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Division V

DEVELOPMENT AND SEISMIC JUSTIFICATION OF A VIBRATION TUNED MASS DAMPER FOR NUCLEAR PIPES

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ABSTRACT

Mechanical vibrations in nuclear power plants (NPP) sometime require the implementation of on-site countermeasures limiting their amplitude. These countermeasures range from additional supports to viscous damping devices, each having their pros and cons depending on the concerned components and their requirements. One of the possible countermeasures, recently implemented on several piping systems of NPP worldwide, is the use of Tuned Mass Damper (TMD) attached to the line and working against the vibratory motion.

The implementation of a TMD to NPP piping lines poses some specific challenges which are a) the request of high material reliability, b) the need of robustness against slight frequency shifts of either the system or the excitation as well as against vibration amplitude changes, c) the demonstration of resistance to thermal transients and earthquake induced loads. These specific challenges led to the development of a TMD based on wire rope springs, that have the advantage of dissipating energy without requiring a viscous fluid, and linear guides, that allow controlling the movement direction, in vibration and earthquake situations.

This paper describes the development and justification of this TMD. It presents the concept of this piece of equipment, its modelling, its seismic qualification, and, finally, its demonstration of efficiency in the reactors where it was installed.

CONCEPT OF THE TMD FOR NUCLEAR PIPES

General principles for the design of a Tuned Mass Damper

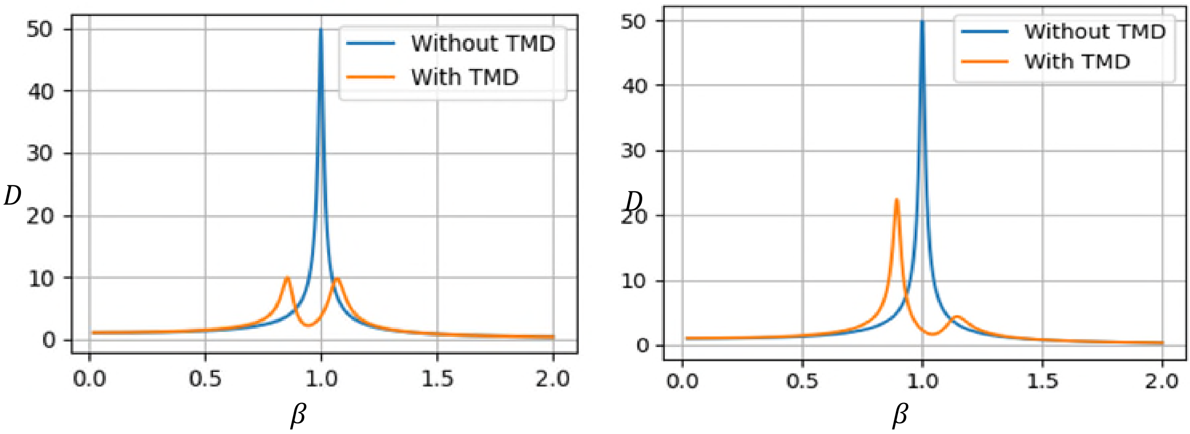
TMDs are a widely used vibration control technique in the non-nuclear industry. The fundamentals are described by Den Hartog (1956) or Harris et al. (1961). The objective of a TMD is to decrease the system vibration amplitude that are due to a resonance between the excitation forces and one of the systems eigenfrequencies.

A TMD is mainly composed of a) a mobile (moving) mass added to the system and b) a connection element between the mobile mass and the system. The mass value m_a and the properties of the connection elements are design parameters. Using a linear spring as a connecting element, its stiffness k_a and its damping ratio ξ_a are the basic properties. m_a and k_a are tuned in a way that the response frequency of the mobile mass alone ($f_a = 1/(2\pi)\sqrt{k_a/m_a}$), is slightly lower than the frequency of the original system (f_s). Den Hartog (1956) gives an estimate of the optimal frequency ratio κ_{opt} and an optimal damping ratio ξ_{opt} as :

$$\kappa_{opt} = \left(\frac{f_a}{f_s}\right)_{opt} = \frac{1}{1+\alpha} \quad \text{and} \quad \xi_{opt} = \sqrt{\frac{3\alpha}{8(1+\alpha)^3}} \quad \text{with} \quad \alpha = \frac{m_a}{m_s} \qquad (1)$$

From this initial estimate, a dedicated tuning is always necessary for the system and the excitation at stake.

For a system, represented by a single degree of freedom oscillator, whose original damping ratio ξ_s is 1%, Figure 1 shows the effect of adding a TMD. The two sides of this figure illustrate the differences between a "perfectly-tuned" system and an "imperfectly tuned" system. The abscissa in these graphs represents $\beta = \varpi/\omega_s$, the ratio between the excitation rotational frequency ϖ and the system eigen frequency $\omega_s = 2\pi f_s$. The ordinate is the dynamic amplification coefficient D, representing the ratio of the system dynamic response $max\ (u_s(t))$ to its static response $u_{s\_sta} = p/k_s$ for any given forced harmonic excitation $F(t) = p.sin(\varpi t)$. It is to be noted that the "imperfectly tuned" system of Figure 1 might be more efficient than the "perfectly-tuned" system if the excitation frequency is close to but slightly above the system eigenfrequency.

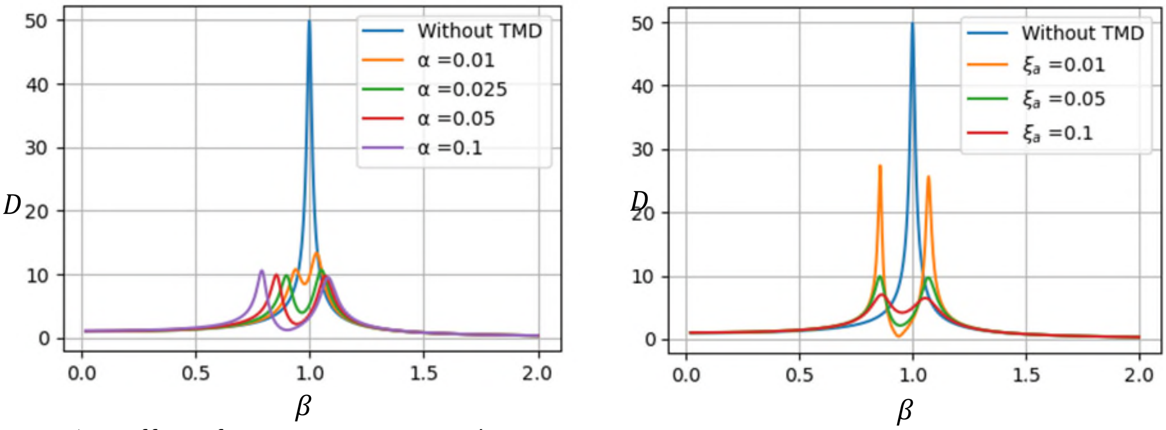


a) "perfectly tuned" TMD b) "imperfectly tuned" TMD

Figure 1. Dynamic amplification factor of a system with and without TMD

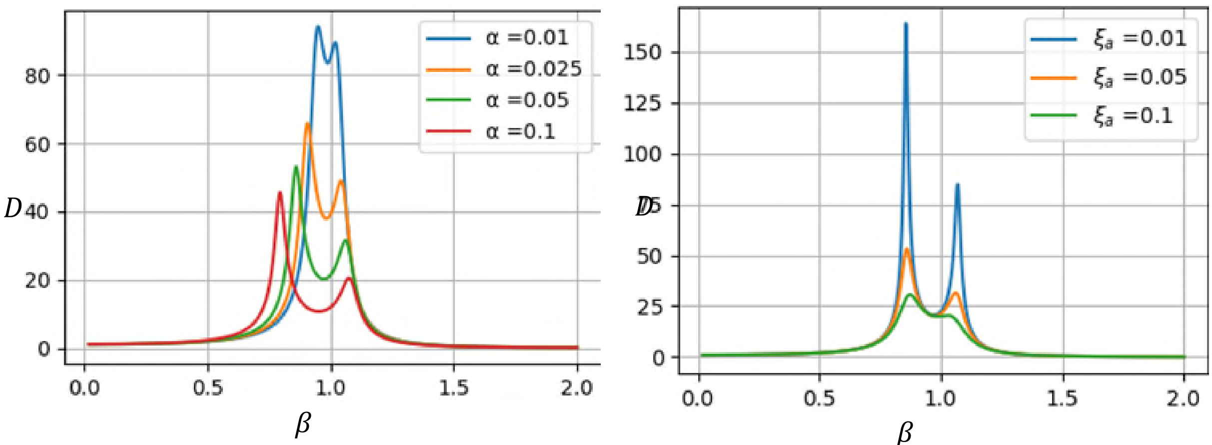
Increasing the mobile mass value m_a leads to a larger frequency band where the TMD is at its best effectiveness. It adds robustness to the system against unexpected small variation of either the system eigenfrequency or the excitation frequency. Increasing the TMD damping value ξ_a reduces the system maximum dynamic amplification factor on one side but increases it around the exact eigenfrequency of the system on the other side. The effects of varying the TMD mass and damping are illustrated in Figure 2.

Finally, as some significant vibration energy is transmitted from the system to the TMD, the TMD itself needs to have a sufficiently large damping ratio ξ_a and a sufficiently large mass m_a to limit its own displacement and make it physically acceptable. Figure 3 illustrates the effect of m_a and ξ_a on the dynamic amplification factor of the TMD, here representing the ratio between the dynamic response of the TMD, $max\ (u_a(t))$, and the static response of the system $u_{s\_sta} = p/k_s$.



a) Effect of mass ratio $\alpha = m_a/m_s$ b) Effect of damping ratio ξ_a

Figure 2. Effect of TMD mass and damping on the system dynamic amplification factor



a) Effect of mass ratio $\alpha = m_a/m_s$ b) Effect of damping ratio ξ_a

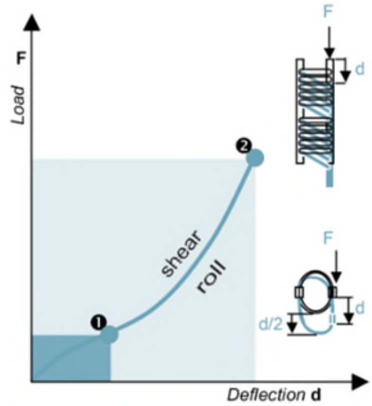
Figure 3. Effect of TMD mass and damping on the TMD dynamic amplification factor

Specificities of wire rope springs based TMD

Wire rope springs are manufactured by rolling a cable in loops between two jaws. The two jaws are the extremities of the spring and any relative movement between them induce a distortion of the cable loops that produce a reaction force in return, equivalent to a stiffness effect. The distortion of the cable loops also triggers a small amount of frictional sliding between the wires constituting the cable. This sliding dissipates energy in the form of heat and thus create a damping effect. A view of a wire rope spring is given in Figure 4a. Stiffness and damping of these elements are intrinsically nonlinear. The apparent stiffness decreases when going from small to larger amplitude of distortion of the loop (from the graph origin to point ① on Figure 4b). For even larger distortion though, the cables start working in tension and the stiffness increases again (from point ① to point ② on figure 4b).

The ability to implement stiffness and damping effects with a single fully metallic element makes this type of springs well adapted for the challenges of application within NPP. Its nonlinear behaviour on the other side, complicates the TMD design as both the apparent stiffness and damping are amplitude dependant.

a)



b)

Figure 4. Wire rope springs and force/deflection characteristics (courtesy SOCITEC)

Adaptation to nuclear pipes

The implementation of the concept of TMD with wire rope springs to nuclear pipe system requires some specific technological adaptation. A first constraint is not to expose the wire ropes springs to too high temperature and not to produce a cold point on the pipe. To achieve these objectives, the springs shall be kept out of the pipe thermal insulation. Indeed, some NPP pipes are heavily insulated, sometimes with insulation thickness as large as the pipe diameter itself. Therefore, a collar with radiating arms is installed on the pipe, with the arms going through the insulation to support the wire rope springs. The collar pre-stressing ensures the TMD stability through any type of thermal transient the pipe could experience.

To avoid that the TMD induce moments on the pipe due to the arms lengths, be it in vibratory or in earthquake situation, the mobile mass takes the shape of an annulus that circle the pipe and its insulation. With this shape, there is no eccentricity between the mobile mass centre of gravity and the pipe neutral axis. For the mobile mass to remain stable on the pipe, three attachment points are provided by three wire rope springs at the extremities of three arms. The mobile mass can be tuned by adding or removing some additional mass plates from the main mass annulus. Finally, to ensure directionality of work of the TMD and to stabilize it in case of earthquake, three supplementary arms are equipped with guiding system oriented in the direction of work of the TMD. A view of one TMD, installed on a test bench on a shaking table at the LNEC laboratory in Lisbon is presented in Figure 5.

Finally, the implementation of the TMD on a nuclear pipe requires it to have a predictable and well-defined behaviour. On-site trial and error type of TMD tuning, that are often used in the non-nuclear industry, shall be minimized in the nuclear industry. For this reason, predictive mechanical models of the TMD, and especially of the wire rope springs were developed and calibrated based on in-lab testing. Qualification against earthquake loads was also demonstrated by in-labs testing. These calibration and qualification are described in the following chapters.

MODEL OF THE TMD AND CALIBRATION

The development and calibration of a TMD model is a necessary step of the design process. It allows the selection of appropriate springs for any given eigenfrequency of the piping system as well as for the actual on-site observed amplitude of vibration.

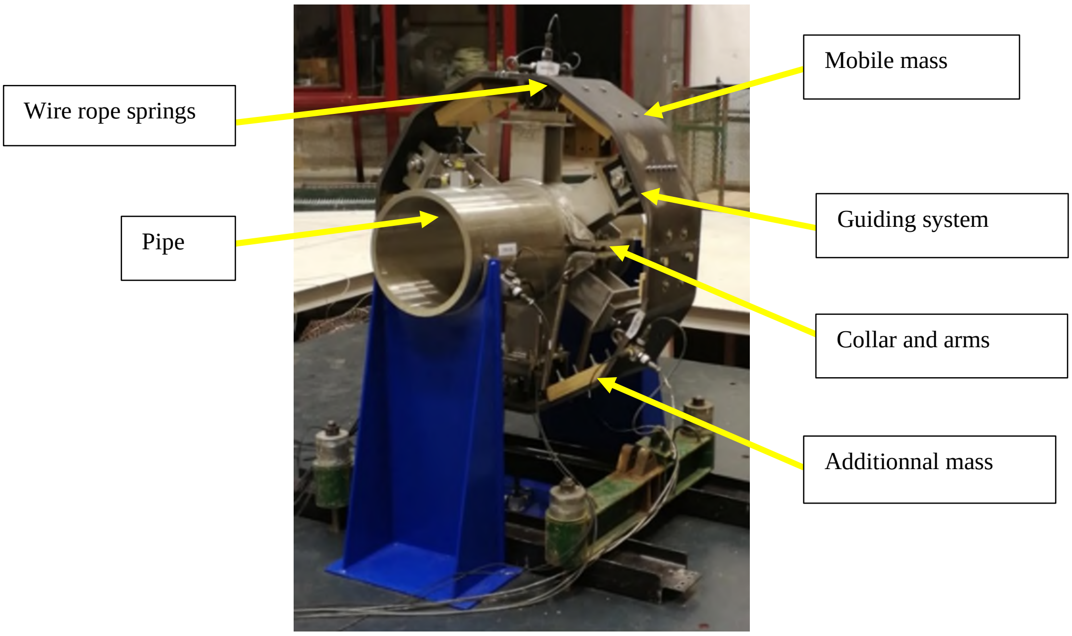


Figure 5. Tuned Mass Damper attached to a dummy pipe – Tests on shaking table

Mathematical models for the wire rope springs

The nonlinear nature of wire rope springs is extensively described in Gerges et al (2005) where 20 springs were tested and empirical formula for determining their stiffness and damping are proposed based on physical properties such as the cable diameter and the loop diameter. Still, this approach required the identification of parameters and their applicability to other type or technologies of spring is not straightforward. The industrial practice then remains to test each particular spring and determine its stiffness and damping values as a function of the cycling displacement amplitude it is submitted to. Based on these experiments, several authors have proposed mathematical models that can be calibrated for the representation of the dynamic behaviour of wire rope springs. Van Aanhold (1999) proposed the use of a Dahl model to represent the force distortion relationship. This model is represented by equation (2):

$$\frac{dF}{du} = k(1 + \frac{F}{F_f} sign\left(\frac{du}{dt}\right))^{\gamma} \qquad (2)$$

It links the force F to the distortion (or displacement) u and requires the identification of the exponent γ, the stiffness k, and the friction force F_f. Subsequently, Mangin et al (2012) proposed the use of a modified Dahl model with $\gamma = 1$ and varying stiffness $k(u)$ and friction force $F_f(u)$ as a function of the distortion amplitude. A good fit with cyclic test of wire rope springs is achieved with this approach. As the Dahl model was not readily available in the simulation tool used for this project, two alternative models were developed and compared to experimental results. The first model assumes of a linear viscous damping c_a associated to a variable stiffness element $k_a(u)$. Figure 6 shows the resulting force displacement curve, in black, as well as its decomposition into a damping force, in blue, and a stiffness force, in green.

The second model uses series of linear stiffness elements and friction elements as illustrated in Figure 7 below. The resulting force displacement curve is illustrated in Figure 8. It works approximately as a Dahl model.

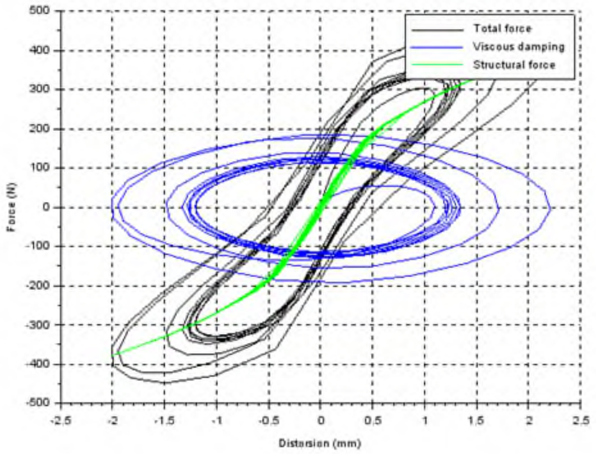


Figure 6. Model 1 for wire rope springs – Force distortion curve

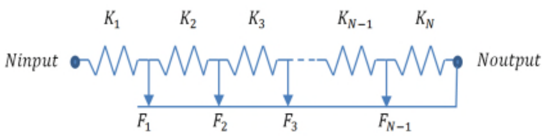


Figure 7. Second model for wire rope springs - Principle

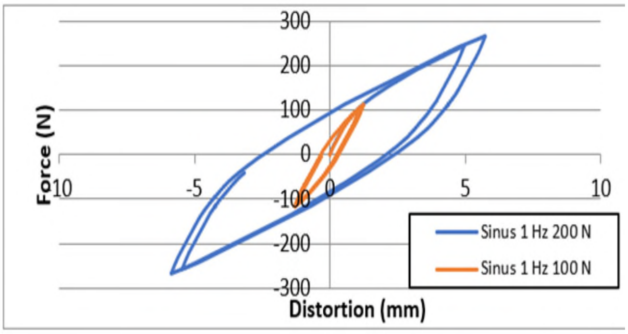


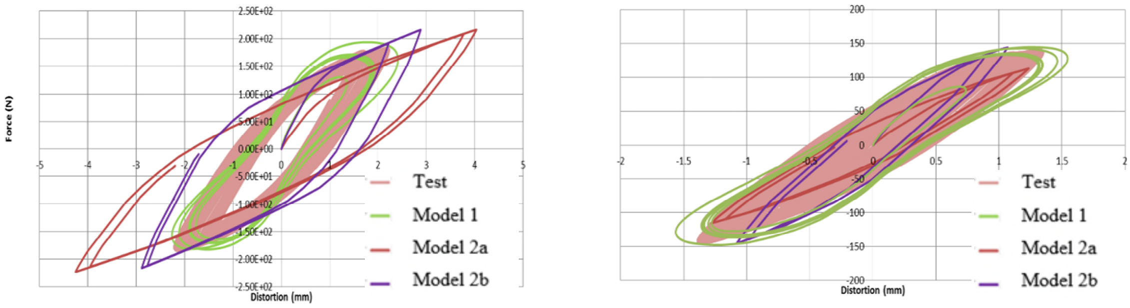
Figure 8. Model 2 for wire rope springs – Resulting force-distortion curve

Calibration tests and results

A full scale TMD was tested in the LNEC laboratory in Lisbon. The test set up is the one shown in Figure 5. A shaking table was used to impose a cyclic vibratory motion to a rigid fake pipe supporting the TMD. Two types of tests were conducted: a) Sine sweep excitations with constant displacements amplitude and b) Constant frequency sine excitation with time varying amplitude. The first type of test is designed to identify the TMD range of response frequency as a function of the amplitude. It allows the calibration of the stiffness and damping parameters of the mathematical model for the wire rope springs. The second type of test evaluates the range of efficiency of the TMD, in terms of amplitude, for the eigenfrequency of the piping system it is designed to damp.

Figure 9 shows two examples of results extracted from the sine sweep tests, with excitations of 0.5 and 1 mm of amplitude. In this figure three models are compared to the experimental results: Model 1 corresponds to the nonlinear stiffness and viscous damping as illustrated in Figure 6, Model 2a corresponds to the nonlinear model illustrated in Figures 7 and 8 and calibrated based on test data on a single wire rope spring and Model 2b is a variation of model 2a with increased stiffness, as model 2a was found to be too soft to represent the results obtained with the full TMD. In Figure 9, the Model 1 remains close to the experimental results whereas the Models 2a and 2b, despite having better looking outside shapes are further away from the target.

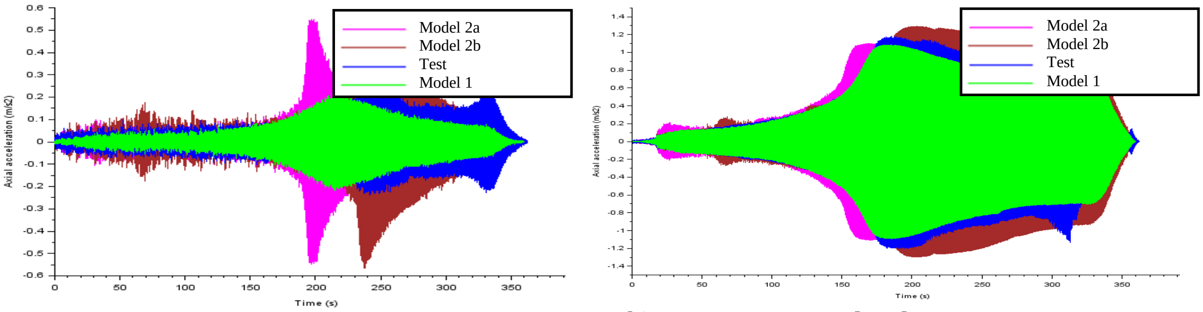
Figure 10 shows the time history response of the mobile mass through the full sine sweep tests with amplitudes varying from 0.1 mm to 1 mm. On each of the graphs displayed in this figure, the first phase of the TMD response is a pseudo static phase: at low frequency, the TMD is not active and simply follows the table movements. Then there is a steep increase in the TMD response as the excitation is approaching the TMD expected working frequency range. The steep increase of response occurs at lower frequencies, so sooner during the sine sweep, as the amplitude of excitation is increased. This is due to the softening behaviour of the springs. At higher frequencies, the TMD response decreases in a slow manner. The width of efficiency of the TMD, in terms of frequency, can be deduced from these graphs. In terms of mathematical models, none of them is fully satisfactory at an amplitude of 0.1 mm, but this correspond to a vibration level well below the one the TMD is designed to work against. Then the Model 1 tends to best predict the time of the steep response increase as well as the amplitude of the response.



a) Excitation amplitude 0.5 mm

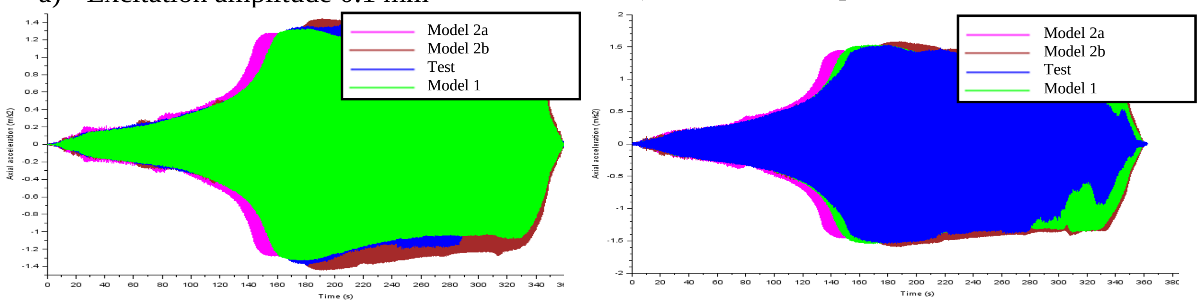
b) Excitation amplitude 1 mm

Figure 9. Result of sine sweep tests – Resulting force-displacement curve



a) Excitation amplitude 0.1 mm

b) Excitation amplitude 0.5 mm



c) Excitation amplitude 0.75 mm

d) Excitation amplitude 1 mm

Figure 10. Result of sine sweep tests – TMD response amplitude as a function of time

Figure 11 presents the result of one amplitude ramp-up test with constant frequency. The selected frequency is the one corresponding to the piping system eigen frequency to be damped by the TMD. These graphs clearly show one specificity of TMD using wire rope springs: it requires a certain excitation amplitude to be activated. Here this threshold amplitude is around 0.1 mm. below this value, the TMD is too stiff. Above this value, the TMD remains active on a relatively large range of excitation amplitude. For even higher vibration amplitude, as will be seen in seismic condition, the TMD is too soft and decoupled from the pipe response.

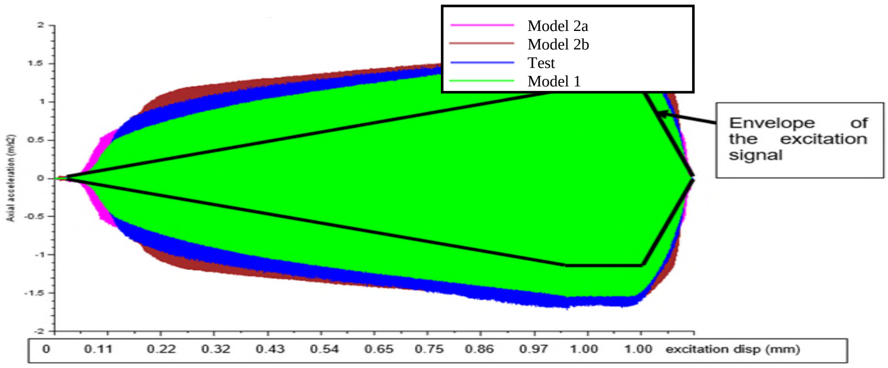


Figure 11. Result of amplitude ramp up test – TMD response amplitude as a function of excitation amplitude

SEISMIC QUALIFICATION OF THE TMD

The TMD is primarily designed for damping the vibrations of the piping system it is attached to. The proper design of the TMD ensures its maximum of efficiency for one given range of amplitude that is representative of the system vibration. In earthquake situations, the system displacements are one to several orders of magnitude higher than the vibration amplitudes. In this case, the wire rope springs experience distortion values that are largely above those experienced in vibration conditions and the models developed and calibrated for vibration are no longer adapted.

To characterize the behaviour of the TMD in seismic condition, the test bench illustrated in Figure 5 was again used. The applicable seismic motion was numerically determined by a time history dynamic analysis of the piping system and then experimentally reproduced with the shaking table. The amplitude of the seismic motion applied by the shaking table was progressively increased from 0.2 times the design basis earthquake motion to the full design basis earthquake motion. The tests were used on the one side to calibrate a new model of the wire rope springs under seismic condition, and on the other side to directly qualify the whole TMD construct against earthquake loads.

Figure 12 illustrates the distortion of the wire rope springs as a function of the increasing seismic motion amplitude, here expressed in terms of spectral value at the piping system eigenfrequency. The relationship between excitation and distortion is almost proportional, which allows to represent the wire rope springs as linear in the seismic analysis of the complete piping system with TMD and therefore greatly simplifies this analysis. The equivalent stiffness of the TMD in seismic condition is significantly below its stiffness in vibration condition. For this reason, this type of TMD does not bring any tangible benefit to the design of the piping system in seismic condition. As it is de-coupled from the system eigenfrequency, it does not induce any increase of the seismic loads either. In summary, the piping system seismic behaviour is only marginally affected by the presence of the TMD.

Finally, the shaking test up to the full-scale design basis earthquake allowed to directly qualify the TMD for seismic conditions.

IN REACTOR PERFORMANCE OF THE TMD

After the design, testing, model calibration and qualification phases, several TMDs were installed on actual piping system of NPP experiencing higher than expected vibration levels. In all cases, the TMD was designed to target the system eigenmode producing the highest vibratory response. Several acceleration sensors were positioned on the system and the vibration level was recorded before and after implementation of the TMD.

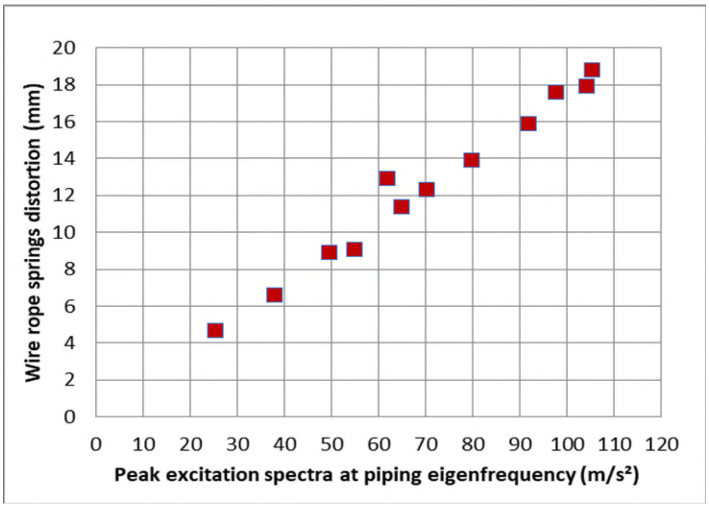


Figure 12. Result of seismic tests – TMD response amplitude as a function of excitation amplitude

In all cases, the TMD was able to decrease the vibration amplitude of the piping system on the targeted eigenmode by a factor 3 to 5. The piping system responses on its other eigenmodes, or on forced excitation far from its eigenmodes were only marginally affected by the addition of the TMD. Depending on the importance of these other modes on the system response, the overall maximum vibration amplitude was decreased by a factor 2 to 3 by addition of the TMD.

Figure 13 illustrates the gain obtained in terms of displacement amplitude (RMS) at one sensor location for 30 recordings slots of 60 s. Four different tuning of the TMD are compared to the reference situation without TMD. The root mean square (RMS) of the system vibration amplitude is reduced from 0.5 mm to about 0.13 mm. This gain is constant whatever the on-site tuning selected. The high damping provided by the wire rope springs makes the system robust against a small change of either the excitation frequencies or the system eigenfrequency, as was illustrated in Figure 2b. Figure 13 also shows that the TMD tends to make the vibration amplitude more stable in time: there is less variation from one recording time slot to another. Figure 14 shows the displacement power spectral density (PSD) at one sensor location, averaged over the 30 recordings of 60 s.

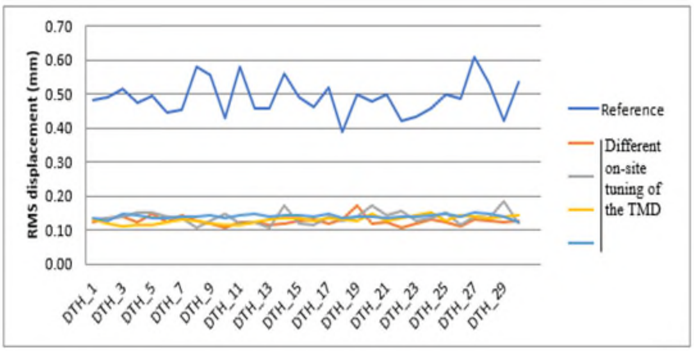


Figure 13. Example of application – Reduction of vibration amplitudes for 30 time slots of 60s

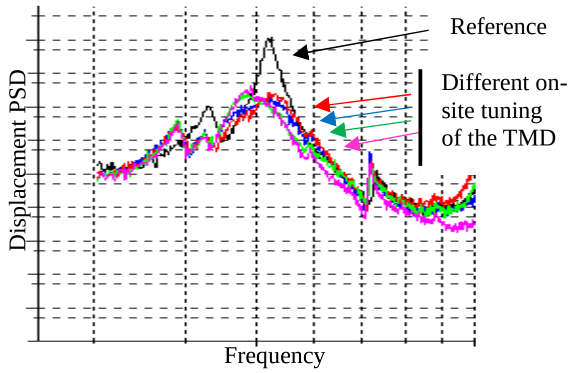


Figure 14. Example of application – Modification of the recorded power spectral density at pipe sensor location

The resonance peak of the piping system is clearly observed in the reference PSD curve in black in Figure 14. The effect of the different TMD on-site tuning is visible in the coloured curves. The resonance peak is damped out by the implementation of the TMD. The small changes in TMD mass and stiffness that were made as part of the tuning have a noticeable effect in this graph, even though their overall efficiency, as identified in Figure 13, is similar.

CONCLUSION

A specific kind of tuned mass damper, based on the use of wire rope springs, was developed for application on nuclear power plants piping systems. The use of purely metallic parts simplifies its acceptance in the nuclear industry and the relatively high energy dissipation capacity of the wire rope springs makes it robust against small possible changes of excitation frequencies or system eigenfrequencies.

As a drawback to this simplicity and robustness, the wire rope springs have a nonlinear behaviour, with stiffness and damping varying as a function of the distortion amplitude. This behaviour requires the use and calibration of specific models for the TMD design. In the present study, a simple model constituted of a multilinear force-displacement element associated to a linear viscous damper was found to be adequate. The calibration was performed with a scale one TMD submitted to series of shaking table tests representing the vibratory motion of the piping system.

In earthquake conditions, a TMD tuned to efficiently damp the vibration is not effective to damp the seismic motions. This is because the high distortion values induced by the earthquake soften the wire rope springs by comparison to small amplitude vibration conditions. The TMD is then mostly de-coupled from the system seismic response and it does not significantly influence this response. A combination of scale one shaking table tests and calculations were used to demonstrate the proper design of both the TMD and the piping system in seismic conditions.

The implementation of the TMD on several nuclear plants has shown its effectiveness in damping the system resonance peak for one given eigenfrequency. Several on-site tuning trials did all lead to satisfactory results. From these experiences, the TMD reduced the piping system response of the targeted mode by a factor 3 to 5. The overall level of vibration, which is usually not due to a single mode responding, has been reduced by a factor 2 to 3. Finally, the wire rope based TMD was found to be a light and adequate countermeasure to decrease the vibration level of a piping system without affecting its seismic and thermal-mechanics design.

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