEW

Design practice has been based on the observation that if the vibration amplitudes were kept small, the induced fluctuating force would be randomly distributed over a broad band of frequencies. The vortex induced force would thus vary randomly along the length of the member. A broad band response would therefore be favourable and used as a criterion for avoiding adverse effects of vortex induced vibrations. The means to control this were to limit the reduced velocity value as a function of the flow regime (Reynolds number) and with additional requirements to the so-called stability parameter (Scruton number) if the reduced velocity requirement could not be met. The effect of vortex shedding was considered negligible for values of Scruton number larger than a certain value. In Statoil (1985) the limiting Scruton number was specified to be Sc= 16. Ref. Statoil (1985), DNV (1991) and ESDU (1985).

After experiencing vibration problems at the Heimdal platform in 1984-85, the Statoil design specification was revised in 1990 shortly followed by a revised NPD guideline. The vibration amplitude was made a function of the reduced velocity level for peak vibration and the stability parameter used without any limitation, Moe (1989). The specification was considered conservative for critical and postcritical flow.

The procedures are, in general, found to overestimate the effect of damping and underestimate the fatigue effect due to high natural frequencies of the structural members, i.e. number of load-cycles. Combined with a high probability of critical velocities, as seen in the long term wind distribution, a number of cracks have been observed, Oppen and Kvitrud (1995).

REFERENCE STANDARDS

The paper is based on DIN 4133 (1991) as a reference standard. For in-line flow vibrations, reference is made to DNV Classification Note No. 30.5 (1991). Further reference may be found in Eurocode-1 (ENV 1991-2-3, 1993) recently issued. Related to vortex induced vibrations, the Eurocode is based on the same principles as DIN 4133 and is foreseen to be used as a reference standard with basically the same modifications, Oppen (1995).

DESIGN PRINCIPLES

The structural design against vortex induced vibrations should be based on the following principles:

- If found economically feasible, the design should aim at avoiding vortex induced vibrations using defined avoidance criteria.
- Alternatively, the design should be controlled by fatigue analysis. Accumulated fatigue damage from vortex induced vibrations and global dynamics should be combined using Miners sum. Expected fatigue life, vibration amplitudes and

probability of occurrence of vibrations should be documented and satisfy operational requirements.

In special cases, a design with vortex reducing devices may be considered e.g. for cases with unacceptable vibrations in a fatigue controlled design or a fatigue exposed design in temporary construction phases.

CRITERIA FOR AVOIDING RESONANT VIBRATIONS

In the following, parameters related to vortex induced vibrations and frequently used in current design practice are discussed.

Basic Assumptions and Critical Velocity

The relation between the frequency of vortex shedding, wind velocity and diameter is given by the Strouhal number.

$$F = St \frac{V}{D} \tag{1}$$

A critical velocity is defined as the velocity giving vortex shedding frequencies equal to the natural frequency of the structural member.

$$Vc = \frac{1}{St} Fn\, D \tag{2}$$

The frequency of vortex shedding is maintained at the level of the natural frequency for a range of velocities around Vc (lock-in) resulting in a state of quasi-resonant vibration.

$$K_1\, Vc < V < K_2\, Vc \tag{3}$$

This implies that, in order to avoid vortex induced resonant vibrations, the following criterion results.

$$Vc > \frac{1}{K_1} V_m \tag{4}$$

The maximum wind velocity should be based on the prescribed design wind, eventually with a limitation in order to account for the effect of up-stream turbulence, and the K-factors selected through experiments or failure statistics.

The Strouhal number is known as a function of the flow regime represented by the Reynolds number,

$$St = f(Re) \tag{5}$$

where

$$Re = \frac{VD}{\nu} \tag{6}$$

In Fig. 2 the Strouhal number as a function of the Reynolds number is given, DNV Classification Notes - No. 30.5 (1991).

The natural frequency is given by the expression

$$Fn = \frac{a^2}{2\pi L^2} \sqrt{\frac{EI}{m}} \approx \frac{(1.59\,\phi + \pi)^2}{2\pi\, L^2} \sqrt{\frac{EI}{m}} \tag{7}$$

The factor a may be given as a function of the degree of end fixities ϕ having values between 1.0 (clamped) and 0 (hinged). The formula can be derived from the equation of motion for free vibration as the solution to the eigenproblem and may be found in general textbooks.

Eq. (7) give an approximation for the first mode natural frequency for an element with an equal partial end fixity at both ends assuming a linearity of factor a between values for clamped and hinged supports. The second mode of vibration may be investigated considering an element with length half the system length and one end partially clamped, the other end hinged.

Expanding the expression for Vc in Eq. (2) as a function of D/L and t/D results in

$$Vc = \frac{1}{St} \left(\frac{D}{L}\right)^2 \frac{a^2}{2\pi} A \sqrt{\frac{E}{8\rho_1}} \tag{8}$$

where

$$A = \sqrt{1 - 2\frac{t}{D} + 2\left(\frac{t}{D}\right)^2} \approx 1 - \frac{t}{D} \tag{9}$$

In Fig. 3 the critical velocity for the first mode of vibration, expressed as Vc/A, is given as a function of the ratio L/D and in Fig. 4 the factor A is given as a function of the ratio t/D. The formulas might be suited for design purposes.

Reduced Velocity

The reduced velocity is given by the relations

$$Vr = \frac{V}{FnD} \tag{10}$$

Inserting this and Eq. (2) in Eq. (3) results in the boundaries

$$K_1 \frac{1}{St} < Vr < K_2 \frac{1}{St} \tag{11}$$

This implies that the criterion for avoiding resonant vibration would read

$$Vr = \frac{Vm}{Fn\, D} < K_1 \frac{1}{St} \tag{12}$$

Scruton Number

Scruton number (also denoted the stability parameter) is given by the relation

$$Sc = \frac{2m\delta}{\rho D^2} \tag{13}$$

Inserting

$$m = \pi\left(\left(\frac{D}{2}\right)^2 - \left(\frac{D}{2} - t\right)^2\right)\rho_1 \tag{14}$$

and expanding in terms of t/D give

$$Sc = 2\pi \frac{\delta\rho_1}{\rho} \left(\frac{t}{D} - \left(\frac{t}{D}\right)^2\right) \tag{15}$$

Sc is seen to be a quadratic function of t/D, for small values small values approximately a linear function, i.e.

$$Sc \approx 2\,\pi \frac{\delta\rho_1}{\rho} \frac{t}{D} \tag{16}$$

In Fig. 5, Sc is given as a function of t/D for a value of damping equal to 0.15 % of critical.

In the expression for Vc, Eq. (8), t/D can be converted to Sc. It is noted that there is an incentive to reduce the values of Sc (i.e. t/D) thereby increasing the critical velocity. A limitation of t/D based on requirements for local structural stability and wall thickness t for welding quality will have to be recognised.

It is concluded that Sc does not represent a boundary for avoiding resonant vibration. Sc may be used indirectly to limit the value of t/D in order to secure local stability and welding quality. However, if resonant vibration cannot be avoided, a large value of Sc will decrease the vibration amplitudes and thereby reduce a fatigue damage.

Length over Diameter

As seen from Eq. (8), the critical velocity is a quadratic function of L/D. Hence, L/D may be used as a practical reference parameter for avoiding resonant vibrations pending values of t/D, end fixations and Strouhal numbers.

Summary of Criteria for Avoidance of Resonant Vibration

A major parameter governing the resonant cross-wind vibration is found to be the critical velocity or, alternatively, the reduced velocity. The criteria are related to the basis for calculating the wind velocity i.e. the averaging time and return period. Furthermore, the flow regime will have to be defined in order to obtain the relevant frequency of vortex shedding and a range of velocities around the critical velocity will have to be specified accounting for the lock-in effect. For the time being the effect of up-stream turbulence on a lift force cut-off level is not taken into account.

Generally, the ratio t/D, or consequently the Scruton number, will have to be limited in order to secure local stability of the pipe section and wall thickness for welding quality.

The following lower bound criteria for avoiding cross-flow vibrations, corresponding to parameter K_1, are proposed

- Critical velocity:

$$Vc = \frac{1}{St} Fn\, D > \frac{1}{K}\, Vm \qquad (17)$$

where:
Factor, K = 0.85
Strouhal number,

$St = 0.2$ when $Re \leq 4 * 10^5$
$St = 0.24$ when $Re > 4 * 10^5$

Maximum wind speed, Vm = V(30, 100, Z), as the 30 s mean wind with a 100 year return period at the relevant height level above mean sea level.

- Reduced velocity (Alternatively):

$$Vr = \frac{Vm}{Fn\, D} < K\, \frac{1}{St} \qquad (18)$$

where values of K, St, Vm are given above.

The criteria are based on Strouhal numbers given by Fig. 2 and simulations with offshore flare boom elements from Table 2. The criteria are considered conservative and are discussed later. For temporary construction phases, the maximum velocity could be based on the relevant wind distribution and return periods.

FATIGUE ANALYSIS

If the criteria for avoiding cross-flow resonant vibration cannot be met a fatigue analysis will have to be performed. Reference is made to DIN 4133, and to the list of notations.

Introduction

A wind speed range around the critical velocity is considered,

$$Vc = \frac{1}{St} Fn\, D \qquad (19)$$

$$\Delta\, Vc = B\, Vc \qquad (20)$$

The maximum amplitude of vibration is calculated,

$$Ym = Kb\, Kw\, Cl\, \frac{1}{St^2}\, \frac{1}{Sc}\, D \qquad (21)$$

The mode shape factor, Kb, the correlation length factor, Kw, and the lift force factor, Cl, as obtained from DIN 4133.

The number of vibration cycles according to DIN 4133 is assumed to be:

$$n = Td\, Fn\, P(\Delta\, Vc) = 10^9\, Fn \left(\frac{Vc}{Vo}\right)^2 e^{-(Vc/Vo)^2} \qquad (22)$$

$P(\Delta\, Vc)$ is the probability of wind speeds within the critical velocity range, Fn is the natural frequency of vibration and Td the design life. A 2-parameter Weibull distribution for the wind speed is assumed and the formula in DIN 4133 is presented.

The capacity of the section for load cycling is obtained from standard S/N-curves for the given stress range,

$$N = f(Sr) \qquad (23)$$

The stress range is derived from the maximum amplitude of vibration, assuming a function of deflection, and relevant stress concentration factors (SCF's).

Using the Palmgren-Miner sum, the damage ratio and fatigue life is obtained as

$$DR = \Sigma \left(\frac{n_i}{N_i}\right) \qquad (24)$$

$$Tf = \frac{Td}{DR} \qquad (25)$$

Proposed modifications to DIN 4133 for offshore applications.

DIN-4133 (Zone IV) is based on the following parametric values:

- Critical velocity limit for vortex shedding vibrations, giving a lift force cut-off value, is Vc = 40 m/s.
- Strouhal number, St = 0.2.
- Critical velocity range, B = 0.3 (implicit).
- Design life, 50 years (Td in sec.).
- Weibull parameters, a value of C=2 is selected and Vo = 7 m/s is recommended as an approximate value.

The following changes and amendments are proposed:

- Lift force cut-off value for

$$Vc = \frac{1}{K}\ Vm$$

Values of Vm and K as defined for the avoidance criteria. For temporary construction phases, the maximum velocity may be based on the relevant wind distribution and return periods.

- Strouhal number as a function of Reynolds number.
  $St = 0.2$ when $Re \leq 4 * 10^5$
  $St = 0.24$ when $Re > 4 * 10^5$
- Design life according to project specification. When utilising a fatigue controlled design, the operational tolerances of vibration amplitudes and probabilities of occurrences should be considered.
- Weibull parameters based on a site specific, longterm, wind distribution. The Weibull scale parameter to be related to the actual height above sea level.
- The effect of wind direction not perpendicular to the member axis may be taken into account i.e. reducing the number of load cycles. The reduction could be based on an annual wind direction frequency distribution. Local wind concentrations or change in direction should be evaluated.
- The stress cycling capacity might be obtained using the S/N-curves of NS 3472, or equal.
- Damping, 0.15 % of critical if not otherwise substantiated, ref. Oppen and Kvitrud (1995).

SPECIAL DESIGN CONSIDERATIONS

In-line Flow Vibrations

In-line flow vibrations are assumed to be avoided for values of Scruton number Sc > 1.8, DNV Classifications Notes - No. 30.5 (1991). A Sc limitation may be questioned on the same basis as for cross-flow vibrations, however, in-line flow vibrations are not considered to be a problem for normal structural elements.

Wake Interactions

Wake interactions should be considered for distances less than 15 times the diameter of the larger (up-stream) tube if the vortex shedding frequency from the upstream tube is higher than 85 % of the natural frequency of the smaller (down-stream) tube.

Frame Vibrations

A frame is defined as a composition of 2 or more members. The vibration may be local, confined to a substructure, or a global dynamic vibration driven by vortex induced vibrations. Possible vibrations may be investigated with a dynamic analysis using a single element vibration as an excitation source.

Local frame vibrations should be investigated when a free span member is supporting other members which are susceptible to vortex induced vibrations. Global dynamic vibrations should be investigated for the case that a vibrating larger member could serve as an excitation source, e.g. a chord member of a truss.

Other Considerations

For slender elements the effect of the 2. order mode of vibration should be investigated. The results of a simulation, using a tube with length 4 m, wall thickness 0.01 m, end fixities 0.7 and with varying diameter, are shown in Fig. 6. Indications are that the critical L/D range is somewhat 60 % higher when the 2. order mode of vibration is taken into account. Utilising slender elements will have to be given special consideration in the conceptual design phase.

Further, the dynamic effect of buffeting in the wind direction should be considered.

VIBRATION REDUCING MEASURES

Measures for reducing vortex induced vibrations generally falls into the following categories:

- Increase the stiffness in order to avoid critical wind velocities.
- Disturb the shedding of vortices and thereby prevent local vibrations.
- Increase damping in order to reduce the amplitude of vibrations.

The effect of vibration reducing devices should be demonstrated with reference to full scale observations or model testing. For temporary construction phases, conventionally designed measures, e.g. mechanical vibration absorbers, spoilers and strapping, may be considered.

A concept of vortex mitigation light-weight sleeves has been subjected to model testing and used on the flare booms of Statfjord B and C and Gullfaks B, Oppen and Kvitrud (1995). The platforms STB, STC and GFB were installed in 1982, 84 and 87 respectively. By 1993, the flare booms had suffered vortex shedding fatigue cracks in a number of 22, 30 and 12 when vortex mitigation sleeves were installed. The flare booms were inspected in June 1995 and one new crack was observed on Statfjord B. The elements have an accumulated fatigue damage and predictions of fatigue lives are uncertain. Analytically, the effect of the sleeves are higher on the GFB flareboom, by a factor of 5, than on the STB or STC flare booms for fatigue sensitive elements. It may be concluded that the effect of the sleeves still looks promising.

DISCUSSION

The vortex avoidance criteria, as related to Vc or Vr, depend on the selection of a mean wind speed with a given probability of occurrence i.e. return period. Based on a 2-parameter Weibull distribution for the yearly 1 hour mean wind speed, the maximum wind speed is selected as the most probable largest velocity at the relevant level above MSL with a given averaging time and return period. Existing standards and specifications use averaging times varying from 30 s to 1 hour and return periods of 50 and 100 years, ref. Table 1. The averaging time should be short enough to capture the physical ability of the structure to respond with resonant vibrations. Since the natural frequencies of structural elements are quite high, a rather short averaging time is expected. Field observations from the Statfjord A flareboom indicate resonant vibrations for a wind speed with an averaging time of about 30 s and natural frequency of about 19 Hz, Oppen and Kvitrud (1995). A return period of 100 year is preferred since this return period is typically selected as the design period for offshore structures in Norwegian waters.

The critical wind velocity may be limited due to up-stream turbulence. However, the available literature gives scarce information on the limitation of resonant vibration due to turbulence. Documentations of the Strouhal number as a function of the Reynolds number, giving information of the frequency of vortex shedding, are mostly assuming laminar wind. DIN4133 (Zone IV) prescribes Vc = 40 m/s as a cut off level for vortex shedding effects. The maximum velocity, based on the given wind pressure 10 m above MSL and an averaging time of 5 s and a return period of 50 years, can be estimated to about 52.4 m/s. The velocity is thus reduced with a factor of about 0.76. However, one field observation from Statfjord A indicates resonant vibrations for values of critical velocities up to 65 m/s (using St = 0.2), Oppen and Kvitrud (1995).

The formula for maximum amplitude of vibrations, Eq. (21), can be derived from the equation of dynamic motion involving the solution for damped forced vibration, i.e. steady state vibration. It assumes equivalent structural damping and contain some simplifications, however, the most important parameters are found to be covered. A preliminary evaluation is found in Moe (1991).

The probability of wind velocities within the critical velocity range is based on an observed long-term distribution. It may be argued that a distribution will not give the actual "visits" of wind in the critical range. However, the model assumes that "resonant" vibration is governed by some average wind velocities. Wind at increased heights is taken into account by increasing the Weibull scale parameter based on conventional logarithmic formula. Correcting the mean wind probability distribution for a reduced averaging time should be further investigated, as well as the effect of local wind concentrations and also the effect of wind directions not being perpendicular to the member axis. The wind direction in a sector of say 45 deg. perpendicular to the member axis, based on a long-term wind frequency distribution, could be taken into account, corresponding to an assumed cosine function distribution effect. With reference to Fig 10, this may result in a 40 % reduction of the probability for critical wind occurrence.

Use of the formulas will imply that fatigue can be avoided for low aspect ratios L/D but also for high values of L/D, i.e. small diameters. A physical understanding of this may be obtained by studying the forcing term in the equation of motion for forced damped vibrations where c is the damping term:

$$c\,\frac{dY}{dT} = F(T) = \frac{1}{2}\,Cl\,\rho\,DV^2\cos(\omega T) \qquad (26)$$

As seen from Eq. (7) and (8), a reduced diameter will give a reduced natural frequency and a reduced critical velocity. Following the equation of motion a reduced diameter will thus give a reduced amplitude of vibration. The effect on the fatigue life is then caused by an increased capacity for load cycling resulting from the reduced stress range. The number of load cycles will further be reduced resulting from the reduced frequency of vibration and probability for staying in the critical velocity range. The boundaries are sensitive to parameters influencing the stress range and the natural frequency, i.e. t/D, SCF and degree of element end fixities. Utilising high values of L/D will imply a reduced capacity for axial loads and occurrences of cross-flow vibrations during periods of the service life. As shown, slender elements may also be susceptible to excitation of higher order (2.) modes of vibration.

Indications are that the DIN-formula underestimate the amplitude of vibration. In Fig. 7 the result of a simulation giving the maximum amplitude over diameter as a function of the Scruton number is presented. The values are obtained according to DIN 4133, an earlier Statoil specification, Statoil (1985), and a formula adopted at MIT. The MIT-formula has been substantiated with test results, Rudge and al. (1992), Vandiver and Fei (1995). Simulation (Fig. 7, left) using the MIT carbon fibre pinned-pinned test element with varying wall thickness cover a range of low Scruton numbers in subcritical flow. Simulation (Fig 7, right) using offshore related elements as given in Table 2, Oppen and Kvitrud (1995), results in a less pronounced difference between the formulas, for higher Scruton numbers the DIN-formula results in even higher values.

Table 1 presents a comparison between the proposal and some existing standards and specifications with basic assumptions and criteria for avoiding resonant vibrations. The table contains resulting requirements for critical velocities for two numerical cases.

Applying the proposed criteria on offshore related elements as given in Table 2, Oppen and Kvitrud (1995), indicate an incentive for using a fatigue controlled design over the avoidance criterion. Fig. 8 presents values of Vc/Vm for the existing design and with diameters giving 100 yrs. fatigue life. The avoidance criterion states Vc/Vm = 1/0.85 = 1.18.

CONCLUSIONS

Fatigue and displacement of tubular elements caused by vortex induced vibrations due to wind is considered. The design should preferably aim at avoiding vortex induced vibrations. Boundaries based on critical velocities may be utilised. An alternative is a fatigue controlled design.

Accumulated damage from vortex induced vibrations and (global) dynamic analysis should be combined using Miners sum.

In special cases, e.g. temporary construction phases, a design with vortex reducing devices may be considered.

A design procedure and criteria is proposed. Generally, the criteria should be adjusted according to relevant model testing or full scale observations.

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FIGURES

1. The vortex shedding process.
2. Strouhal number for a circular cylinder as a function of the Reynolds numbers.
3. The critical velocity as a function of the ratio L/D.
4. Factor A as a function of the ratio t/D.
5. Scruton number as a function of the ratio t/D.
6. Fatigue life and damage ratio as a function of the ratio L/D - Simulation of first and second order mode of vibration.
7. Vibration amplitude as a function of the Scruton number - Simulation with test element and offshore flare boom elements.
8. Values of Vc/Vm - Offshore flare boom elements designed and redimensioned to 100 yrs. fatigue life.

TABLES

1. Comparison of vortex shedding avoidance criteria.
2. Geometry parameters and fatigue response.

NOTATIONS

A	Factor, critical velocity.
a	Factor, natural frequency.
B	Fraction of Vc as critical velocity range.

C	Weibull shape factor.
Cl	Aerodynamic lift force factor.
D	Pipe outer diameter.
DR	Fatigue damage ratio.
E	Modules of elasticity.
F	Frequency of vortex shedding.
Fn	Natural frequency of a structural member.
I	Moment of inertia.
Kb	Mode shape factor.
Kw	Correlation length factor.
Kz	Altitude wind speed factor.
K_1	Factor of lower bound lock-in.
K_2	Factor of upper bound lock-in.
L	Length.
m	Mass pr. length.
N	Capacity cycles.
n	Load cycles.
$P(\Delta Vc)$	Probability of critical wind range.
Re	Reynolds number.
Rp	Return period.
Sc	Scruton number.
Sr	Stress range.
St	Strouhal number.
t	Pipe wall thickness.
T	Time.
Td	Design life time.
Tf	Fatigue life time.
V	Wind velocity.
Vc	Critical velocity.
Vm	Maximum (design) wind velocity.
Vo	Weibull scale factor.
Vr	Reduced velocity.
Y	Amplitude of vibration.
Z	Height above mean sea level (MSL).
ΔVc	Range of critical velocities (lock-in).
δ	Damping, log. decrement.
ν	Kinematic viscosity (air).
ρ	Density of fluid (air).
ρ_1	Material density.
ϕ	Degree of fixity (1: Clamped, 0: Hinged).

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Figures

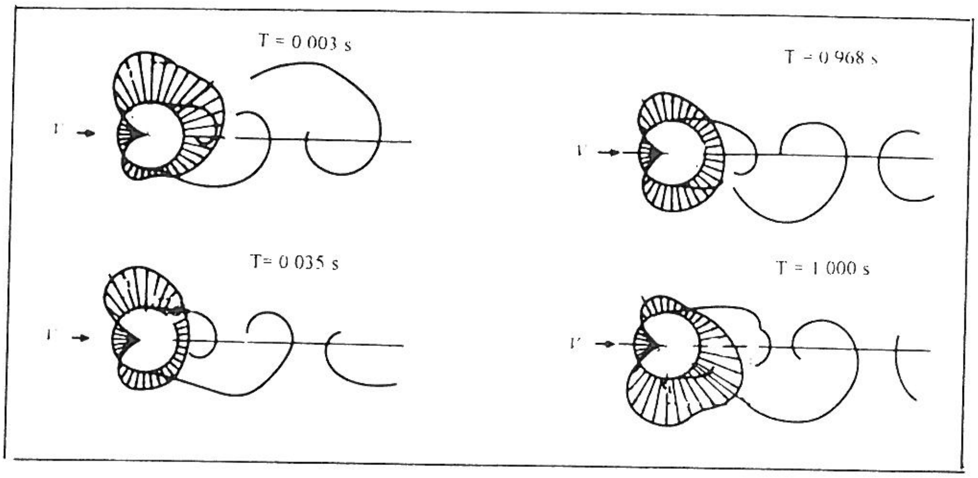


Fig. 1 **The vortex shedding process.**
Ref. Mujamdar and Douglas (1973)

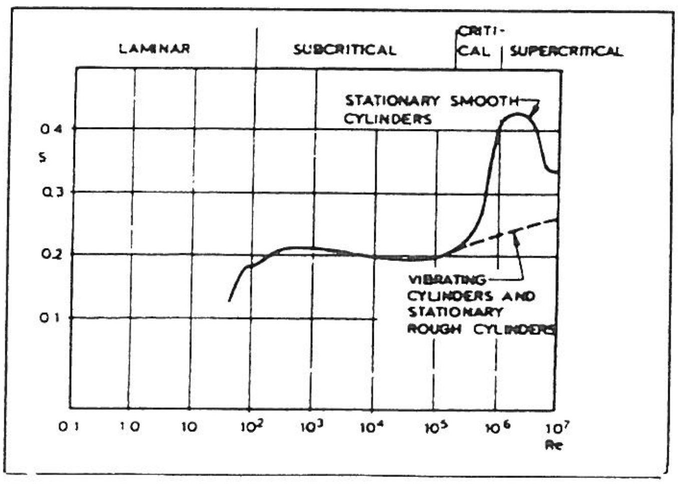


Fig. 2 **Strouhal number for a circular cylinder as a function of Reynolds number.**
Ref. DNV Classification Notes - 30.5 (1991)

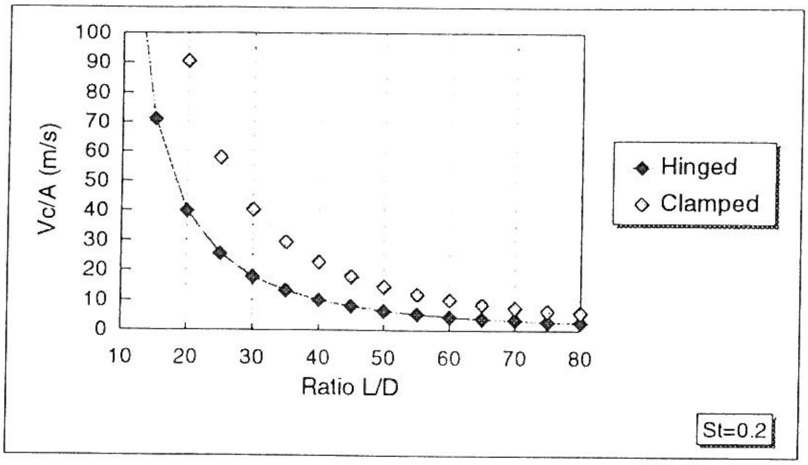


Fig. 3 **The critical velocity as a function of the ratio L/D.**
Ref. Equation (8) / Oppen (1995)

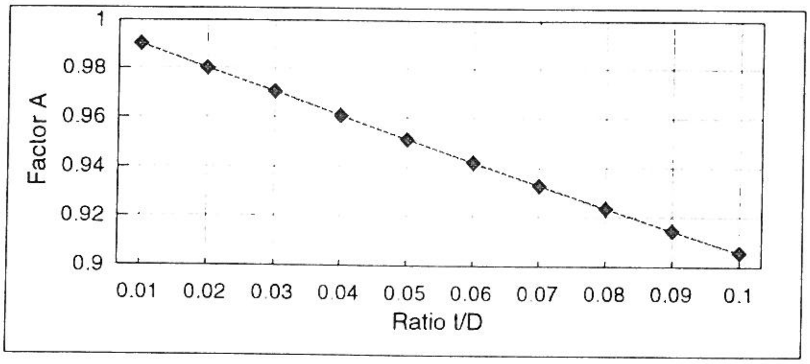


Fig. 4 **Factor A as a function of the ratio t/D.**
Ref. Equation (9) / Oppen (1995).

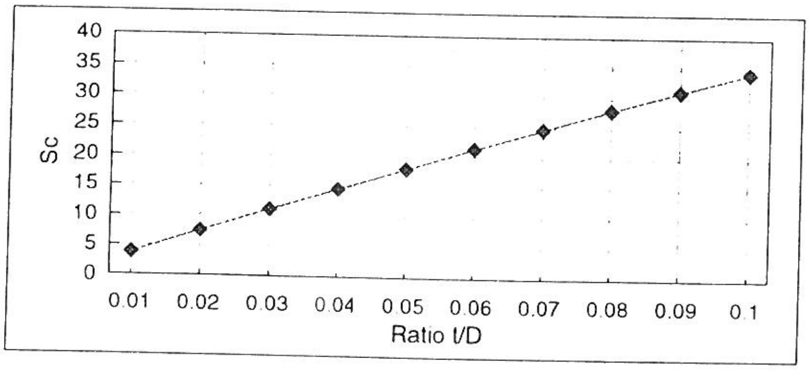
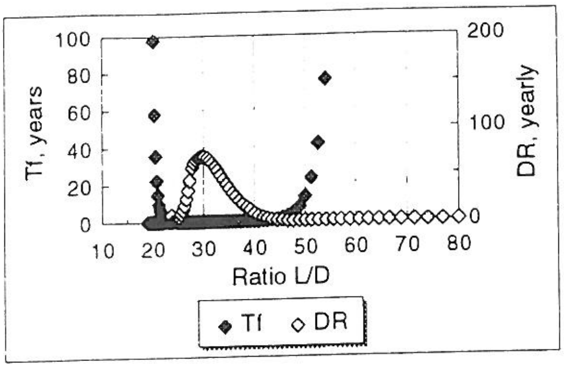


Fig. 5 **Scruton number as a function of the ratio t/D.**
Ref. Equation (15) / Oppen (1995)



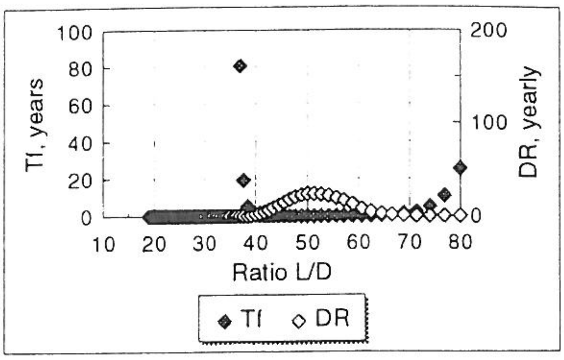
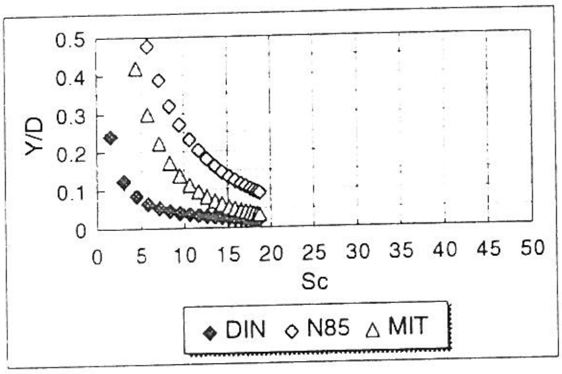


Fig. 6 Fatigue life and damage ratio as a function of the ratio L/D - Simulation of first (left) and second (right) mode of vibration.



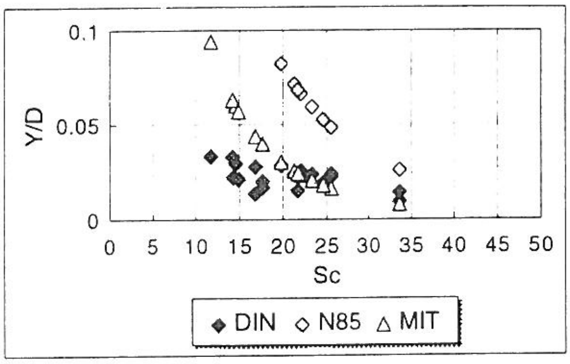
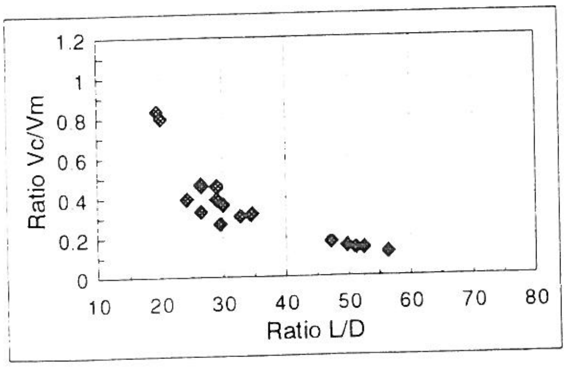


Fig . 7 Vibration amplitudes as a function of the Scruton number - Simulation with test element (left) and offshore flare boom elements (right).



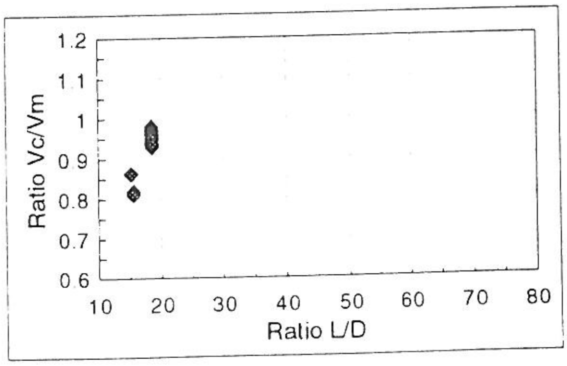


Fig. 8 Ratio Vc/Vm - Offshore flare boom elements designed (left) and redimensioned to 100 yrs. fatigue life (right).

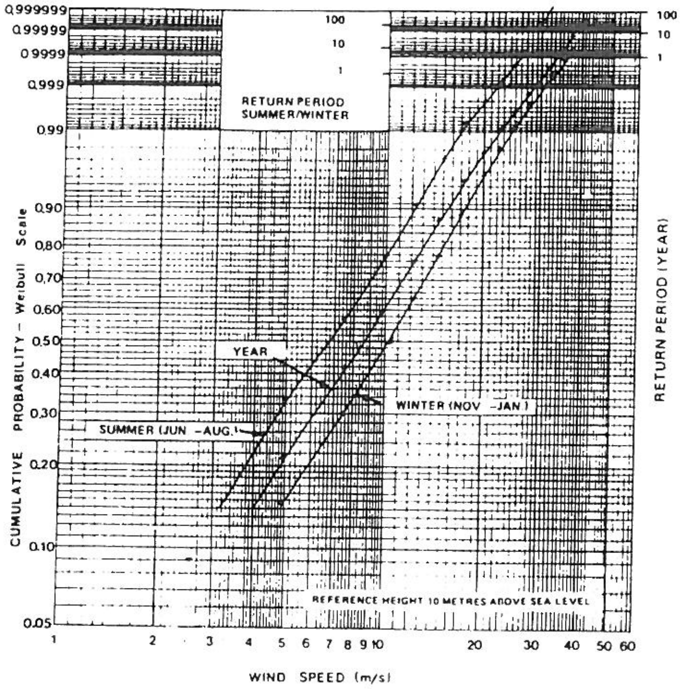


Fig. 9 **Cumulative distributions for the 1 h wind speed.**
Ref. Oppen and Kvitrud (1995)

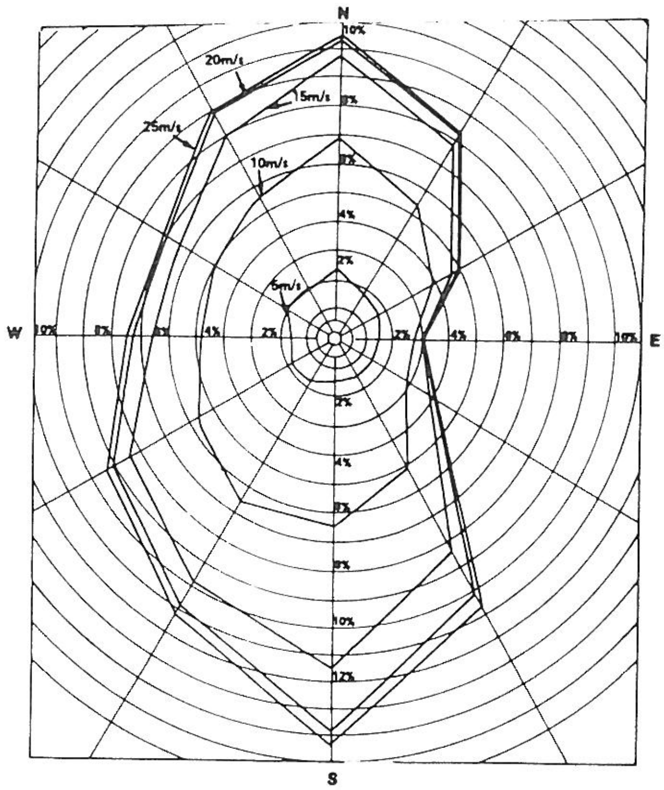


Fig.10 **Annual wind frequency distribution.**
Ref. Oppen and Kvitrud (1995)

Tables

Table 1. Comparison of vortex shedding avoidance criteria.

	Basis			Criteria				Testcases 2) Vc (% of DIN)	
STANDARD	T(s)	Rp(yrs)	St	Sc>	Vr<	Vm/Vc<	Vc(m/s)>	Z1	Z2
DIN 4133	5	50	0.2				40*	100	100
EC-1	600	50	0.2		4.0	0.8*		175	189
BS 8100	3600	50	0.2		3.8	0.77*		142	154
DNV-30.5	60	50	3)	25*	4.7*	0.94		136	146
AE/SLA	30	100	0.2				Kz40*	122	131
DNV/STB	30	100	0.2		5.0	1.0*		135	145
Proposal	30	100	1)		1)	0.85*		160	170

Notes:

* :	Explicit criterion (or inverse value)
Kz:	Wind speed factor (above 10 m from MSL)
1) :	St=0.2 when Re<4*10^5, Vr=4.25
	St=0.24 when Re>4*10^5, Vr=3.54
2) :	Weibull parameters, Vo=10 m/s, C=2
	Height level, Z1=40 and Z2=80 m
3)	St=f(Re)

Table 2. Geometry Parameters and Fatigue Response.
Ref. Oppen and Kvitrud (1995)

	L [m]	D [m]	t [m]	N [hz]	Sc	Vc [m/s] 1)	'Re 2)	FL[yrs} 3)	Observations
STATFJORD A									
1. Main tie chord	5.8	0.2191	0.0159	19	26	21	307	OK	Vibrations
2. Main tie brace-hor.	3.312	0.1143	0.0063	52	20	30	226	<1	Vibrations, possibly
STATFJORD B/C									
1. Wind strut chord	5.3	0.2191	0.0159	23	26	25	368	OK	None
2. Wind strut brace-hor.	2.588	0.0889	0.0055	66	22	29	173	<1	Cracks
3. Wind strut brace-vert.	3.29	0.168	0.01	77	21	65	725	OK	Crack, indication
GULLFAKS B									
1. Wind strut chord	4.5	0.1524	0.01	22	23	17	173	OK	Crack, indication
2. Wind strut brace-hor.	5.2	0.1016	0.01	18	34	9	62	10	Cracks
3. Wind strut brace-vert.	3.5	0.1016	0.01	40	34	20	136	<1	Cracks
HEIMDAL									
1. (290-314)	15.2	0.273	0.0078	6	10	8	150	<1	Vibrations, single
2. (313-314)	13	0.273	0.0078	8	10	11	205	<1	Vibrations, frame
3. (313-294)	11	0.219	0.0103	9	17	10	146	<1	Vibrations, frame
4. (294-292)	17	0.324	0.0084	6	10	9	201	<1	Vibrations, frame
ODIN									
1. (43-52)	7.25	0.273	0.0127	26	17	36	648	<1	Cracks
2. (43-42)	5.5	0.273	0.0127	45	17	62	1128	OK	Cracks
3. (23-20)	9.73	0.324	0.0127	17	14	28	607	<1	Cracks
4. (33-30)	8.55	0.323	0.0127	22	14	36	778	<1	Cracks
5. (33-40)	9.45	0.273	0.019	15	25	21	373	<1	Cracks
VALHALL PCP									
1	13.32	0.406	0.013	12	12	24	642	<1	Cracks
2	9.39	0.324	0.013	19	15	30	651	<1	Cracks
3	3.07	0.406	0.016	218	14	442	11988	OK 4)	Cracks

Notes:
1. Critical velocity, Vc=1/St*N*D, St=0.2
2. Reynolds number at critical velocity in thousands
3. Fatigue life of single element joints in years; indicative
4. DIN 4133 not applicable; k-brace.