THE CONTROL OF FLEXURAL POWER TRANSMISSION IN PIPELINE STRUCTURES

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1. INTRODUCTION

The control of vibrations in beams or other one-dimensional structures is of considerable interest in practical engineering. With the majority of industrial applications 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] has employed both theoretical and numerical parameter studies on the effects of utilising these types of device for practical vibration control purposes. In this work complimentary experimental studies are presented with view to validating those previously presented theoretical results.

In the experimental work considered here flexural vibrations only are considered and it is shown that by using the concept of vibrational power transmission it is possible to design and test practical flexural vibration control devices on an experimental pipeline structure.

2. THEORY

For the purposes of the previous flexural vibration analytical studies, various mounting configurations were proposed for the vibration neutralizer. In all the cases considered, however, the basic device was

1 This work was carried out whilst both authors were at the ISVR, University of Southampton, UK.
considered to be a simple lumped mass/hysteretically damped spring single degree of freedom system attached to a beam-like structure subjected to flexural motion by a harmonic force remote from the location of the discontinuity. Three mounting configurations were considered in the theoretical work, first a simple “force-only” device, secondly the corresponding “moment-only” device and finally a combined “force and moment” device. The theoretical studies presented in [1,2] have shown the combined force and moment device is capable of offering the highest degree of vibration attenuation for a given neutralizer mass. It therefore follows that this design of flexural neutralizer was utilised in this experimental work. Accordingly the overall schematic representation of the beam and neutralizer discontinuity is shown in Fig. 1. The neutralizer can be seen to comprise of lumped mass element, \( M_n \), which is then attached to the primary vibrating structure by means of a light hysteretically damped spring, \( k_n = k_n' (1 + j\eta) \), where \( \eta \) represents the spring loss factor, and a light rigid moment arm of length, \( a \).

By considering the conditions of equilibrium and continuity at the joint it is possible to determine the resulting reflected and transmitted wave amplitude coefficients. From a knowledge of these wave coefficients the resulting proportions of transmitted and reflected vibrational power can be determined and the performance of the device can then be assessed in terms of its ability to attenuate vibrational power transmitted to a position downstream of its location.

3. EXPERIMENTAL APPARATUS AND METHOD

The experimental structure consisted of a straight 2 inch nominal bore empty PERSPEX pipe approximately 3.5 m long. The pipe was mounted vertically and driven in flexure at the free end using an electrodynamic shaker. 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 consisting of an exponentially shaped "sock" filled with dry sand.

To facilitate the mounting of the device on the pipe wall, the experimental structure was fitted with a flange set approximately 2 m from the anechoic termination. During the experiments the driving point input force and acceleration were measured. In addition to this, the response of the pipe wall downstream of the flange set was measured using a combination of accelerometer pairs. The vibration of the pipe was determined at each

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\[ \text{At frequencies well below the ring frequency of the pipe only beam-like motion can occur and so the pipe can be considered to exhibit the dynamical properties of a simple beam or rod [3].} \]
measurement position by considering the response of a pair of accelerometers mounted orthogonally on the pipe wall. In this way the effects of extraneous wave-types in the structure could be minimised [4].

The experimental neutralizer consisted of a mild steel ring fitted around the pipe attached to the mounting flange by a series of aluminium alloy cantilevers. A diagrammatic representation of the complete installation is shown in Fig. 2.

Although simple lumped parameter models are useful for predicting the approximate frequency range of operation for these types of device, the natural frequency of the installed device is highly dependent on the actual mounting stiffness. In order to overcome these problems a simple method of determining the installed natural frequency was developed [2] by considering the relative responses of the pipe wall at the point of attachment with respect to the neutralizer mass.

4. RESULTS

First the flexural attenuation device was fitted to the pipe flange and the relative response of neutralizer mass and pipe wall were measured, Fig. 3. From this result the natural frequency was found to be 46 Hz with a damping loss factor of 0.054. The neutralizer mass was then accurately measured and found to be 0.82 kg whilst the installed moment arm was 0.075 m. The predicted attenuation performance was then determined using the theoretical models developed in [1,2] and the resulting theoretical and experimental attenuation performances are given in Fig. 4.

5. CONCLUSIONS

In this work, measurements of the attenuation performance of a flexural vibration neutralizer have been presented. Corresponding theoretical predictions have indicated reasonable agreement with experimental behaviour, provided the dynamical properties of the device are determined “in-situ”.

6. ACKNOWLEDGEMENT

The work has been carried out with the support of the Marine Technology Directorate Ltd, the Science and Engineering Research Council, and the Procurement Executive, Ministry of Defence.
7. REFERENCES


FIGURES

Fig. 1 Schematic representation of Neutralizer mounted on the Pipe.

Fig. 2 Schematic representation of the pipe apparatus.

Fig. 3 Frequency response of the installed neutralizer.

Fig. 4 Measured v. predicted attenuation. Solid line : expt; dashed : theory.