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EFFECT OF WALL-AXIAL WAVE COMPONENT ON THE ACTIVE CONTROL OF WATER-BORNE NOISE IN STEEL PIPES

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ABSTRACT

Active control of low frequency pump noise in water-filled pipes is being developed at AMRL for acoustic signature control on maritime platforms. The present approach is based on separate fluid and wall-axial wave control with collocated control sources. One of the main problems encountered in the design of suitable control sources is their operating effectiveness over a large range of internal hydrostatic pressures. A prototype of a fluid-wave control source has been built which is able to provide some control at low frequencies in a steel pipe system. However, at higher frequencies, the control is considerably compromised through a mechanism believed to be linked to flanking via wall-axial wave components. This paper reports the results of experiments designed to study the effect of flanking with the ultimate aim of establishing requirements for the design of a complementary source for the control of wall-axial waves.

INTRODUCTION

Active control of noise and vibrations is being investigated in the Maritime Platforms Division of the Aeronautical and Maritime Research Laboratory as a means of enhancement of acoustic signature reduction on naval platforms. Active control systems are expected to offer critical advantages in the low frequency regime as a compact, light-weight and high performance supplement to passive measures. One particular aspect of active control under investigation is the reduction of low frequency pump noise transmission through a system of water-filled steel pipes and flexibles which may be connected to the sea.

The virtues of an active system applied to the control of pump noise in pipes filled with heavy fluid have been discussed by several investigators [1-4]. Nearly all have recognised the advantage of developing a control source which would act externally on the pipe and induce the required pipe-wall and fluid displacements for the control of both the fluid and the structural wave components. However, some of the more recent experimental investigations [2,4] were performed on perspex pipes which, when compared to steel pipes used on naval vessels, are more amenable to deformation by external forces. The approach to active cancellation of noise in water-filled pipes reported by Harper and Leung [1] is broadly similar to that pursued at AMRL, where separate control actuators are employed in the reduction of fluid- and wall-borne noise.

The work presented in this paper was carried out with the aim to establish the effectiveness of a fluid-wave control source [5,6] incorporated in a steel piping system filled with water. In the experimental investigation, a single channel active control system incorporating a fluid-wave control source was employed to reduce noise propagation through a 12 m long water-filled steel pipe, with nominal inner diameter of 100 mm and wall thickness of 6 mm. Two configurations were used where the control source was terminated by a flexible PVC composite pipe in the first series of experiments, and by a steel pipe in the second series. Results show that the presence of wall-axial modes strongly impaired the control authority of the fluid-wave control source and that an additional, largely independent, control source for wall-axial waves would be required for the effective control of noise propagation in the water-filled steel pipe.

BACKGROUND

A water pump on a naval vessel, connected to the sea through steel and flexible piping is expected to exert vibrational forces on the pipe structure and generate acoustic pressure waves in the pumped fluid. The coupling between the fluid and the structure may be quite complex depending on the piping geometry, inclusion of other components such as passive mufflers, valves, particle filters and resonant absorbers and the use of different materials for the various sections of the piping.

In our experiments, we were mostly concerned with propagation of zero order mode waves in the fluid, and this has an inherent coupling associated with it where the fluid and wall-axial waves are coupled through a Poisson-type mechanism. This has been described by many investigators and is also explained in References [2, 3, 4, 6, 7, 8]. The 'acoustically slow' fluid wave is often referred to as the $s = 1$ wave, and the 'acoustically fast' wall-axial wave is referred to as the $s = 2$ wave. From the view point of active control, it is significant that wall axial waves can generate acoustic pressures in the fluid through Poisson coupling and that this occurs at a wave speed different to the fluid-wave speed. Since a fluid-wave control source generates a cancellation wave at the '$s = 1$' speed, it will not be able to achieve global downstream control for a mixture of waves with '$s = 1$' and '$s = 2$' speeds. Provided that only zero order modes are excited, the active control of noise in a water-filled steel pipe will require control of both the fluid-acoustic and wall-axial wave components.

In naval vessels, it is preferable to use passive control measures as these are inherently more robust and reliable. Brennan et al. [4] made reference to work of de Jong [9] where it was shown that flexible bellows section fitted in an unpressurised pipework system is effective in suppressing the propagation of fluid pulsations generated by a rotary pump and that in practical pipework systems, such flexible sections were less effective in a pressurised system due to stiffening. The work by Podlesak [10] and Kuhl [11] shows that flexible pipe sections could be designed with a specific absorption frequency band and at the same time reduce the effect of wall-axial waves. However, these would also become ineffective due to stiffening in a pressurised system.

Therefore, from the viewpoint of naval platform operations, a hybrid passive-active approach is likely to provide the most favourable solution to the control of pipe-borne noise, where passive components of the noise control system provide a suitable performance baseline, and further noise reductions are achieved via active measures which can also compensate for any loss of passive performance due to pressurisation.

EXPERIMENTAL SET-UP

A schematic diagram of the experimental set-up is shown in Figure 1. The basic set-up consisted of a 0.25 m long stainless steel pipe section housing the noise source, a 12 m long steel noise transmission section, a 0.2 m long steel control source section, a 0.32 m long steel sensing section, and a terminating section. In the first series of measurements, the terminating section comprised a 3.7 m long composite PVC pipe [10] with diecast metal flanges, and in the second series, a 3 m long steel pipe with a 0.48 m long steel elbow section attached to it. In all measurements, the various sections were horizontally supported and bolted together via 10-12 mm thick flanges with 215 mm outer diameter and lined with a flat 3 mm thick rubber gasket. The noise generating end was terminated with a 12 mm thick blind flange and the end

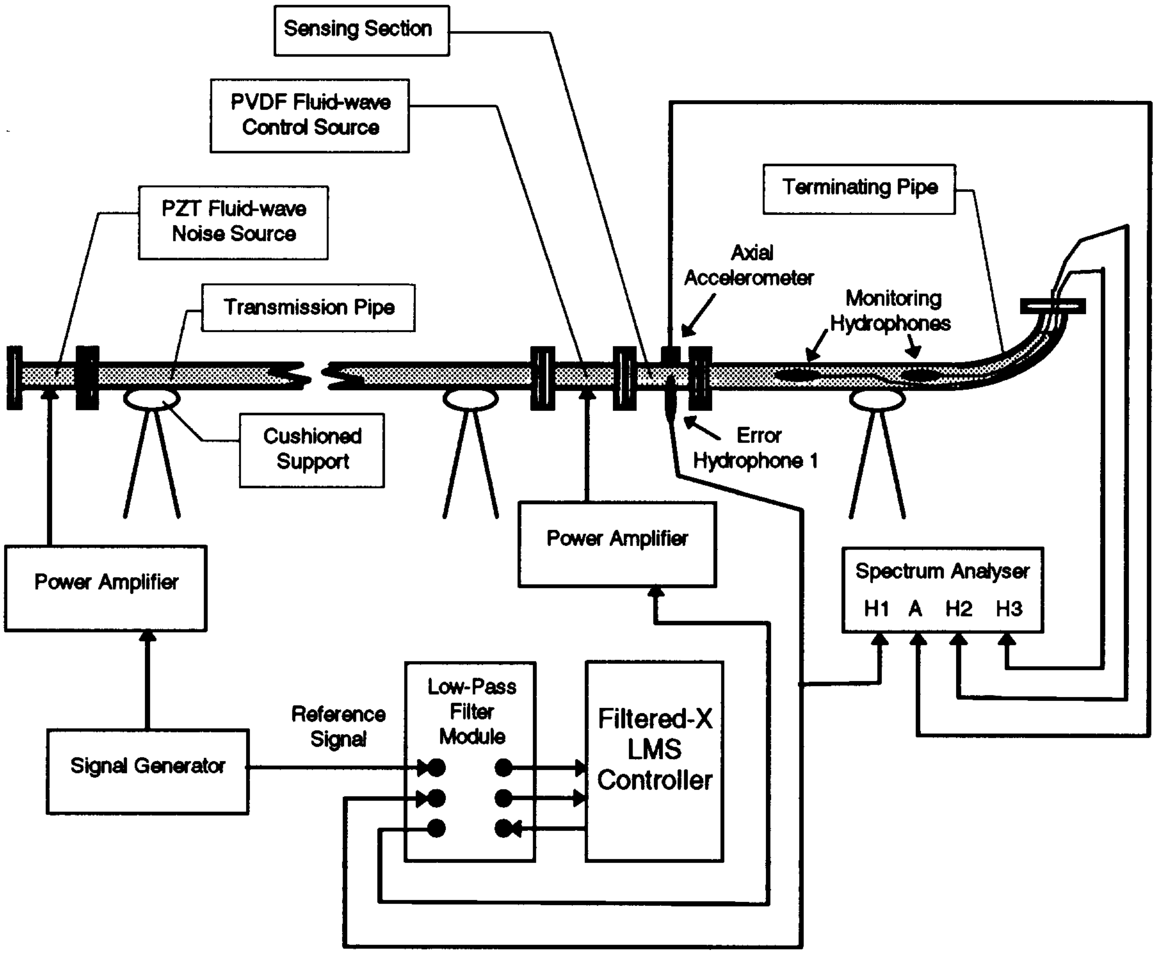


Figure 1: Schematic diagram of experimental set-up

of terminating section was turned up to provide an open boundary for the water in the system.

The noise source section housed a 100 mm long and 3 mm thick PZT cylinder with 100 mm outer diameter. The cylinder was mounted compliantly and concentrically on the stainless steel pipe wall and was driven by Analogic 2030 Multifunction Waveform Synthesizer via a B&K Type 2713 power amplifier.

The control source section had a centrally mounted 0.1 mm thick PVDF (Polyvinylidene fluoride) film cylinder, 100 mm long, with nominal mean diameter of 100 mm. It was compliantly attached to the inner steel cylinder wall with silicone adhesive and driven by the adaptive controller via a second B&K Type 2713 power amplifier. Design and performance of the PVDF device is described in references [5, 8, 12].

The sensor section housed two sensors. The first was a B&K Type 8103 hydrophone, denoted here as H1, which was mounted radially in the pipe wall and used as the error sensor in the adaptive control system. The second sensor was a B&K type 4343 accelerometer mounted horizontally on a 15 mm cube Aluminium block attached to the pipe wall which enabled the measurement of mainly axial acceleration of the pipe wall. Both the accelerometer and the hydrophone were mounted 315 mm away from the nearest end of the PVDF film cylinder.

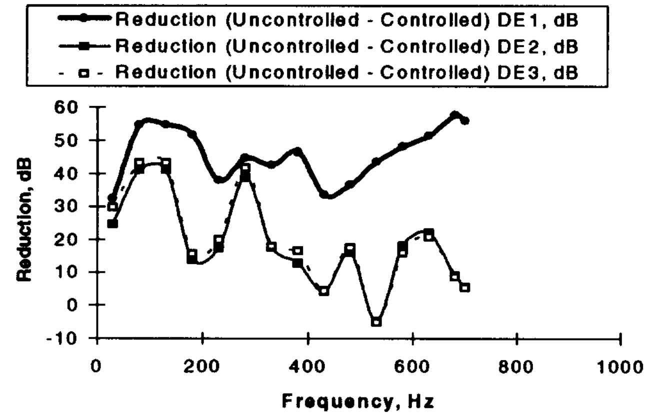
The terminating pipe contained two movable B&K Type 8103 hydrophones, labelled as H2 and H3, equipped with an additional brass mesh grounding shield to suppress mains frequency electrical interference. These were used primarily for monitoring of the fluid-wave field downstream from the error sensor, with H3 position fixed at 3.00 m and H2 varying between 1.00 and 3.00 m from the open end of the pipe. However, at the 3.00 m position, H2 was slightly offset from H3 to avoid physical contact of the hydrophone grounding screens in order to avoid strong mains frequency interference. All hydrophones were coupled to B&K Type 2635 charge amplifiers, with low-pass filter setting on 1 kHz.

The adaptive single channel Filtered-X LMS controller is based around a TMS 320 C40 floating point DSP processor, interfaced to a Matlab front end. An IIR filter model was used with 20 feedforward and 20 feedback taps. The error path was characterised with white noise band-limited from 0 to 1 kHz. The hardware included 12 bit D/A and A/D and a software controllable multi-channel antialiasing low-pass filter module.

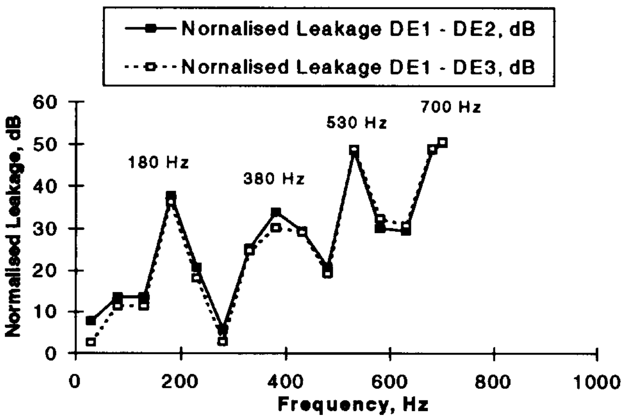
PROPAGATING AND NON-PROPAGATING MODES

Before discussing the experiments and the results, the following questions had to be addressed. First, what type of modes were likely to be excited during the experiments and second, the effect of non-propagating modes on the error hydrophone.

Since the pressure generating surfaces of the noise and control source were axisymmetric, only radial axisymmetric modes were expected to be generated. The ring frequency for both the steel and PVC flexible pipe [13] were well outside the 0 to 1000 Hz measurement range even with waterloading taken into account [14]. However some coupling to asymmetric wall modes may have occurred through curved terminating pipe sections. For the steel pipe, the first waterloaded radial-circumferential flexural mode was estimated at 1100 Hz, while similar estimates for the flexible PVC pipe yielded 112, 370, 764 and 1316 Hz for the first four modes. Clearly, the radial-circumferential flexural modes in the steel pipe would not have been excited within the frequency range of our measurements, though the first three in the flexible PVC pipe have had the potential to be excited. However, compared to steel, these modes were highly damped and may not have therefore contributed significantly.



(a)



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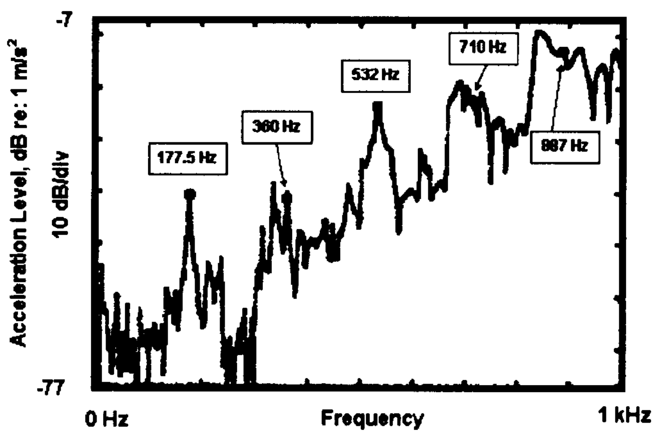

(c)

Figure 2: Results from first series of experiments (PVC terminating pipe). Frame (a) shows the reduction DE1 obtained at the error hydrophone H1, reduction DE2 from H2 at 1.00 m from open end, and reduction DE3 from H3 at 3.00 m from open end. Frame (b) shows noise leakage as DE1-DE2 and DE1-DE3 at H2 and H3 respectively. Frame (c) shows the axial accelerometer output at signal generator output level of 1 V.

The effect of non-propagating components generated by the fluid-wave control source on the error hydrophone H1 was estimated from modal calculations for fluid-acoustic waves in a rigid-walled cylindrical wave guide. The attenuation factors obtained for the first six higher order, axisymmetrically excited modes are 210, 380, 550, 720, 890 and 1060 dB, respectively. With such high attenuation levels, the non-propagating modes are not expected to degrade the error sensor performance.

EXPERIMENTS AND RESULTS

As difficulties were expected in the control of the combined fluid and wall axial wave fields, the first series of experiment was conducted with a flexible composite PVC terminating pipe [10]. The pipe is known to be a relatively poor conductor of wall-axial waves and previous measurements. indicated no presence of wall-axial modes at 70 dB below the fluid-acoustic wave level. Under such circumstances, it was expected that mainly fluid waves would propagate down the PVC terminating pipe and hence a considerable authority could be exerted by the control source.

The results of the first series of experiments are portrayed in Figure 2. Figure 2(a) shows the acoustic reductions at the error hydrophone H1 and the two monitoring hydrophones H2 and H3 as a function of frequency. The frequency steps were chosen so as not to coincide with 50 Hz mains frequency and the higher harmonics. Since H2 and H3 were inserted through the open end of the terminating pipe, their positions are referred relative to the open end, and these were 1.00 m and 3.00 m respectively. The H1 position relative to the open end of the PVC pipe was 3.74 m.

Figure 2(b) shows the acoustic leakage at H2 and H3, normalised with respect to the reduction at the error sensor H1. It is interesting to note that the leakage is nearly the same at both H2 and H3 positions and that the peaks coincide with peaks in the wall-axial acceleration spectrum shown in Figure 2(c). This suggests that the wall-axial noise component is not being controlled by the fluid-wave control source and significant noise energy is passed down the duct when wall axial waves are strongly excited. The highlighted frequencies in the wall-axial acceleration peaks are harmonically related to the fundamental component at 177.5 Hz.

The results of the second series of experiments are shown in Figure 3, where the flexible PVC terminating pipe was replaced by a steel one. The graphs are set out in the same manner as in Figure 2., where H2 and H3 positions were measured at 1.00 m and 3.00 m from the open end of the steel pipe,

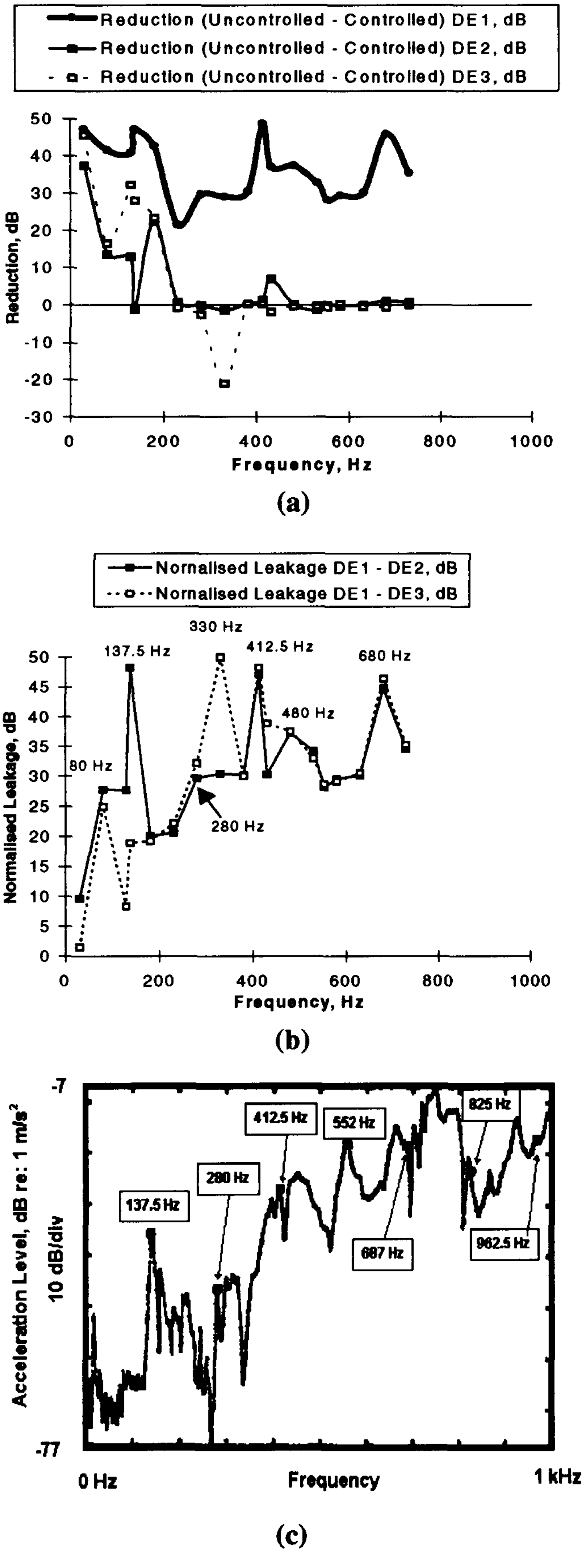

Figure 3: Results from second series of experiments (steel terminating pipe). Frame (a) shows the reduction DE1 obtained at the error hydrophone H1, reduction DE2 from H2 at 1.00 m from open end, and reduction DE3 from H3 at 3.00 m from open end. Frame (b) shows noise leakage as DE1-DE2 and DE-DE3 at H2 and H3, respectively. Frame (c) shows the axial accelerometer output at signal generator output level of 1 V.

respectively, and where the H1 error sensor position was 3.53 m from the open end. In contrast with the results in Figure 2, the acoustic leakage portrayed in Figure 3(b) is very large from about 200 Hz onward, showing practically no attenuation, and at 300 Hz, the sensor H3 displays large negative attenuation during the controlled state.

A qualitative check of the radial wall vibration levels downstream from the error sensor revealed little or no reduction between the controlled and uncontrolled condition, suggesting that a significant portion of the noise energy continued to be transmitted past the error sensor due to a strong presence of the wall-axial vibration component.

The leakage peaks in Figure 3(b) indicate some coincidence with the peaks in the acceleration spectrum in Figure 3(c), but the correlation is no longer as clear as in the case of the PVC terminating pipe.

It is only at the very low frequencies (<50 Hz) where wall-axial waves could not be easily excited, that the fluid-wave control source was reasonably effective in cancelling the downstream noise field.

CONCLUSION

Above experiments have highlighted the importance of the need to control the wall-axial wave components during active control of noise propagation in water-filled steel pipes. Some amount of control was achieved with a fluid-wave control source, but only at very low frequencies at which wall-axial waves could not be easily excited. From about 100 Hz onwards, however, little reduction was achieved, and at times, active control led to an increase rather than decrease in the downstream noise level. Such behaviour indicates that the noise contribution from the wall-axial components is of a similar order of magnitude as the fluid-wave one and therefore a specific wall-axial wave control source is required to supplement the fluid-wave control source.

It is quite likely that the blind flange used in the noise source segment and any curved sections in the terminating pipe provided additional coupling between the fluid and wall axial waves and this may have accentuated the problem. However, in a practical applications, this type of coupling is to be expected and makes therefore the requirement for wall-axial wave control more pertinent.

While other investigators have proposed various methods for the control of the wall-borne noise components, the one envisaged by the author consists of PZT patches mounted axially on the external wall of a pipe section placed in series with the fluid-wave control source. This arrangement caters specifically for the control of zero order wall-axial and fluid waves and is expected to provide a reasonably independent control of the two wave regimes, while maintaining insensitivity to the variation in coupling efficiency between the fluid-borne and structure-borne modes.

Future work will involve test and evaluation of combined fluid-structural control sources under various boundary conditions and the investigation of active control of 'hybrid' fluid-structural modes [8].

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