FIV for Subsea Production Systems

Part 1: challenges and case example

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What is a Subsea Production System (SPS)?

Shell Perdido SPS, Brazil, 2900 m water depth
Why are we concerned about vibrations from the flow?

High-cycle fatigue: long term risk of structural damage due to small amplitude cyclic motions of the piping and equipment

Flow-induced transient forces

Cyclic structural response (vibration)

Cyclic stresses applied at hot spots

Fatigue cracking $\rightarrow$ crack growth $\rightarrow$ fatigue failure

Source: Xodus Group
Types of flow-induced transient forces

**Vortex-Induced Vibration (VIV): vibration of long spans of piping due to the external sea water current**

One mechanism: sea current flowing around the circular cross-section of the piping. Vortices are shed alternately in the wake, inducing pressure oscillations at the pipe external surface.

Technical solution: break coherence of the vortex shedding along the pipe length by adding helical “strakes”.

Source: VIV Solutions LLC
Types of flow-induced transient forces

Flow-Induced Vibration (FIV): vibration of piping and equipment due to the internal flow of hydrocarbons, chemicals, water, etc…

Multiple mechanisms
- Acoustic pulsations in gas filled systems
- Multiphase flow excitation at bends in the piping
- Fast closing valves in liquid system
- Vortex shedding around intrusive elements
- Etc…

Example: flow-induced pulsations through corrugated piping
- Known as “singing riser” issue
- Cause: flow-induced pressure perturbations
- due to the cavities in the piping walls

Source: TNO
Types of flow-induced transient forces

**Example: flow past a closed side branch ("dead leg")**

Vortices are formed at the upstream edge of the side branch opening, with a frequency $F_{\text{fluid}} = \frac{S_r U_o}{D}$ where $D$ is the side branch diameter and $S_r$ is the Strouhal number.

The vortex formation and impingement on the downstream edge creates an oscillating pressure field in the closed branch.

If the frequency of the perturbation matches one of the acoustic modes, acoustic resonance will produce high pressure pulsations inside the closed branch.

Acoustic natural frequencies: $F_n = c \frac{(2n-1)}{4L}$ where $c$ is the speed of sound and $L$ is the length of the side branch.

Source: TNO
Types of flow-induced transient forces

Example: multiphase excitation of pipe bends

Source: FMC Technologies
**Definition of vibration**

Mechanical oscillation about an equilibrium position, i.e. repeating variation of a certain measure about a central value or state.

**Vibrations are caused by dynamic forces: structural dynamic problem**

Differences with a structural static problem are

- The time-varying nature of the excitation
- The inertial forces in the system affect the response.

![Beam loaded by a static force](image1)

![Beam loaded by a dynamic force](image2)
A structural system will vibrate according to its eigenmodes

- An eigenmode is a pattern of motion in which all parts of the system move sinusoidal with the same frequency and with fixed phase shift. The corresponding frequencies of eigenmodes are called eigenfrequencies or natural frequencies.

A mode is the combination of

- A deflection shape
- A frequency of vibration.

Example: cantilever beam modes 1 and 2

Excitation at 19 Hz

Excitation at 120 Hz

Source: Tech Damper AS
Forms of vibration

**Periodic vibration**

The amplitude is repeated after a discrete time period
The amplitude can be decomposed as a series of sine waves

**Harmonic vibration**

The amplitude corresponds to a single sine wave
Can only occur if the excitation force is at a single frequency

**Non periodic**
General equation of motion: \( m\ddot{x} + c\dot{x} + kx = Fe^{i\omega t} \)

where

- \( x \): displacement
- \( m \): mass
- \( k \): spring stiffness
- \( \omega \): excitation frequency
- \( F \): excitation force
- \( c \): viscous damping

Free undamped vibration: \( m\ddot{x} + kx = 0 \)

In free vibrations the system will vibrate at its natural frequency

Solution: \( x = A_{init} \sin(\omega_0 t), \quad \omega_0 = \sqrt{k/m}. \)
Single degree of freedom system

**Free damped vibration:** \( m\ddot{x} + c\dot{x} + kx = 0 \)

**Solution:** depends on the damping ratio \( \xi = \frac{1}{2} \frac{c}{\sqrt{km}} \)

- \( \xi = 0 \)
- \( \xi \ll 1 \)
- \( \xi < 1 \)
- \( \xi = 1 \)
- \( \xi > 1 \)
- \( \xi \gg 1 \)

**UNDERDAMPED SYSTEMS**

Ex: subsea production piping, \( \xi \approx [0.02, 0.1] \)

**OVERDAMPED SYSTEMS**

Ex: door closing mechanism
Single degree of freedom system

**Forced damped vibration:**

\[ m\ddot{x} + c\dot{x} + kx = F e^{i\omega t} \]

Steady-state solution:

\[ x = \frac{1}{k} \left( \frac{1}{1 - (\omega/\omega_0)^2 + 2i\xi(\omega/\omega_0)} \right) F e^{i\omega t} \]

- The displacement \( x \) is proportional to the excitation force, with the proportionality factor being \( H(\omega) = \left[ \frac{1}{1 - (\omega/\omega_0)^2 + 2i\xi(\omega/\omega_0)} \right] \)
- Complex frequency response

The solution can be re-arranged as

\[ x = \frac{1}{\sqrt{(1-(\omega/\omega_0)^2)^2 + (2\xi(\omega/\omega_0))^2}} \left[ \frac{F e^{i(\omega t - \theta)}}{k} \right] \]

where \( |H(\omega)| = \frac{1}{\sqrt{(1-(\omega/\omega_0)^2)^2 + (2\xi(\omega/\omega_0))^2}} \) is the gain (amplification factor)

and \( \theta = \tan^{-1}\left( \frac{2\xi(\omega/\omega_0)}{1-(\omega/\omega_0)^2} \right) \) is the phase difference between the displacement and the excitation force.
Single degree of freedom system

Forced damped vibration amplitude response characteristics

Peak response occurs at frequency \( \omega = \omega_0 \sqrt{1 - 2\xi^2} \)

Peak value of the gain is \( |H(\omega)| = \frac{1}{2\xi \sqrt{1 - \xi^2}} \)
Resonance

Amplification of the system response occurs when the excitation frequency coincides with one of the natural frequencies.

For systems with low damping, resonance can lead to structural damage.

Military helicopter ground resonance test
How to limit mechanical response to vibrations?

First assessment

Increase stiffness $k$ (lower amplitude response over all frequencies)
- Ex: add supports to the piping.

Reduce/suppress excitation force $F$
- Ex: strakes for VIV.

Avoid resonance

Structural eigenfrequencies must be outside $\pm 20\%$ of excitation frequencies
Achieved by tuning $m$, $k$ or (if possible) the excitation force frequency $\omega$
- Ex: increase diameter at the mouth of a closed branch.

If resonance cannot be avoided

Increase damping $c$ to lower amplitude response
- Ex: damper solution applied to the structure.
Case example: subsea production jumper

**Function:** carry production fluid (oil and gas) from the production tree

**Requirements:**
- Flexibility
- Length
- Durability.

**Sources of vibrations:**
- Vortex-Induced Vibrations from the external sea current
- Flow-Induced Vibrations from the internal flow of oil and gas.
Vortex Induced Vibrations (VIV) mechanism

System: cylindrical body (the pipe) with perpendicular current

Vortices are formed downstream of the pipe cross-section with a frequency \( F_{fluid} = \frac{S_r \cdot U_0}{D} \) where \( D \) is the pipe outer diameter and \( S_r \) is the Strouhal number. For a circular cross-section, \( S_r \approx 0.2 \)

Assumption: 2D system, no damping. \( F_{mech} = \frac{1}{2\pi} \sqrt{\frac{k}{m + m_a}} \), where \( m_a \) is the added mass due to the water around the body.

For a circular cylinder, \( m_a = \) displaced mass of water

\[
= \frac{\pi \rho_{water} D^2 L}{4}
\]
Application (exercise 1)

The jumper is modeled as a 40 m long straight pipe fixed at both ends and filled with oil. The seawater current is perpendicular to the jumper and has a velocity of 30 cm/s.

In order to avoid any risk of VIV, the vortex shedding frequency must be outside $\pm 20\%$ of any structure natural frequency.

Is there a risk of VIV for the jumper?

Input data:

- Pipe material: structural steel (A36 grade)
- Pipe outer diameter: 168.3 mm
- Pipe inner diameter: 124.4 mm
- Oil density: 800 kg/m$^3$
- Water density: 1014 kg/m$^3$
Determination of jumper natural frequency

First mode natural frequency is given by $f_1 = \frac{\mu_1}{2\pi L^2} \sqrt{\frac{EI}{w}}$

- $\mu_1 =$ frequency factor for mode 1
- $L =$ pipe length between the supports [m]
- $w =$ jumper mass (including oil and added mass of seawater) per unit length [kg/m]
- $E =$ pipe material modulus of elasticity [N/m²]
- $I =$ moment of inertia [m⁴].

$$I = \frac{\pi}{64} (D_{\text{outer}}^4 - D_{\text{inner}}^4)$$
Flow Induced Vibration (FIV) mechanism

Flow reaction force on a pipe bend for steady-state, single phase flow

Conservation of momentum applied on a control volume inside the bend

\[ \sum \vec{F} = \frac{d}{dt} (m\vec{v}) = \frac{\partial}{\partial t} \iiint_V \rho \vec{v} dV + \iint_S \rho \vec{v} (\vec{v} \cdot \vec{n}) dS \]

0 for steady-state flow

Conventions:
- Normal vector \( \vec{n} \) is defined as positive outwards
- Pressure forces are defined as positive inward

Friction and gravity forces neglected
- Only forces considered are the reaction force from the bend element and the pressure force
- \( \sum \vec{F} = \vec{F}_P + \vec{F}_{b \rightarrow f}, F_{b \rightarrow f} = \text{Force applied by the bend element on the control volume} \)
- Pressure force: \( \vec{F}_P = \iint_S -P\vec{n} dS \)
Flow Induced Vibration (FIV) mechanism

Characteristics of multiphase flow

- Transient distribution of the phases
- Superficial phase velocities: $U_{i,j}^s = \frac{Q_{i,j}}{A}$ where $Q_i$ is the volumetric flow rate of phase $i$
- Mixture velocity: $U_m = U_i^s + U_j^s$

Flow regimes for horizontal flow

A: pipe cross-sectional area

Flow

- Liquid
- Vapor
- Bubbly flow
- Stratified flow
- Stratified-wavy flow
- Plug flow
- Slug flow
- Intermittent flow
- Annular flow
- Mist flow

Dispersed-bubble or bubble flow

Intermittent: Elongated bubble, slug, and churn flow

Stratified-smooth flow

Stratified-wavy flow

Annular flow

$U_i^s$

$U_g^s$
Flow Induced Vibration (FIV) mechanism

Simplified slug flow force calculation

Assumptions

• No gas entrainment in the liquid slug
• $\rho_l \gg \rho_g$
• Steep liquid front and tail
• No liquid film in the gas bubble
• $U_{slug} \sim 1.2U_m$

Example for a 90° bend:

\[
\vec{F} = \frac{\partial}{\partial t} (m \vec{v})
\]

\[
Fx = \rho_l A L_{slug} \frac{U_{slug} - 0}{\Delta t_{passage}}
\]

\[
Fy = \rho_l A L_{slug} \frac{0 - (-U_{slug})}{\Delta t_{passage}}
\]

$\Delta t_{passage} \sim \frac{L_{slug}}{U_{slug}}$ (valid if $L_{slug} >>$ length of the bend)

\[
F = \sqrt{F_x^2 + F_y^2} = \sqrt{2} \rho_l A U_{slug}^2
\]
Application (exercise 2)

Calculate the force due to the flow of water in a 45° bend. Friction and hydrostatic forces in the bend are neglected. Operational pressure is expressed as relative to the external pressure.

Input data

- Operational pressure: 150 bar (relative to external pressure)
- Water density: 1010 kg/m³
- Volume flow rate: 50 000 barrels/day
- Pipe inner diameter: 139.8 mm.

Calculate the force due to an oil slug in a U-bend

Input data

- Gas superficial velocity: 5 m/s
- Oil superficial velocity: 2.8 m/s
- Oil density: 650 kg/m³
- Pipe inner diameter: 10 inches.