Experimental Investigation of the Sensitivity to In-line Motions and Magnus-like Lift Production on the Vortex-Induced Vibrations

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ABSTRACT

Experiments were carried out with a transversely oscillating cylinder in steady current, to investigate the sensitivity to small cylinder pitch rotations and in-line motions on the vortex-induced lift forces. The in-line motions of a spring blade mounted cylinder are convex when towing forward (pushing) and concave when towing backwards (pulling). At the same time the cylinder performs a small pitch rotation (about its central axis), which could induce Magnus-like lift forces. An unexpectedly large sensitivity was found to the in-line motions in certain tests with oscillation amplitudes of over half a diameter. The observed sensitivity to the cylinder pitch rotations was rather small. The tests were carried out in the lock-in regime for VIV and at sub-critical Reynolds numbers, which are close to the critical Reynolds regime. The results presented in this paper will initiate further investigation and will potentially have practical implications as well.

KEY WORDS: Vortex-induced vibrations (VIV); risers; experimental investigation; Reynolds scale effects; Magnus lift forces; in-line motions; forced oscillation

INTRODUCTION

One of the great challenges in the offshore industry is still the assessment of the motions of a circular cylinder in waves and current for application to risers or riser bundles in up to 3,000 m (approximately 10,000 ft) water depth. Here, the fatigue life of riser systems is often dominated by Vortex-Induced Vibrations (VIV). Also, the main concern of riser interference can be largely attributed to VIV effects. Nowadays VIV prediction tools are often based on empirical data from experiments with relatively small riser sections. Such tools allow for three-dimensional analysis of rigid drilling risers, free standing risers, steel catenary risers and flexible risers. User defined current profiles in two directions can be specified.

Large scale VIV tests were carried out at MARIN for filling the VIV database at prototype Reynolds number up to 0.7 million. A flexible mounted model riser of 3.84 m length and 206 mm diameter was towed at high speed through the towing tank. The overhead carriage allowed for the tests with large drag and oscillating lift forces and an electric oscillator was deployed for forced oscillation tests. Earlier experiments in the critical Reynolds regime (drag bucket), have revealed some very important scale effects for the smooth cylinder, similar to what has been reported by Allen and Henning (OTC, 2001).

The spring blade mounted cylinder does not perform a true transverse oscillation but an oscillation that actually lies on an arc of a circle, as shown in Figure 4. The direction of in-line motions change with the towing direction of the carriage. The circular motions are convex with respect to the flow when the carriage moves forward (pushing) and concave when the carriage moves backwards (pulling). At the same time the cylinder performs a small pitch rotation “θ” about its central axis, when oscillating (see Figure 4).

An unexpectedly large sensitivity to the in-line motions was found in certain tests with oscillation amplitudes of over half a diameter. Considerable discrepancies were observed when comparing the measured hydrodynamic reaction loads of the forward and backward towing tests. To investigate the physical background of these discrepancies a number of possible influences have been distinguished, such as: effects of turbulence, damping in the spring system, Mathieu instability, Magnus-like lift effects and in-line motion variations.

The results presented in this paper will initiate further investigation and will potentially have practical implications as well. Flexible risers subjected to VIV do oscillate in a pure cross flow direction, but often describe “figure of eight” type of orbits in two degrees of freedom. Catenary risers in current may have natural modes that consist of combined in-line and cross flow components.

EXPERIMENTS

The experiments were carried out with the new High Reynolds VIV tests apparatus in MARIN’s 210 x 4.0 x 4.0 m (LxBxH) towing tank. The apparatus is schematically depicted in Figure 1.
The test cylinder (No. 3) is suspended from the carriage at approximately 1.7 m submergence on two streamlined vertical struts (No. 1) and two 2.48 m long horizontal spring blades (No. 2). The test cylinder was 206 mm in diameter and 3.84 m in length. The large end plates (No. 4) ensure two dimensionality of the flow. The cylinder was instrumented with a two-component accelerometer and a two-component strain gauge on both ends.

In the present round of experiments a smooth cylinder was tested in forced oscillation, using a 30 kWatt electric oscillator as shown in Figure 2. The adjustable crank wheel (No. 6), the 2.5 m long vertical drive shafts (No. 5) and the adjustable speed of the electro motor (No. 7) provided an accurate control over the cylinder motions.

More details of the High Reynolds VIV test set-up can be found in the ISOPE 2001-JSC-285 paper (de Wilde and Huijsmans, 2001).

**DATA ANALYSIS**

The spring blade mounted cylinder does not perform a true transverse oscillation but an oscillation that lies on an arc of a circle. The motions are convex with respect to the flow when towing forward (pushing) and concave when towing backwards (pulling).

The equation of motion can be written in plane polar co-ordinates ($\theta$, $r$).

$x = r \cos \theta$

$z = r \sin \theta$

The spring blade deflections follow from beam theory, resulting in the following pitch angle and radius:

$\theta = \frac{3z}{2L}$

$r = \frac{z}{\sin \theta} \approx \frac{2}{3}L$

The generalised equations of motions become:

$m\ddot{r} - m\dot{r}\dot{\theta}^2 = F_t$

$m\dot{r}^2\dot{\theta} + 2mr\ddot{\theta} = rF_0$
By subtracting the cylinder inertia and centrifugal loads from the measured loads, the effective hydrodynamic loads can be derived. The cylinder-fixed loads can be transformed to Cartesian loads, with normal components in-line with the flow (drag) and lateral to the flow (lift).

The oscillating lift force may have a phase difference with the cylinder motion, which can be represented as follows:

\[ z(t) = A \sin(\omega t) \]
\[ F_L(t) = F_{L0} \sin(\omega t + \phi_0) \text{ or } F_{L0} \sin \phi_0 \cos(\omega t) + F_L \cos \phi_0 \sin(\omega t) \]

The in-phase and the out-of-phase lift forces relate to the added mass and the power transfer from the fluid to the cylinder respectively. The power transfer can be either positive (exciting) or negative (damping). The lift coefficient in-phase with the velocity and the lift coefficient in-phase with the acceleration can be defined as follows:

\[ C_{LV} = \frac{F_{L0} \sin \phi_0}{DL \frac{1}{2} \rho U^2} \quad \text{and} \quad C_{LA} = \frac{-F_{L0} \cos \phi_0}{DL \frac{1}{2} \rho U^2} \]

The added mass coefficient can be calculated from the in-phase lift forces:

\[ C_M = \frac{-F_{L0} \cos \phi_0}{\rho \frac{\pi D}{4} L (2\pi f)^3} \]

Using the same sign conventions as Gopalkrishnan (1993), a positive \( C_{LV} \) coefficient denotes an increasing effect, or power transfer from the fluid to the cylinder oscillation and a positive \( C_{LA} \) coefficient denotes a negative added mass.

**IN-LINE MOTIONS AND MAGNUS FLOW**

An unexpectedly large difference was found in our experiments, when towing forward and backwards. Initially, it was believed that the turbulence from the vertical struts could explain the discrepancy. However, repeat tests with similar “dummy” struts upstream did not show such sensitivity. Moreover, by comparing existing data, it turned out that the forward towing tests without turbulence were behaving unexpectedly anyway.

In case of free vibrating VIV tests, a large number of possible explanations can be given for the observed differences (see Table 1). However, our forced oscillation VIV tests provide a much better defined situation, in which at least the motions are completely under control. The physics decrease basically from a fully coupled hydro-structural problem to a normal hydrodynamic problem. The important damping parameter is eliminated from the experiments as well. In fact only effects “E” and “F” (see Table 1) remain as possible candidates in the forced oscillation tests.

**Table 1 – Explanations for forward-backwards discrepancy**

<table>
<thead>
<tr>
<th>Effect</th>
<th>Comments</th>
</tr>
</thead>
</table>
| A: Turbulence | – Seems unlikely for streamlined struts
| | – Tests with upstream flow disturbances showed insufficient sensitivity
| | – Also the forward towing tests without turbulence behaved unexpectedly |
| B: Damping of spring blades | – Not applicable to forced oscillation |
| C: Drag loads of pitching cylinder | – The conservative force field cannot explain damping or excitation
| | – The spring blade fixed loads can be transformed to Cartesian co-ordinates with normal in-line and cross flow components |
| D: Mathieu type instability | – Non-linear oscillations, in which there is not a periodic excitation, independent of the motion, but rather the system parameters such as the mass and the spring stiffness are periodic in time (Hagedorn, 1988)
| | – Not applicable to forced oscillation |
| E: Magnus-like lift production | – The pitch motions of approximately 7 degrees are small
| | – A Magnus lift force may occur for the pitching cylinder in current
| | – A high sensitivity has not been reported before |
| F: In-line motions | – The in-line motions of approximately 3% are very small
| | – It is known that in-line motions can affect the transverse motions, but a high sensitivity has not been reported before |

A modified apparatus was used to isolate the Magnus effect from the list of possible explanations. Details of this modification will be discussed at the end of this section.

**Magnus Effect**

The flow about a spinning cylinder in current is schematically shown in Figure 6. The rotation alters the pressure distribution and also affects the boundary layer separation. Separation is delayed on the upper surface and occurs earlier on the lower surface. The classical Magnus lift force is upwards for backward spin.

![Magnus flow](image-url)
In non-separating potential flow theory, the Magnus lift force can be calculated from the circulation $\Gamma$:

$$F_m = \rho U \Gamma$$

For real flows, the Magnus lift force scales with the dynamic pressure:

$$F_m = C_m \frac{1}{2} \rho U^2$$

with:

$$C_m = f \left( \frac{\Omega D}{2U} \right)$$

$\Omega$ is the rotation velocity of the cylinder.

For the oscillating spring blade mounted cylinder it can be expected that a Magnus-like lift force will be in-phase with the cylinder velocity. Thus being a non-conservative force that could either damp or excite the motions. In our experiments with 2.48 m long spring blades and an oscillation amplitude of approximately one diameter, the amplitude of the speed ratio $v/U$ is approximately 7%. The Magnus lift coefficient for a steady rotation would then be in the order of $C_m = 0.03$, which is significant (Hoerner, 1985). Since the Magnus force for small speed ratios can be either positive or negative, either exciting or damping effects can be expected.

**In-Line Motions**

The circular motions of the spring blade mounted cylinder can be expressed in polar co-ordinates as discussed in the previous section. For small amplitudes the in-line motions can be approximated as follows:

$$x = -\frac{2^2}{2R} \sin^2(\omega t) = -\frac{2^2}{4R} + \frac{2^2}{4R} \cos(2\omega t)$$

The in-line motions have a double frequency and change sign with the direction of towing. For our 2.48 m long spring blades the in-line motions are approximately 3% of the cross flow motions. The pitch rotation amplitude is approximately 7 degrees.

**Modified Apparatus for Avoiding Magnus Effect**

The test apparatus was modified for testing with and without pitch rotation, as shown in Figure 7. The cylinder was then attached to the spring blades with rotational bearings on both ends. A stiff vertical shaft with a gliding and rotating bearing on the top-end was used to prevent the cylinder rotation. With the 2.8 m long shaft, the pitch rotations could be reduced by a factor 28 from approximately 7.0 to 0.25 degrees. Tests with and without the anti-Magnus shafts were carried out. The name was assigned, because the suspected physical phenomenon was that of Magnus-like lift production.

### RESULTS AND DISCUSSION

A series of experiments was carried out to investigate the sensitivity to the small in-line and rotational motions of our spring blade mounted cylinder. The forward and backward towing tests provided two distinct situations of in-line motions (respectively convex and concave). The sensitivity to the pitch rotation was investigated by either rotating the cylinder with the spring blade deflection (without anti-Magnus shafts) or with virtually no cylinder rotations (with the anti-Magnus shafts). The test conditions and the test matrix are given in Tables 2 and 3. A sub-critical Reynolds number was chosen for better comparison with existing data. The tow speed and oscillation frequency was chosen in the middle of the lock-in regime for VIV ($U_r = 6.9$).

In some of our further investigations of the forward-backward sensitivity, the spring blades were replaced by hinged beams of twice the original length. This modification reduces the in-line motions by more than a factor 3. These forward and backwards towing tests are respectively designated as “pos long” and “neg long”.

**Table 2 - Test conditions**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reynolds number</td>
<td>$Re$</td>
<td>$7.85 \times 10^4$</td>
</tr>
<tr>
<td>Reduced velocity</td>
<td>$U_r = \frac{UT}{D}$</td>
<td>$6.9$</td>
</tr>
<tr>
<td>Amplitude ratio</td>
<td>$A/D$</td>
<td>$0.3 – 1.4$</td>
</tr>
</tbody>
</table>

**Table 3 - Test matrix**

<table>
<thead>
<tr>
<th>Designation</th>
<th>With Magnus effect (cylinder rotates)</th>
<th>Without Magnus effect (cylinder does not rotate)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Tests with original spring blades</td>
<td>Tests with enlarged spring blades</td>
</tr>
<tr>
<td>Forward</td>
<td>pos w M</td>
<td>pos long</td>
</tr>
<tr>
<td>Backwards</td>
<td>neg w M</td>
<td>neg long</td>
</tr>
<tr>
<td></td>
<td></td>
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<tr>
<td>Backwards</td>
<td>neg w M</td>
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<td></td>
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</tr>
<tr>
<td>Backwards</td>
<td>neg long</td>
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</tbody>
</table>
Fig. 8 – Time traces of lift loads
Examples of the time traces of the measured lift loads are presented in Figure 8. A large difference between the forward and backward lift loads can be observed. The forward loads are much larger in amplitude and the phase angle is different as well. The forward loads in the example seem more stable than in the backward ones. However, significant beating-like effects were observed in certain forward towing tests as well (e.g. at other reduced velocity or amplitude ratio). The tests with the longer spring blades show clear beating-like effects both for forward and for backward towing.

The lift coefficients in-phase with the velocity and the added mass coefficients are presented in Figure 9. Gopalkrishnan (1993) data at the same reduced velocity and at a Reynolds number of approximately 10^5 are presented for comparison.

A large sensitivity to Magnus-like lift force has not been observed in our experiments. Some tests show some sensitivity, but on a much smaller scale than the forward-backward sensitivity.

In some of our further investigations of the forward-backward sensitivity, the spring blades were extended considerably, reducing the in-line motions by more than a factor 3. The repeated tests showed a significant reduction in the discrepancies with existing data as well as in the forward - backward discrepancies. The time traces in Figure 8 show a switching behaviour (beating) between two distinct “modes of oscillation” of which one is close to the mode observed when towing backwards. Finally, it should be noted that some exploratory tests with a rough cylinder showed a decrease of the discrepancies as well.

CONCLUSIONS AND RECOMMENDATIONS

Based on the present results of experiments with the smooth cylinder on the 2.48 m long spring blades, the following conclusions and recommendations seem justified:

1. An unexpectedly large sensitivity was found to the in-line motions in our VIV tests with a transversely oscillating cylinder in flow. At oscillation amplitudes of more than half a diameter, a large and consistent difference in the hydrodynamic lift forces was found when comparing the forward and backwards towed tests. The oscillations of the spring blade mounted cylinder are not purely transverse, but actually lie on an arc. The motions are convex when towing forward (pushing) and concave when towing backward (pulling). The experiments were carried out at sub critical Reynolds numbers in the lock-in regime for VIV.

2. Although it is known that the in-line motions can affect the lateral oscillations, such a large sensitivity has not been reported before. The measured lift loads in our experiments deviated in some cases very significantly from existing data with true lateral oscillations. This large sensitivity is not properly understood at present.

3. The observed sensitivity may have significant practical implications as well. Further tests at other Reynolds numbers and reduced velocities are required to further explore the parameter range of the sensitivity. The effect may for instance only be applicable to very smooth cylinders in the critical or close to critical Reynolds regime.

4. The present results of experiments do not show a large sensitivity to the cylinder pitch rotations and associated Magnus-like lift forces. Further investigation into this effect with higher Reynolds numbers for instance, seems still worthwhile.

5. The nature of the in-line motions in the experiments was prefixed, due to the set-up of the experiments. Independent in-line motion variation experiments (on top of vertical motions) will have to be performed to verify the impact on real life situations. Tests with in-line springs (e.g. 2 degrees of freedom) are recommended as well.

REFERENCES


