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# THE CONTROL OF VIBRATIONAL POWER TRANSMISSION IN ONE-DIMENSIONAL STRUCTURES

P Clark (1) and R G White (1)

(1) Institute of Sound and Vibration Research, University of Southampton, Southampton, UK

## 1. ABSTRACT

When attempting to control the vibration transmitted from machinery installations, perhaps with a view to reducing the unwanted radiation of noise at a point remote from the source, it is essential that all possible transmission paths are considered. With the majority of installations, one-dimensional structures such as beams, pipework vibrating at low frequencies and other mechanical linkages, form some of the main vibration paths which bypass isolator systems. It is of interest to consider different types of discontinuity that may be incorporated into these types of structure as they significantly affect the vibration transmission properties of the complete system. One such discontinuity that may be employed as a vibration control device is the vibration neutralizer. In this work, the vibration attenuation performance of these types of device fitted to one-dimensional structures is considered using the concept of vibrational power transmission.

## 2. INTRODUCTION

The control of vibrations in beams or other one-dimensional structures is of considerable interest in practical engineering. With the majority of industrial machinery installations, it is this type of structure, for example, pipework vibrating at low frequencies and other mechanical linkages, which forms one of the main vibration paths which bypass isolator systems. In addition to this, it is well known that the introduction of discontinuities into these types of structure significantly affects the vibration transmission properties of the complete installation. One such discontinuity that may be employed as a vibration control device is the vibration neutralizer (often referred to as a "Dynamic Vibration Absorber"). Previous research in this area [1, 2] has employed both theory and numerical parameter studies into the effects of utilizing these types of device for practical vibration control purposes. In these studies, both axial and flexural vibration were considered and the vibration characteristics of the device were assessed using the concept of vibrational power transmission. In this work, complementary experimental studies are presented with a view to validating those previously presented theoretical results.

In the experimental work, axial vibrations only are considered and practical axial vibration neutralizer devices have been designed, built, fitted to an experimental pipeline structure and their vibration attenuation performance examined using the concept of vibrational power transmission.

## 3. THEORY

For the purposes of the previous analytical studies, the neutralizer was considered to be a simple lumped mass/hysteretically damped spring single degree of freedom system attached to an infinite beam-like structure subjected to axial motion by a harmonic force at a point remote from the location of the discontinuity. A schematic representation of the general beam discontinuity is given in Fig. 1 whereby an impinging axial wavefield, $A_i$ impinges on the general discontinuity, of arbitrary dynamic stiffness, $DS_{aux}$, which then gives rise to a transmitted wave, $A_4$, and a reflected wave, $A_3$. Fig. 2 concerns the specific discontinuity of interest in this study, the vibration neutralizer. In this work, the neutralizer comprises a lumped mass element, $M_d$, which is then attached to the primary beam-like structure by means of a light, hysteretically damped spring, $k_d = k_d'(1 + j\eta)$ where $\eta$ represents the spring loss factor.

By considering the conditions of continuity and equilibrium at the joint the resulting wave amplitude reflection and transmission coefficients can be found. From a knowledge of the respective reflected and transmitted wave amplitude coefficients, proportions of reflected and transmitted axial vibrational power can be determined and the resulting performance of the device ascertained by considering a measure of the attenuated vibrational power transmitted.

## 4. EXPERIMENTAL APPARATUS AND METHOD

The experimental structure consisted of a straight 2 inch nominal bore axially driven empty perspex pipe, approximately 3.5 m long*. The pipe was mounted vertically and drive axially at one end using an electrodynamic exciter. In order for the experimental properties of the structure to be "semi-infinite", the opposite end of the pipe was fitted with an anechoic termination. This termination was composed of a series of flat plates sized such that total flexural mechanical impedance of these plates matched the axial mechanical impedance of the pipe. Good dissipation of energy in the plates was then achieved by ensuring that the dimensions of the plates were at least one-quarter wavelength of the lowest frequency of interest in each direction [4]. To ensure that the resulting dynamical behaviour of the structure was as predicted, the driving point accelerance of the experimental pipe was measured. Fig. 3 shows the result which compared well with theory for the frequency range from 30 Hz to 300 Hz.

To facilitate the mounting of a neutralizer device onto the pipe wall, the experimental structure was also fitted with a flange set approximately 2 m from the anechoic termination. A schematic representation of the complete experimental apparatus is given in Fig. 4. During the experiments the pipe was driven at its free end and the resulting driving point input force and acceleration measured. In addition to this the response of the pipe wall downstream of the mounting flange was measured using a combination of two accelerometer pairs. The response of the pipe wall was determined at each position using a pair of accelerometers since a simple arithmetic mean response of the accelerometer pair could then be used to negate the effects of additional wave-types in the structure should they be induced. The resulting vibrational power transmitted "downstream" of the mounting flange was then determined using a finite difference technique [5] with and without the experimental neutralizer in position. From these results the vibration attenuation performance or insertion loss of the device was determined.

The experimental neutralizer consisted of a mild steel ring (or rings) attached to the perspex pipe flange by a set of four aluminium alloy "cantilevered" beams. The resulting neutralizer mass and natural frequency could then be varied in discrete intervals by incorporating one or more mild

* At frequencies well below the ring frequency of the pipe only beam-like motion can occur and so the pipe can be considered to exhibit the dynamical properties of a simple beam or rod [3].

steel rings. A diagrammatic representation of the axial vibration neutralizer device used in the experiments is given in Fig. 5.

Although simple lumped parameter models are useful for predicting the approximate frequency range of operation for these types of device, the resulting installed natural frequency is highly dependent on the mounting stiffness of the device when attached to the primary structure since the ideal assumptions used for the basic design would not be realized in practice [6]. It therefore follows that the installed natural frequency is most effectively determined in situ. For this programme of work, respective natural frequencies were determined by considering the response of the pipe wall at the point of attachment of the device relative to the response of the neutralizer mass. Throughout the frequency range of the experiment, the response of the neutralizer mass remained finite, however at the natural frequency of the device, the response of the pipe wall becomes a local minimum. Therefore, the natural frequency and damping terms of the neutralizer were determined by considering the response of the neutralizer mass relative to response of the pipe wall adjacent to the device.

## 5. RESULTS

In the first experiment, an axial vibration neutralizer was fitted to the pipe flange and the relative response of the mass and pipe wall determined, Fig. 6. The natural frequency and damping terms were then found by considering the ratio of the pipe wall response to that of the neutralizer mass shown in Fig. 7. The natural frequency was then found to be 160.5 Hz and the damping loss factor, using a half power point method, 0.023. The neutralizer mass was then accurately measured and found to be 1.01 kg. The predicted attenuation performance of the device was then determined using the theoretical model developed in [1, 2] and the resulting experimental and theoretical performances are compared in Fig. 8. In each example, the resulting spring stiffness could have been determined from a knowledge of the natural frequency and mass of the installed device; however, for the purposes of comparing theoretical and experimental results a knowledge of the neutralizer spring stiffness is superfluous.

In addition to the above, a second experiment was performed using a neutralizer with a larger mass of 2.02 kg. The resulting dynamical properties of this device were determined in the previously described manner such that the natural frequency and damping terms were found to be 126.25 Hz and 0.029, respectively. The resulting comparison between predicted and measured performance is given in Fig. 9.

In the previous experiments, the neutralizer mass was formed from a continuous ring (or rings) of mild steel fitted around the pipe by splitting the ring in up to four places and rejoining it when attached to the flange. In the final experiment the neutralizer was fitted to the flange with the joints left unfastened such that the device consisted of four discrete masses. The aim of this experiment was to assess the possibilities for a tuneable device whereby the mass elements could be moved along the spring elements to vary the natural frequency. The comparison in attenuation performance for a single mass versus multiple mass system is given in Fig. 10. The results indicate that the installed natural frequency of the device was approximately 90 Hz, however, by simply adjusting the device to unfasten the joints in the mass the natural frequency reduced to approximately 85 Hz, therefore indicating the high dependence of natural frequency on mounting stiffness. Finally, the attenuation performance is slightly reduced for the multiple mass system, a result which was probably caused by minor asymmetries occurring in the device which resulted in additional wave-types being induced in the pipe which therefore reduces neutralizer effectiveness [7].

## 6. CONCLUSIONS

In this paper, measurements of the attenuation performance of an axial vibation neutralizer have been reported. Corresponding theoretical predictions have indicated good agreement with experimental behaviour, provided the dynamical properties of the device are determined "in situ". Initial experiments to assess the possibilities for a tuneable axial device have produced promising results, though care must be taken to ensure the device is mounted symmetrically on the pipe.

## 7. ACKNOWLEDGEMENT

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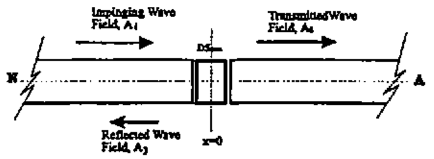


Figure 1: Uniform infinite beam in axial vibration fitted with an arbitrary point discontinuity.

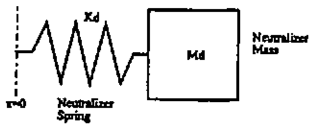


Figure 2: The vibration neutralizer discontinuity.

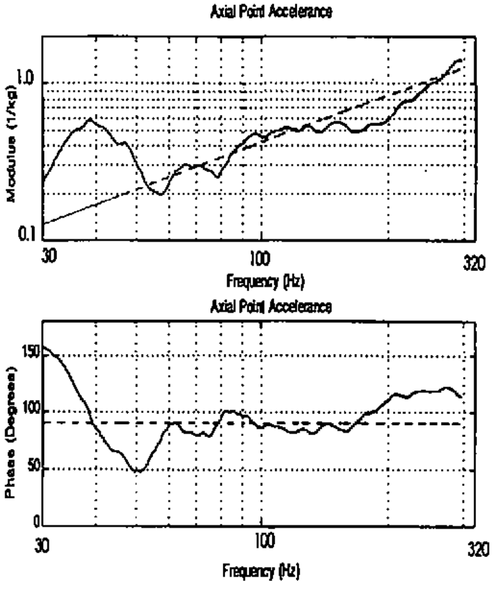


Figure 3: Low frequency axial point accelerance of experimental "semi-infinite" pipe. Solid line: measured; dashed line: predicted.

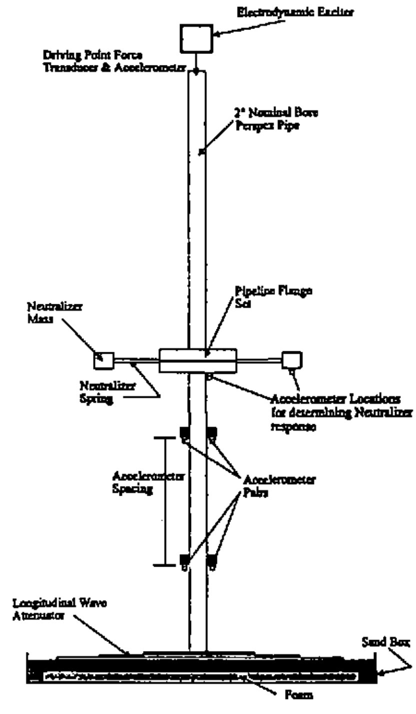


Figure 4: Schematic representation of experimental semi-infinite pipe apparatus fitted with axial pipeline neutralizer.

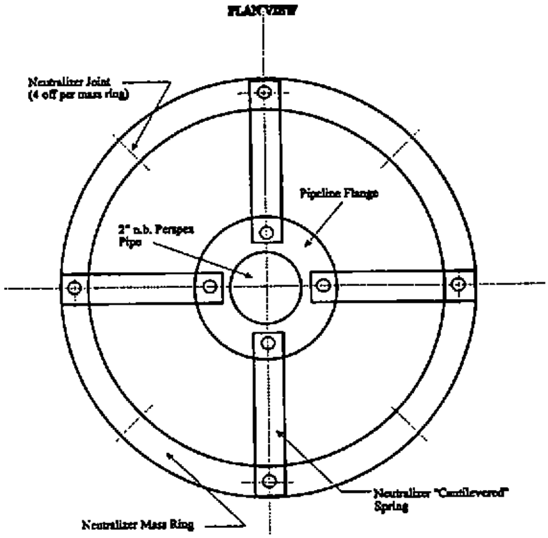


Figure 5: Schematic representation of the pipeline axial vibration neutralizer.

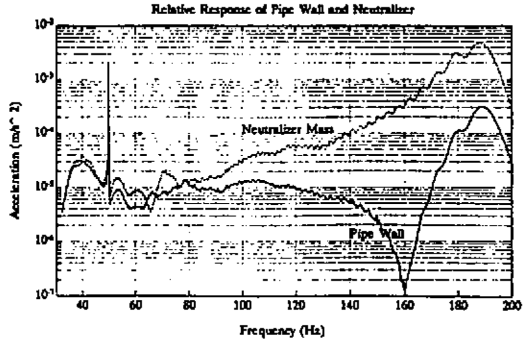


Figure 6: Relative response of neutralizer mass to adjacent pipe wall – acceleration (m/s$^2$).

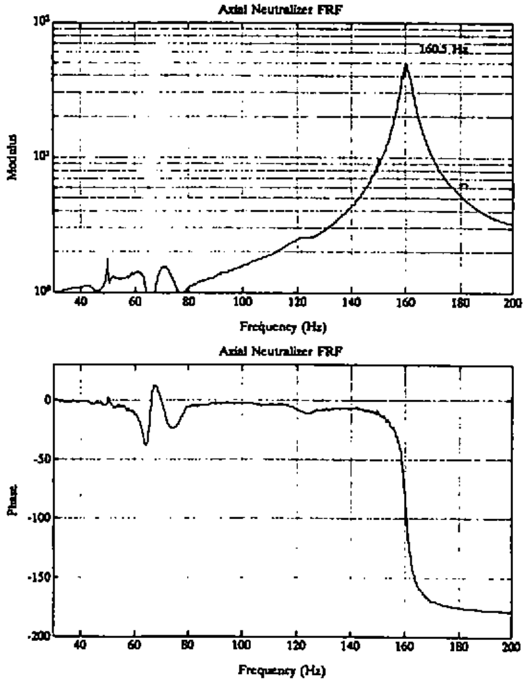


Figure 7: Derived frequency response function of the installed neutralizer – modulus and phase.

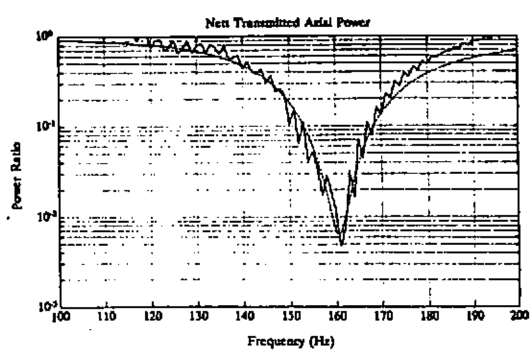


Figure 8: Predicted v. measured attenuation from "light" axial neutralizer. Solid line: measured; dashed line: predicted.

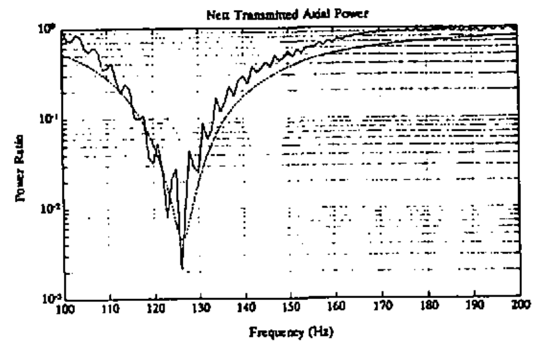


Figure 9: Predicted v. measured attenuation from "heavy" axial neutralizer. Solid line: measured; dashed line: predicted.

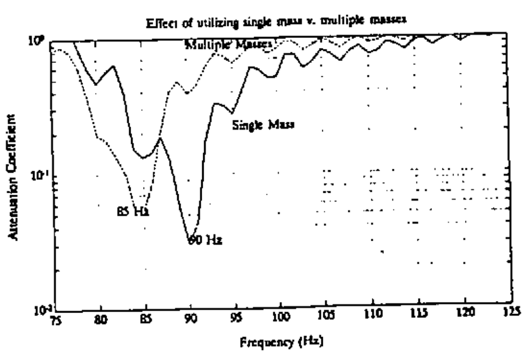


Figure 10: Assessment of the effects of using multiple masses v. single neutralizer mass.