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Pierre Moussou, Vincent Fichet, Luc Pastur, Constance Duhamel, Yannick Tampango. Experimental Investigation of the Flow-Induced Vibrations of a Rod Cluster Control Assembly Inside Guides With Enlarged Gaps. ASME 2019 Pressure Vessels & Piping Conference, American Society of Mechanical Engineers, Jul 2019, San Antonio, TX, United States. 10.1115/PVP2019-93143 . hal-04492460

HAL Id: hal-04492460

<https://hal.science/hal-04492460>

Submitted on 11 Mar 2024

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PREPRINT PVP2019-93143

EXPERIMENTAL INVESTIGATION OF THE FLOW-INDUCED VIBRATIONS OF A ROD CLUSTER CONTROL ASSEMBLY INSIDE GUIDES WITH ENLARGED GAPS

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ABSTRACT

In order to better understand the mechanisms of fretting wear damage of guide cards in some Pressurized Water Reactor (PWR) Nuclear Power Plant (NPP), an experimental investigation is undertaken at the Magaly facility in Le Creusot. The test rig consists of a complete Rod Cluster with eleven Guide Cards, submitted to axial flow inside a water tunnel. In order to mimic the effect of fretting wear, the four lower guide cards have enlarged gaps, so that the Control Rods are free to oscillate. The test rig is operated at ambient temperature and pressure, and Plexiglas walls can be arranged along its upper part, and a series of camera records the vibrations of the control rods above and below the guide cards. The vertical flow velocity is in the range of a few m/s. Beam-like pinned-pinned modes at about 5 Hz are observed, and oscillations of several mm of the central rods are measured, which come along with impacts at the higher flow velocities. A simple non-linear calculation reveals that the main effect of the impacts between Control Rods and Guide Cards is an increase of the natural frequency of the rods by about 10%. Furthermore, as the vibration spectra collapse remarkably well with the flow velocity, the experiments prove that turbulent forcing is responsible for the large oscillations of the control rods, no other mechanism being involved.

k_{equ} bending stiffness term of a control rod (in $N.m^{-1}$)
 S_{m1m1} PSD of the modal displacement (m^2/Hz)
 U fluid velocity in the guide card (in m/s)
 u dimensionless displacement
 ρ fluid density (kg/m^3)

INTRODUCTION

Despite many years of monitoring in various nuclear power plants [1, 2, 3], the vibrations of Control Rod Assemblies (CRA) do not seem to have drawn a large attention but for a few papers [4, 5, 6]. More efforts have been devoted to the study of wear of Control Rods and Guide Cards [7, 8], but less to their root cause. From a flow-induced vibration perspective, these devices do not differ from other flexible structures submitted to turbulent forcing, but for one point: they do not fit simply into one of the traditional categories of phenomena that textbooks promote [9, 10, 11]. As formulated by Païdoussis at the Karlsruhe conference [12], when referring to a guide tube vibration issue: ‘There is insufficient information on the geometry of the control-rod/guide tube assembly and the hydrodynamics involved. It is clearly a case of flow-induced vibration of the control rods against the guide tube.’

In Pressurized Water Reactors (PWR), control rods are slender cylinders several meters long, attached at their upper end to a mechanical mobile part called a ‘spider’, as shown in Fig. 1. Their role consists in quenching the nuclear reaction by introducing a neutron moderator material into the nuclear core. To fulfill that purpose, they are dropped from about 4m into fuel assemblies, and the fall must be perfectly centered into the guide tubes of the fuel assemblies to achieve a safe control of the nuclear reaction. In the upper part of the control rod housing, the centering of the rods before entering into the fuel assemblies is achieved by a series of plates called ‘guide cards’ with holes drilled at the reference position of each control rod (see Figures 1 and 2).

NOMENCLATURE

CRA Control Rod Assemblies, sometimes referred to as Rod Cluster Control Assemblies (RCCA)
CRC Control Rod Cluster
NPP Nuclear Power Plant
PWR Pressurized Water Reactor
PSD Power Spectrum Density (in physical units² / Hz)
 D Rod diameter, equal to 9.68 mm
 f average frequency (in Hz)
 I inertia of a control rod (in m^4)
 k wavenumber of a sine-like mode (in m^{-1})

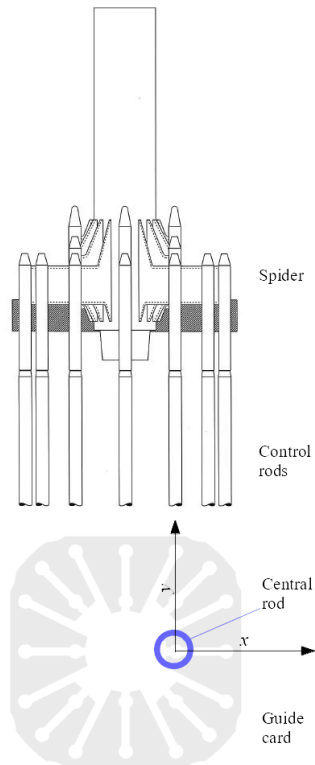


Fig. 1: Spider and upper part of the control rods (upper drawing) and a guide card seen from above (lower drawing)

When the control rods are raised up in nominal operating condition, water flows from bottom to top, and flow-induced vibrations are generated by the axial flow in the housing, and also by the partial cross-flow and the turbulence generated by the guide cards which behave like orifice plates. The vertical velocity is estimated to be of the order of one m/s, a relatively small value compared to usual flow velocities in water pipes and vessels, yet high enough to generate flow-induced vibrations because, by design, the control rods have only one support at their upper end. In these conditions, they bend easily and a small unsteady pressure is enough to trigger oscillations up to several tens of diameters in amplitude. For that reason, wear issues have been observed with control rods and guide cards during the lifetime of PWR nuclear power plants in the past decades.

The present study aims at determining the vibration pattern of a control rod in a mock-up in a comprehensive way, and more specifically to determine the mode shapes involved, to investigate the effects of spurious impacts of a rod against the guide and to determine whether turbulent forcing only is involved, or if other mechanisms like leakage-flow vibrations play a role. Wear issues range outside the scope of the current study.

EXPERIMENTAL APPROACH

The Magaly test rig is partially shown in Fig. 2. It reproduces the housing and the guides of a control rod at scale 1. A cluster of control rods is arranged inside it, and Plexiglas walls make a direct observation of the vibrations possible. For a better representativeness, a partial fuel assembly is arranged at the bottom. The test rig is submitted to a variable ascending flow rate of the order of $70 \text{ m}^3/\text{h}$, in water at ambient temperature and low pressure. The prediction of vibrations in reactor conditions requires hence an extrapolation of the fluid forces through dimension analysis.

The experiments reported in the present paper rely on enlarged gaps in the five lower guide cards, so that large amplitude vibrations of the rods can be easily observed in the lower housing. These artificial gaps mimic also an important wear of the cards, with the idea of investigating the dynamic behavior of a control rod guided by excessively damaged cards. The current study is focused on the vibrations of a central rod, as shown by its location inside the blue circle in Fig.1, because previous tests have shown that it was prone to large oscillations.

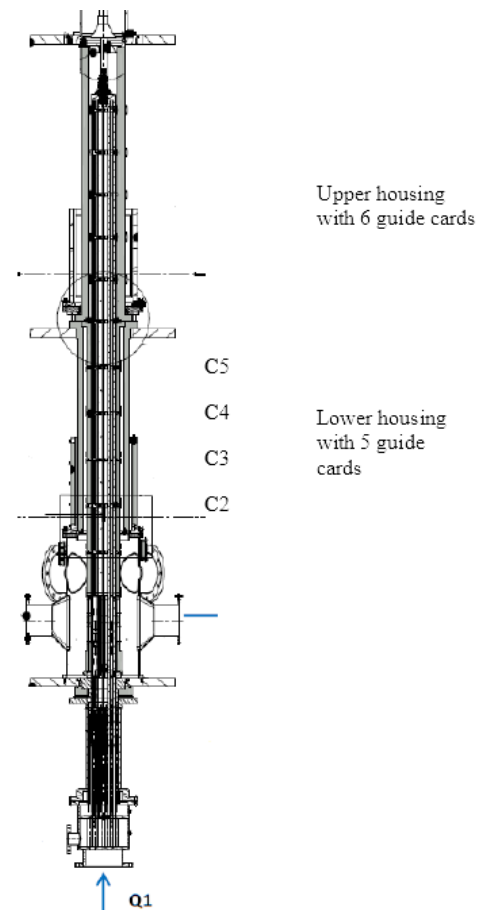


Fig. 2: Partial view of the Magaly test rig, reproducing at scale 1 the housing of an array of control rods hanging above the reactor core

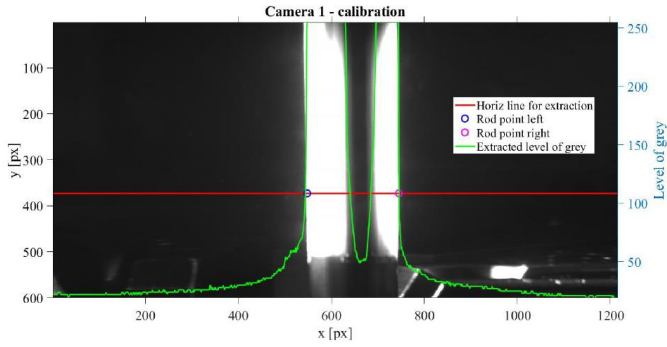


Fig. 3: Image calibration before determining the rod position

Four couples of cameras are mounted in the horizontal plane just above the upper sides of guide cards #2 to #5 (in ascending order), so that the x and y displacements of the control rod are simultaneously recorded at four different elevations. With an adequate side light, the recorded pictures can be processed in real-time to estimate the x and y positions of the rod with a sampling frequency equal to 80 Hz and an uncertainty equal to $\pm 0.01\text{mm}$, i.e. less than 1% of the rod diameter. 4000 points are recorded for a total duration equal to 50 seconds (see illustration in Fig. 3). In the local frame reference, x is the direction of the guide card axis.

Three different test cases denoted C1, C2 and C93 are performed, corresponding to flow rates equal to average flow velocities inside the orifice plates, estimated to 13.2, 14.5 and 18.5 m/s by mass conservation, respectively.

TIME SIGNALS

As a first step, the time signals obtained in the three test cases are reproduced in the first column of Figs. 4-6, by increasing flow velocities. Further details can be found in appendix, where it can be observed that the dominant displacement occurs in the direction x , which is the only one discussed further on. The vibration amplitude increases as expected with the flow velocity, and a baseline becomes visible, especially for the higher guide cards. This baseline is obviously due to impacts occurring in the upper part of the Control Rod Cluster. This effect can be made evident by plotting the extrema distribution (central column in the figures): the maximum distribution has a Gaussian shape in all cases, whereas the minimum distribution departs from it in large proportions, with a large number of minima occurring at -1mm , to be associated with at least one guide card edge. The third column of the figures shows the Power Spectrum Densities of the rod displacements: a dominant peak is present at about 5 Hz, and this frequency increases slightly with the flow velocity. A second peak at twice this frequency is also present at the higher flow velocities, which raises the question of the impacts influence on the PSD.

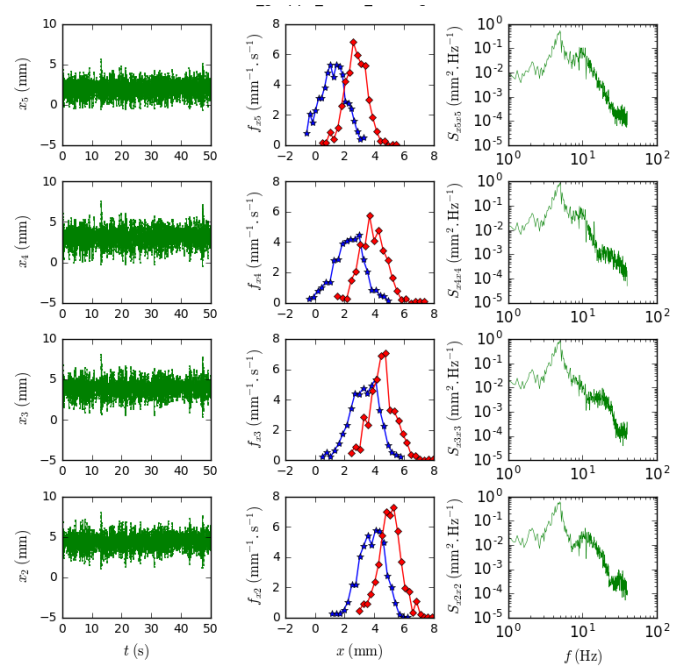


Fig. 4: Time signal, extrema distribution (blue: minima, red: maxima) and Power Spectrum Density of the cylinder displacement from top to bottom, C1 case

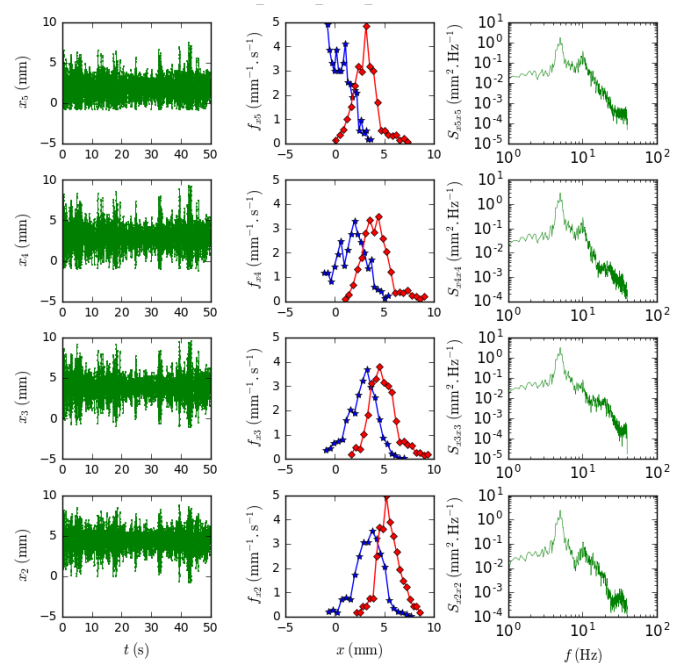


Fig. 5: Time signal, extrema distribution (blue: minima, red: maxima) and Power Spectrum Density of the cylinder displacement from top to bottom, C2 case

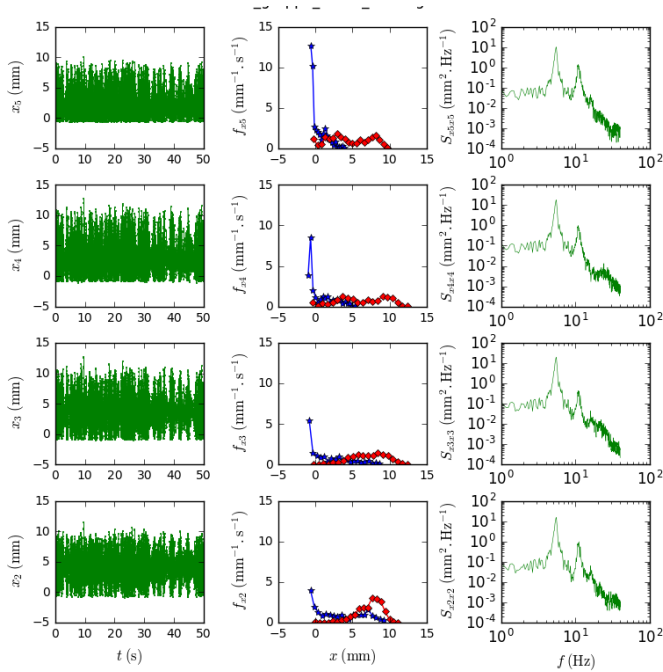


Fig. 6: Time signal, extrema distribution (blue: minima, red: maxima) and Power Spectrum Density of the cylinder displacement from top to bottom, C93 case

INFLUENCE OF THE IMPACTS

Though the impacts are not perfectly periodic, a sine function model can be proposed to understand their effect in terms of PSD. For that purpose, a pure sine signal is synthesized with a dimensionless amplitude equal to unity, and a sampling frequency identical to the tests' one. Impacts are modeled by an ideal rebound without energy loss: when the signal reaches a threshold, the velocity sign changes and its amplitude is conserved, as illustrated in Fig. 7. In other words, if the time signal is a pure sine in the absence of threshold (black curve in the upper graph of Fig. 7), the presence of an obstacle generates an ideal bounce after which the ingoing velocity is turned into its opposite (see for instance the orange curve in the upper graph Fig. 7). Three different values of the threshold are investigated: it can be seen that even a limited truncation of the sine generates a visible second harmonic in the PSD. Furthermore, the velocity sign inversion can be thought of as a timewise truncation of the signal, with the part of the period below the threshold being removed. This representation provides a simple explanation to the frequency evolution with the impacts: it increases because the signal is shortened. This effect is clearly visible in the PSD graph shown in Fig. 7. It can hence be assessed without further proof that the spectral signature evolution of the measurements with the flow velocity can be explained by a truncation mechanism involving a frequency increase and the apparition of harmonics.

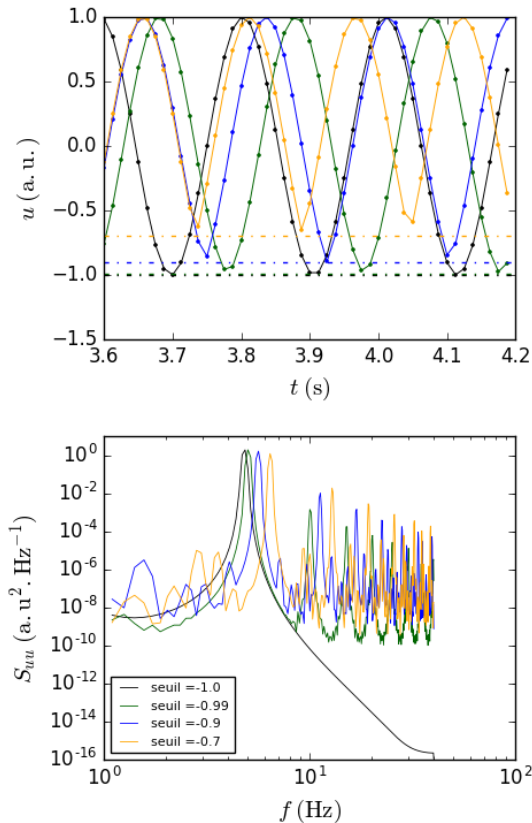


Fig. 7: Illustration of the non-linear effects of an impact upon an ideal sine signal u with an amplitude equal to 1

NATURAL FREQUENCIES IDENTIFICATION

The question which arises now is the number of modes needed for describing the dynamics of the rod. One way to shed light on this matter consists in plotting the transfer function between the four displacements: according to linear dynamics theory [13], all displacements are in phase when the structure oscillates at one of its natural frequencies. A natural mode is then measured when a peak is present and when all measurement points exhibit the same argument.

The application of this criterion is straightforward, and illustrated by the Figs 8-10, where the cross spectra of the four displacement are plotted with the upper displacement being chosen as reference signal. For illustration purposes, four vertical lines are plotted. The cyan one stands for a quasi-steady deformation range, and, though a peak seems to be present in the C1 case, it cannot be considered as a natural mode because it is not present in the other test cases. Without further indication, this small peak is deemed irrelevant for the analysis, and it may be due to an external source. The second vertical line in blue is centered on the dominant peak at 5 Hz: it can be observed that the four displacements are in-phase in the three test cases. The criterion is hence fulfilled and there is a natural mode at 5Hz. The orange vertical line corresponds to the second harmonic of the first mode.

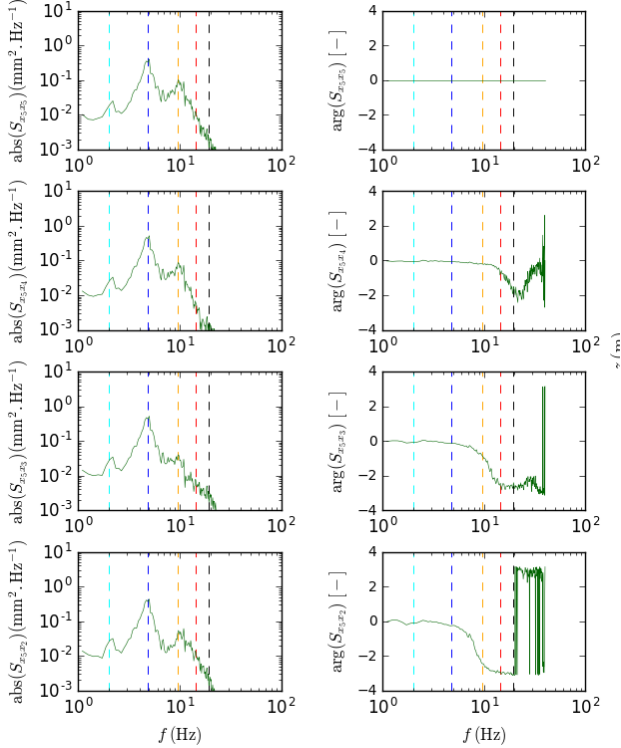


Fig. 8: Cross-spectra of the rod displacements from top to bottom, C1 case

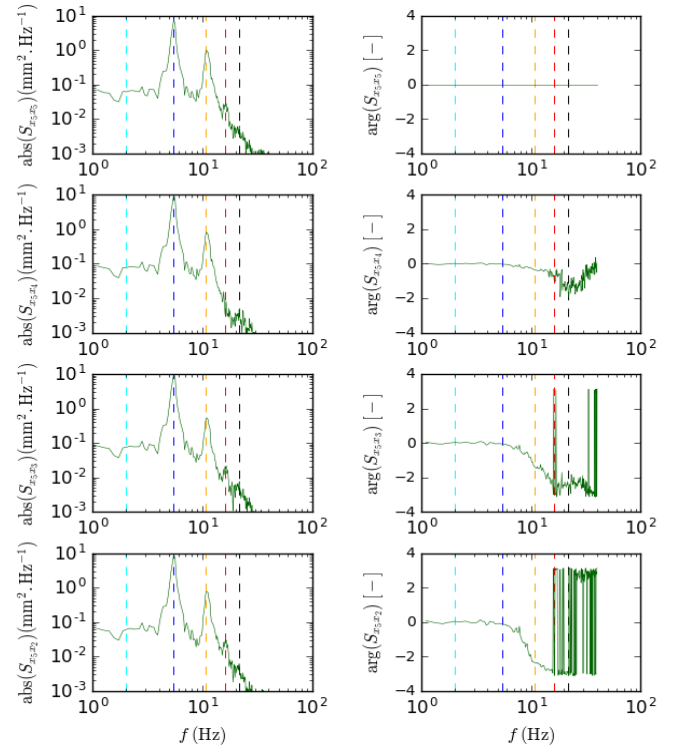


Fig. 10: Cross-spectra of the rod displacements from top to bottom, C93 case

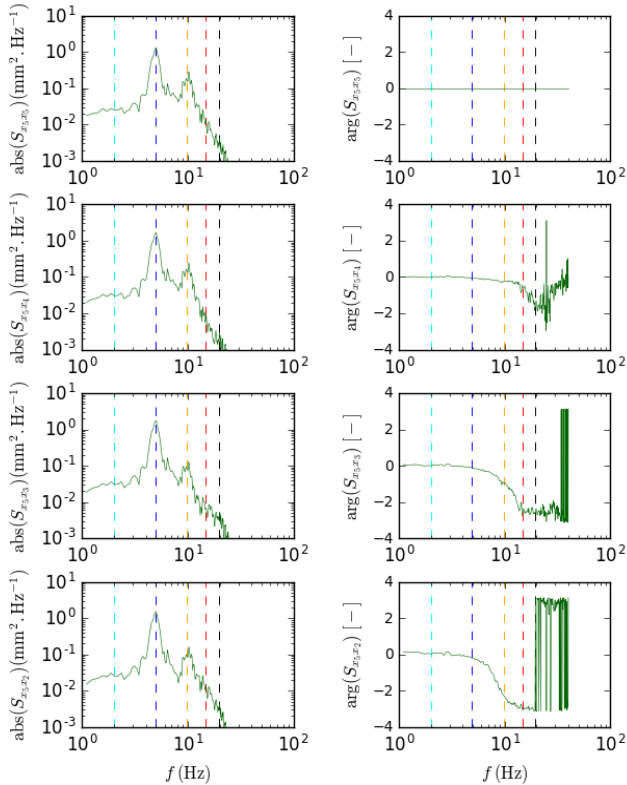


Fig. 9: Cross-spectra of the rod displacements from top to bottom, C2 case

As can be seen, the arguments of the four displacements are not identical nor opposite, which clearly indicates that this second peak is not a natural mode, and, according to the previous section, it is a non-linear consequence of the impacts. A fourth vertical line is plotted at three times the first natural frequency but no harmonic peak is present. Finally, a fifth vertical line is plotted at four times the first natural frequency, and a small peak is observed in the three test cases. This frequency is identified as a the second natural mode of the rod, because the four displacements have identical or opposite phases, and because the natural frequencies of a pinned-pinned beam follow a progression proportional to the square of the mode order [14]: the first natural frequency of the rod being equal to some 5 Hz, the second natural frequency is expected at about 20 Hz, and this is what is actually observed.

EXPANSION OF THE DEFLECTION SHAPE IN NATURAL MODES

The previous section has shown that two natural modes were needed for describing the vibrations of the control rod in the frequency range of the study. The simpler approach to build up an experimental modal shape consists in considering the real parts of the displacement cross-spectra shown in Figs. 8-10 at the frequency peaks, and to plot them in dimensionless form. As can be seen in Fig. 11, the three test cases bring out similar shapes which can reasonably well be fitted by a sine function.

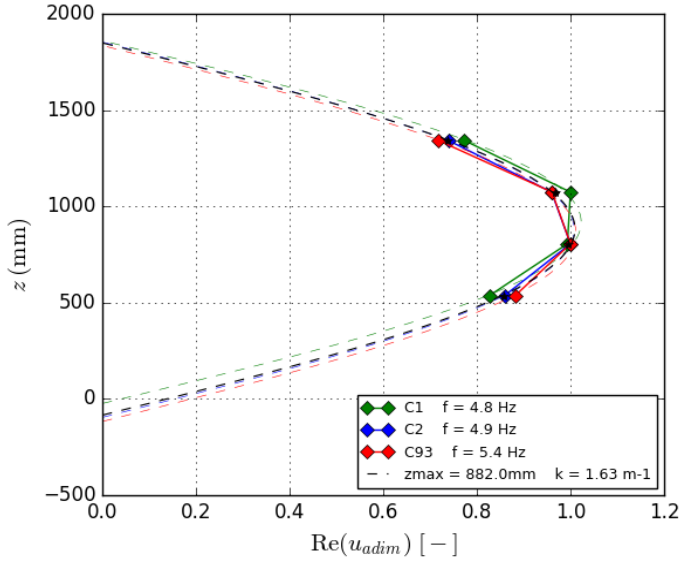


Fig. 11: first mode shape of the rod with a sine-like fit

Furthermore, the same sine fit holds well for the three cases, which proves that the upper and lower guide charts that were not enlarged play the role of pinned conditions, a self-evident outcome that is nonetheless satisfying when performing structural dynamics experiments. Another observation is that the modal deformation is at first order not altered by non-linear effects.

The second natural mode can be theoretically deduced from the boundary conditions, assuming that if the first one is proportional to $\sin kz$, the second one is proportional to $\sin 2kz$. A slightly better fit is obtained by removing a small contribution of the first mode to the measured data, which is of no practical consequence in the framework of the present study. Be it enough here to indicate that the first mode is the major contributor to the vibration pattern of the rod, as can be seen from the respective contributions of the first and second mode shown in Fig. 12, obtained by performing the scalar product of the measurement vector with the theoretical mode shape. A similar conclusion can be drawn by comparing the time signals with their modal projections as shown in Appendix. For all practical purposes, the dominant term in the vibration pattern of the control rod is its first mode.

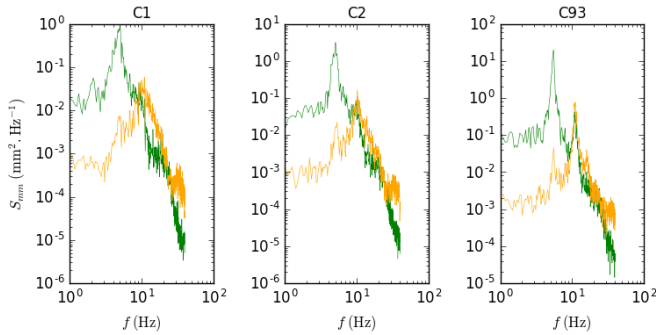


Fig.12: Power Spectrum Density of the first and second modal displacements

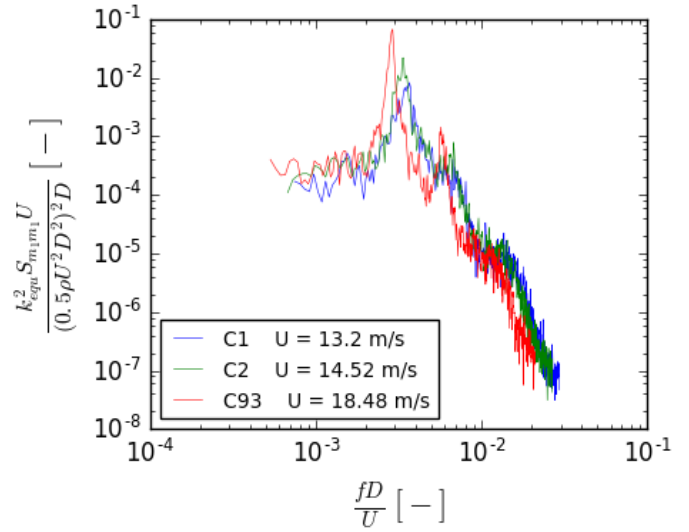


Fig. 13: cylinder vibration in dimensionless form

DIMENSIONLESS VIBRATION SPECTRUM

The final step of the study consists in comparing the vibration modal amplitudes in the three test cases, with an adequate presentation in dimensionless form. The frequency can classically be made dimensionless by introducing a flow velocity, chosen here as the average velocity through a guide card, and a typical length, chosen equal to the rod diameter. The PSD amplitude requires some more efforts to be made dimensionless, the choice was made here to introduce a structural stiffness based on the distance between the guide cards and the flexural inertia I of the tubes. The results of the three test cases are plotted in Fig. 13.

As can be seen, the three curves collapse fairly well, and the average levels and the final slope are consistent from one test to the next. No deviation from the average spectrum is observed. This simple result based on a dimensionless form consistent with turbulent forcing indicates that there is no indication of another type of excitation in the present study. Turbulent forcing can hence be considered as the only source of vibrations for the tested control rods.

CONCLUSION AND PERSPECTIVES

The study reported here has highlighted the physical phenomena involved in flow-induced vibration of a rod control inside guide cards with enlarged gaps, based on tests performed with the Magaly test rig facility. The objective was to determine if turbulent forcing was the root cause of the vibrations.

The measurement pattern is due mainly to the first pinned-pinned mode of the control rod. The investigation of the time signals in three different hydraulic regimes has shown that

impacts between control rods and guide cards appear at high velocities. A truncated sine model has been proposed to understand the effects of these impacts on the dynamic behavior, and it was found that the dominant peak increase with the flow velocity is a normal consequence of the impacts. The collapse of the three different modal spectra in dimensionless form proves that turbulent forcing is the only observable cause of flow-induced vibrations. Other phenomena like vortex shedding or leakage flow instabilities would be dependent on the velocity flow up to the point where the three curves would not collapse.

An important side effect of the analysis is that the impacts do not alter significantly the vibration pattern, so that a linear approach should be sufficient to derive a reference fluid force model. Further work is needed for comparing these experimental results with unsteady CFD predictions combined to a simple mechanical model of the rods.

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APPENDIX: SUCCESSIVE POSITIONS OF THE CONTROL ROD IN THE THREE CASE TESTS

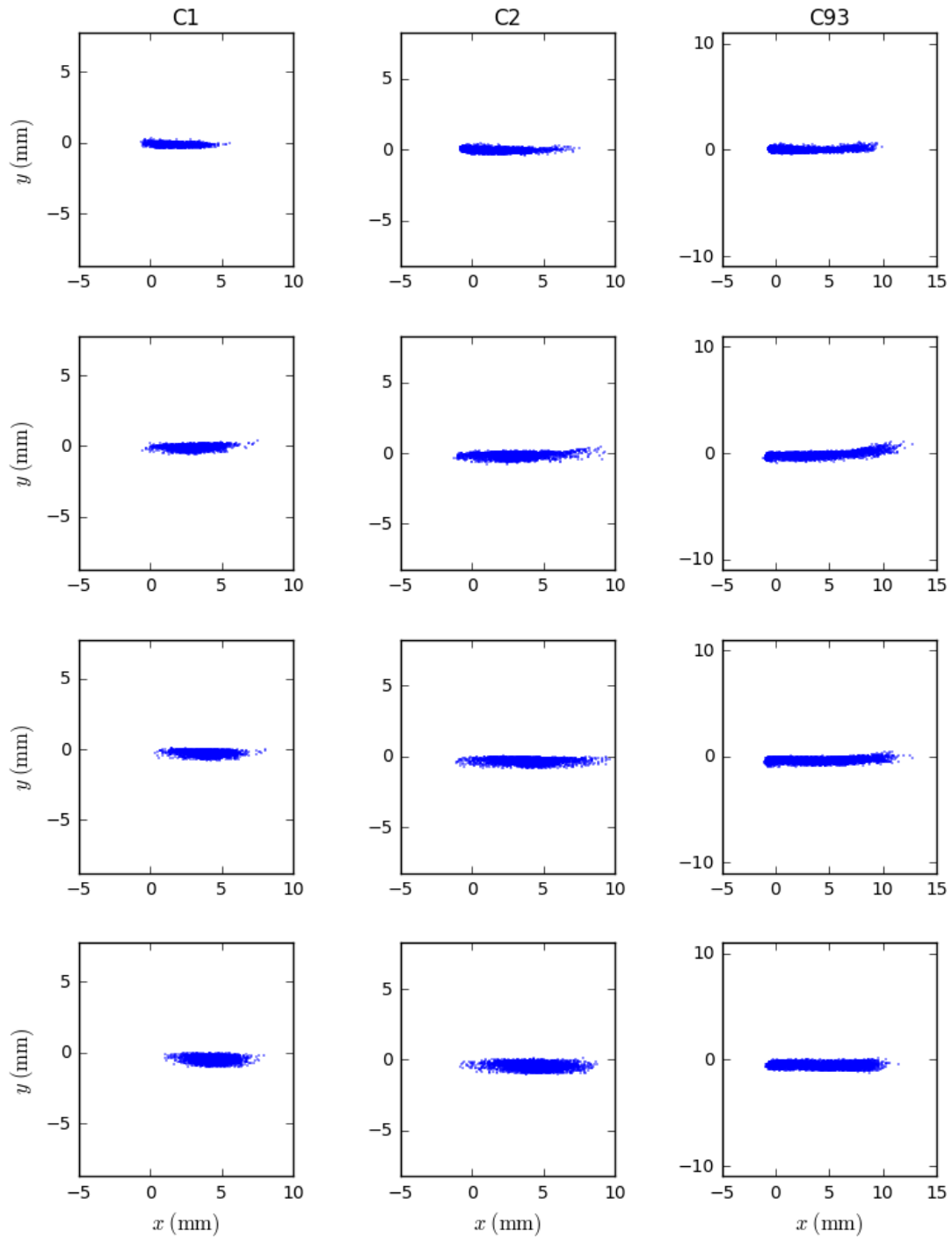


Fig. 14: Successive locations of the cylinder in four horizontal planes

APPENDIX: MODAL CONTRIBUTIONS IN THE THREE CASE TESTS

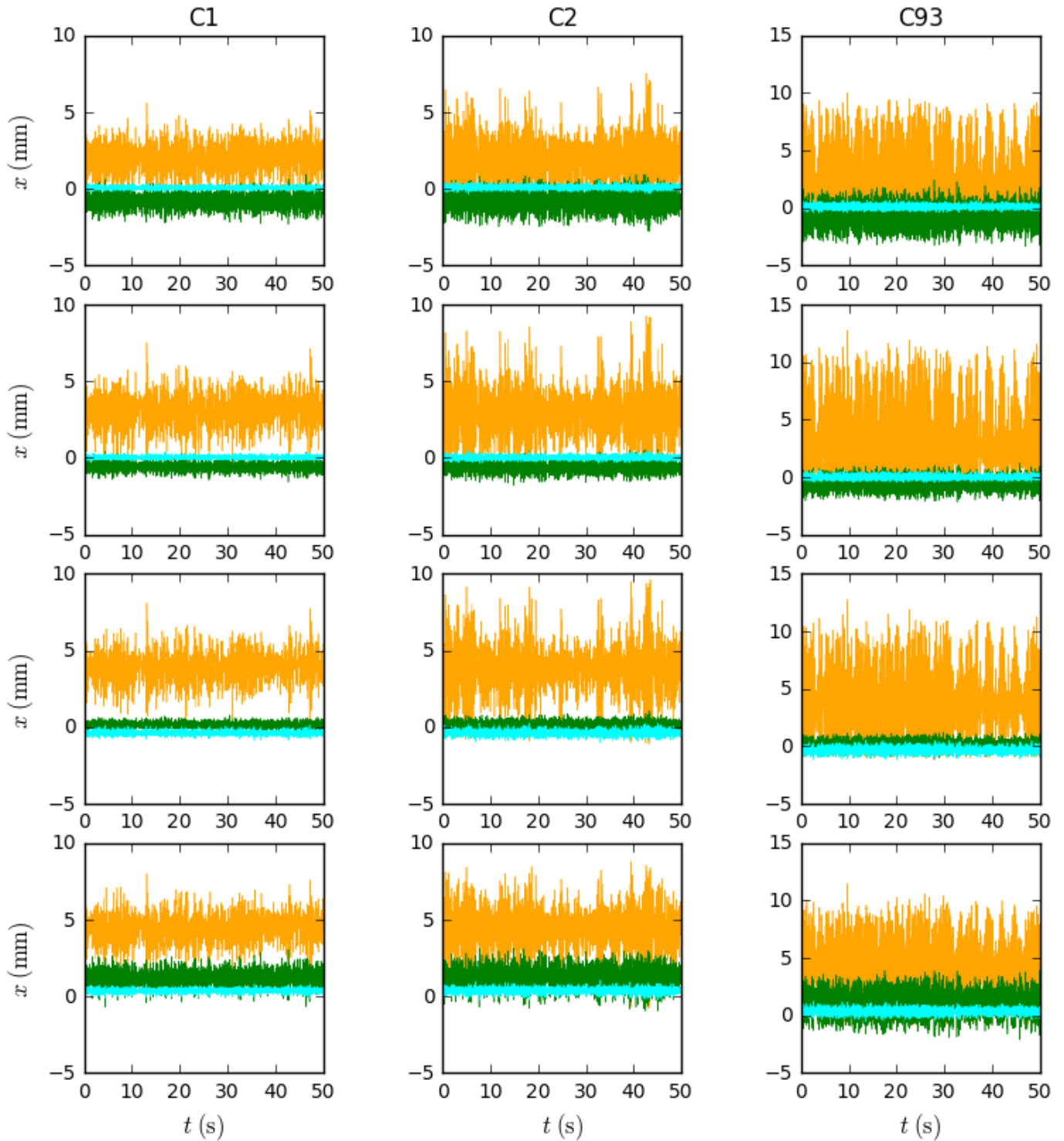


Fig. 15: Modal contributions of the first and second mode: orange = time displacement, green = time displacement minus the first mode contribution, cyan = time displacement minus the first and second mode contribution.