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Addressing Source-Induced Structural Vibrations in an Interventional X-ray System

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Abstract

Interventional radiology is a medical discipline, in which imaging equipment is used to diagnose and treat diseases throughout the human body. Dedicated systems, based on X-ray, computed tomography and ultrasound technology, allow for 2D and 3D imaging of (contrast enhanced) physical structures. Using these images, physicians are able to perform minimally invasive, image-guided therapy. Interventional X-ray systems typically feature an X-ray source and detector mounted to the ends of a C-shaped arm (C-arm). The C-arm is part of a larger structure, called stand. The X-ray source produces distinct disturbance forces at a specific frequency, which act upon the C-arm. This leads to vibrations of the stand, which could potentially result in acoustic noise and image quality degradation. As these effects could be clinically unacceptable, their extent is being investigated.

Modal analyses have been performed to investigate the dynamic behaviour of the stand. This has revealed a number of eigenmodes with eigenfrequencies close to the excitation frequency of the X-ray source. These eigenmodes involve local deflections of the end of the C-arm, and the X-ray source interface. Two methods are explored to decrease the resonant effects corresponding to the excited eigenmodes. The first approach aims to apply Tuned Mass Dampers (TMD) to the C-arm. The second approach proposes local stiffness enhancement of the C-arm and the X-ray source interface, at similar mass and inertia. A comparison of the first approach (TMD) with the second, more classical approach, will be given in terms of the reduction in vibration levels during operation of the X-ray source.

Interventional Radiology, X-ray, Modal Analysis, Structural Vibrations, Tuned Mass Damper (TMD), Light-weight and stiff design

1. Introduction

Interventional X-ray systems are used in an increasing amount of procedures, across multiple medical disciplines. Typically, such systems feature an X-ray source and detector mounted to the ends of a C-shaped arm (see Figure 1). The C-arm allows the imaging equipment to be positioned in a range of orientations w.r.t. the patient. A relatively slender construction of the system's structural components aims to minimise obstruction to the physician's workflow. However, it also results in a small stiffness-to-mass ratio, leading to quasi-static deflections, and resonant behaviour at relatively low frequencies. Many of the related eigenmodes are effectively excited by the distinct disturbance forces from the X-ray source. The resulting vibrations of the system might cause acoustic noise, and could potentially degrade image quality. Therefore, this paper explores two methods, aimed at decreasing the resonant effects corresponding to the excited eigenmodes.

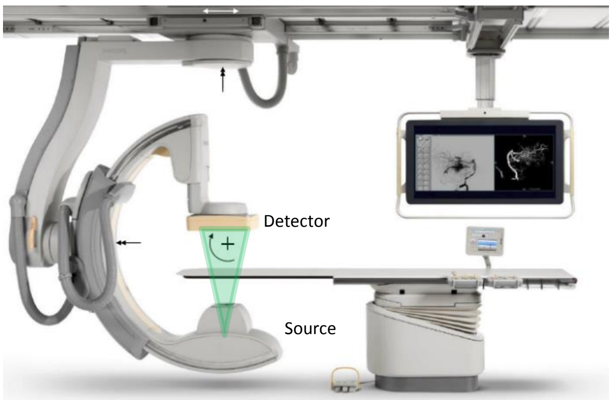

Figure 1. Interventional X-ray system, X-ray source at bottom of C-arm

The first approach aims to apply Tuned Mass Dampers (TMDs) to the C-arm. In [1], the potential of TMDs to decrease the magnitude of resonances, is used to achieve bandwidth increase in feedback controlled motion stages. In contrast to this work, we intend to apply TMDs to passively suppress the steady state amplitude of vibrations when the X-ray source is in use.

The second approach proposes local stiffness enhancement of the C-arm and the X-ray source interface, at similar mass and inertia. This approach aims to reduce the local deflection amplification towards the C-arm tip. Thereby making the eigenmodes less effectively excited by the X-ray source disturbance forces. A comparison of the first approach (TMD) with the second, more classical approach will be given in terms of the reduction in vibration levels during operation of the X-ray source.

2. Modal analysis: modelling versus measurements

A predictive model of the dynamic behaviour of the stand, in a frequency range near the disturbance frequency, is required to compare both research approaches. Therefore, an experimental modal analysis is performed using an impact hammer and multiple 3D acceleration sensors. Several experimental Frequency Response Functions (FRFs) are obtained, which relate X-ray source excitation to accelerations of points on the C-arm. Several eigenmodes are identified which have an eigen-frequency near the frequency of the disturbance forces. These eigenmodes are dominated by compliance in lateral (X-) direction (see Figure 2). The largest deflections occur at the tip of the C-arc.

A second modal analysis is performed using a Finite Element Analysis (FEA), schematically shown in figure 2. The compliance of the lower end of the C-arm and the X-ray source interface is found to be most dominant in the experimental modal analysis. As these parts of the system also contain intricate shapes, they are modelled using 3D elements. The remaining structural components and bearing points are modelled using beam elements (blue) and spring elements (red).

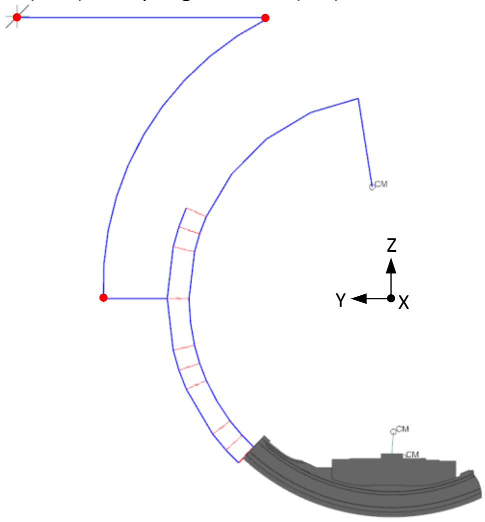

Figure 2. FEA model: lower part of C-arm is discretised using 3D elements. Beam (blue) and spring (red) elements are used to model other structural components and bearing points.

The state space matrices given by equation (1) determine the frequency response function, from a disturbance force at the X-ray source, to the acceleration of relevant points on the X-ray system [2]. Here, the modal mass and stiffness matrix ($\underline{M_0}$, $\underline{K_0}$), and matrix of eigencolumns ($\underline{U}$) are acquired in the finite elements analysis. The damping matrix $\underline{D_0}$ is defined by equation (2), where $\underline{Z_0}$ is a diagonal matrix, specifying the modal damping ratio (ζ) for each eigenmode.

$$\underline{A} = \begin{bmatrix} \underline{0} & \underline{0} \\ -\underline{M_0}^{-1}\underline{K_0} & -\underline{M_0}^{-1}\underline{D_0} \end{bmatrix}, \quad \underline{B} = \begin{bmatrix} \underline{0} \\ -\underline{M_0}^{-1}\underline{U} \end{bmatrix}$$
$$\underline{C} = \begin{bmatrix} \underline{U} & \underline{0} \\ -\underline{U}\,\underline{M_0}^{-1}\underline{K_0} & -\underline{U}\,\underline{M_0}^{-1}\underline{D_0} \end{bmatrix}, \quad \underline{D} = \begin{bmatrix} \underline{0} \\ \underline{U}\underline{M_0}^{-1}\underline{U}^T \end{bmatrix} \tag{1}$$

$$\underline{D_0} = 2\underline{M_0}\underline{Z_0}\underline{\Omega}, \qquad \underline{\Omega}^2 = \underline{M_0}^{-1}\underline{K_0} \tag{2}$$

To validate this model, the FRFs are compared to those acquired in the experimental modal analysis. Figure 3 illustrates this comparison, for a point on the C-arc, near the X-ray source. In the frequency range of interest, the modelled FRF (red) shows a strong agreement to the measured FRF (blue), both in magnitude, as in the location of resonance peaks. The accuracy of the model is therefore assumed to be sufficient for further analysis.

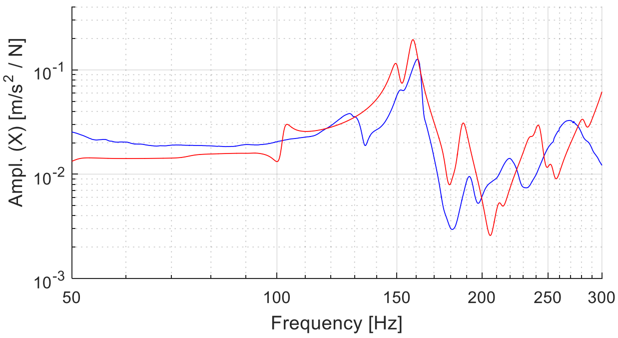

Figure 3. Frequency response function from X-ray source disturbance force to acceleration in x-direction of point on C-arc, blue measured, red: modelled.

3. TMD modelling

Figure 4 graphically depicts the modelling approach used to analyse the effect of applying TMDs to the system. The TMD (red frame) is modelled as a moving mass (m_m) and base mass (m_b), connected by a linear spring (c_d) and viscous damper (d_d). A connecting stiffness (c_c) allows a simple model to be derived, which expresses an FRF from the input displacement (x_c) to the spring force (F_c). The TMD model is connected to the system model (represented by mass: m_s) by cross-connecting the in- and outputs ($x_c = x_s$, and $F_s = F_c$). For the actual analysis, this model is extended in multiple directions, where c_d and d_d vary between the TMD's oscillating, and non-oscillating direction.

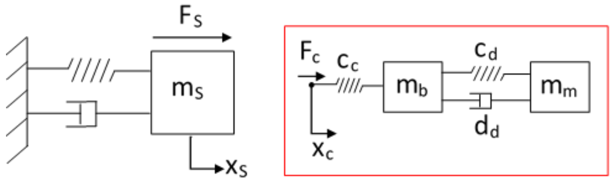

Figure 4. Graphical representation of TMD modelling

4. Conclusion

The aforementioned models are used to compare two methods aimed at decreasing the resonant effects due to the excited eigenmodes. The results of this comparison are shown in Figure 5. The red curve depicts the modelled FRF of the X-ray system. Two TMDs (m_m = 0.5 kg, m_b = 0.2 kg, ζ = 0.12) are added to the lower C-arc tip (green curve). The TMDs are tuned to a natural frequency of 158 Hz. The resulting suppression of the resonances is largest near this frequency. Due to the relatively high modal damping ratio (ζ = 0.12), the neighbouring resonance peak is also attenuated (Robust Mass Damper, [1]). Overall, the resonances between 140 and 200 Hz are reduced in magnitude by a factor 2 - 5. The black curve shows the behaviour of a design with increased stiffness of the C-arm tip and X-ray source interface. The additional local stiffness causes some frequency shift of resonances, and significantly reduces deflection amplification towards the tip of the C-arm. This causes a decrease of the FRF magnitude at frequencies between 100 and 200 Hz, which indicates that all eigenmodes in this frequency range are less effectively excited by the disturbance forces.

In future work, we intend to analyse the extent to which the reduced vibration levels affect acoustic noise, through both analysis and experiments.

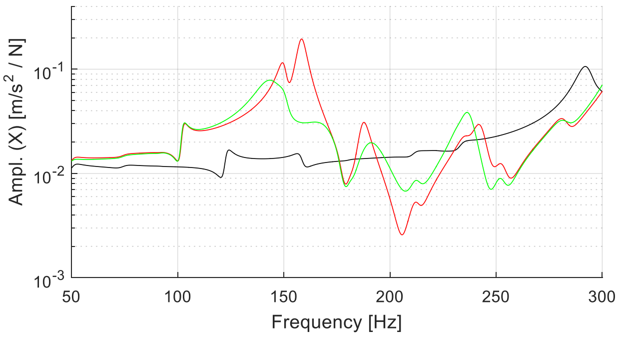

Figure 5. Modelled frequency response function from X-ray source disturbance force, to acceleration in x-direction of point on C-arc, red: original, green: application of TMDs, black: stiffness enhancement.

References

[1] Verbaan, C. A. M., 2015, Eindhoven. Robust mass damper design for bandwidth increase of motion stages. PhD thesis.

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