MECHANICAL DESIGN OF FLOW CONTROL OF THE EXHAUST NOISE FROM A V6 PETROL ENGINE

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Abstract

The concept of reducing petrol engine exhaust noise by regulating exhaust gas flow has been previously investigated theoretically and demonstrated. One method by which this can be achieved is by the optimal regulation of the gas flow by accurately controlling an oscillating valve. This paper describes design improvements of a standard butterfly valve configuration by utilising a linear electromagnetic shaker and a rocker arm, to regulate the flow of exhaust gases from a petrol engine. By redesigning the mechanism and increasing the stiffness of the system, so that its natural frequency was close to the cylinder firing frequency, permitted the valve to oscillate at high amplitudes using minimal force applied by the actuator. Experimental results of this flow control valve design showed that the noise at the engine firing frequency was attenuated by over 10dB.

1. Introduction

The acoustic spectrum of the exhaust noise from a reciprocating engine is dominant at the cylinder firing frequency and harmonics and is given by

$$f_{\text{cylinder}} = \frac{\text{RPM}}{60} \times \frac{\text{cylinders}}{2} \tag{1}$$

where RPM is the rotational speed of the engine in revolutions per minute, and cylinders is the number of cylinders in the engine. Although passive dissipative mufflers are commonly used on exhaust systems, Caillé [1] comments that it is difficult to reduce low frequency noise using passive silencers. One alternative is to control the exhaust gas flow fluctuations in real time which can reduce the exhaust noise. Hardouin et al. [2] showed that the movement of an oscillating valve synchronised with the fluctuations in the gas flow reduced the amplitude of pressure pulsations, and hence reduced the acoustic noise. Hardouin demonstrated a system using a butterfly valve to regulate downstream pressure utilising an adaptive feedback controller. Carme et al. [3, 4] demonstrated this concept on an 11 litre, 6 cylinder diesel engine, using an oscillating butterfly valve system and a feed-forward LMS algorithm controller. However, this system relied heavily on the exhaust gas flow speed and any change in the exhaust flow caused issues with the stability of the controller. Boonen and Sas [5, 6] were able to regulate the gas flow from an engine using an electro-dynamically driven globe valve. Boonen and Sas [5] were able to reduce exhaust noise by 13 dB with only 10 kPa back pressure on the engine.

This paper describes a valve design that can operate and control engine gas flow at the cylinder firing frequencies. Results are reported in terms of valve displacement and sound pressure levels downstream of the active valve.

2. Design

The active exhaust valve is placed downstream from a buffer volume, as shown in Figure 1. When the butterfly valve closes, it obstructs the flow of exhaust gas and causes an increase in the back pressure, which is stored in the buffer volume. The buffer volume is essentially an expansion chamber silencer that is often used on larger engines. The function of the active exhaust valve is to reduce the peak-to-peak amplitude of the pressure pulsations in the exhaust gas stream, as shown schematically in Figure 2 by regulating the mass flow rate to a constant flow rate. The amount by which the back pressure is increased is determined by the size of the of the buffer volume – the larger the buffer volume, the lower the increase in back-pressure.

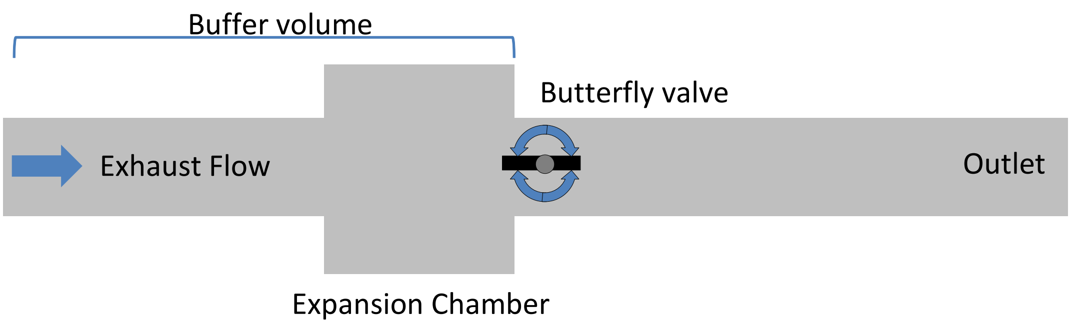

Figure 1. Configuration of the active exhaust valve placed downstream of a buffer volume

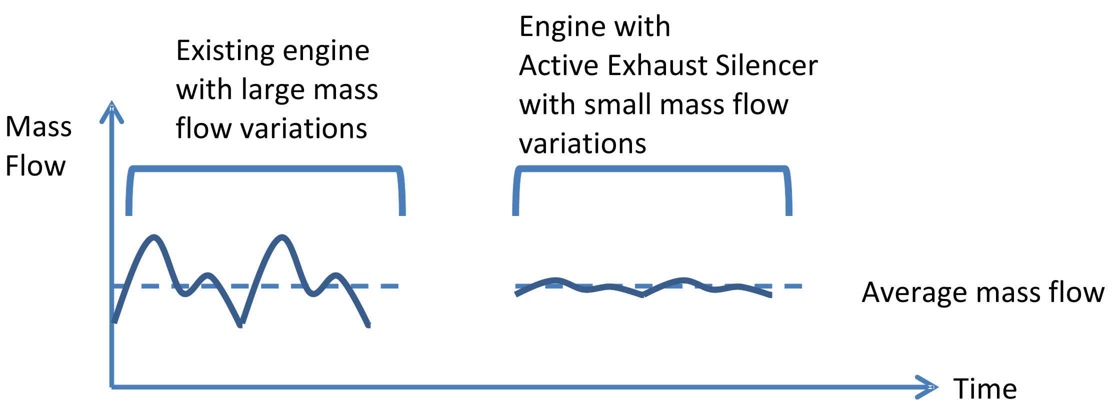

Figure 2. Reducing the amplitude of the mass flow in exhaust with the addition of the active exhaust silencer

By adjusting the amplitude and phase (timing) of the angle of the butterfly valve to coincide with the arrival of pressure pulses in the exhaust gas stream, it is possible to attenuate these pressure pulsations, thereby reducing the noise that is radiated from the exhaust.

The active exhaust silencer that was developed comprises an oscillating butterfly valve inserted into the exhaust gas flow as shown in Figure 3. A linear electro-magnetic shaker (LDS V203) is used to oscillate a crank mechanism, which in turn rotates a shaft on which the butterfly valve is mounted.

Three prototype designs were evaluated under representative operational conditions. Figure 4 shows a photograph of a prototype active exhaust valve, where a laser sensor was used to measure the displacement of the valve during excitation from the shaker.

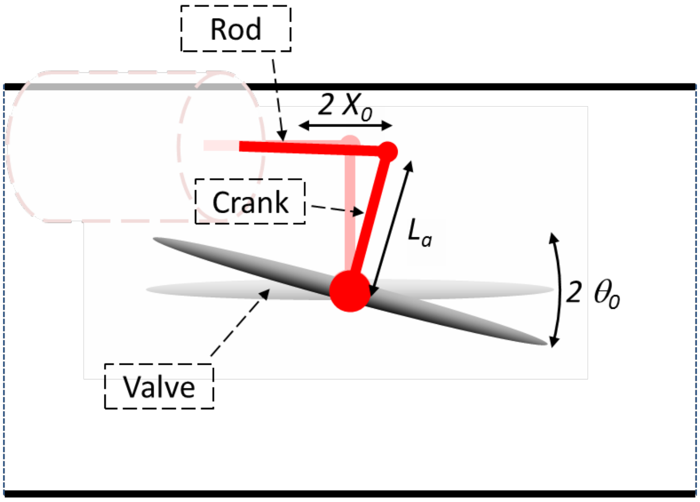

Figure 3. Schematic of the actuator and crank mechanism used to oscillate the butterfly valve

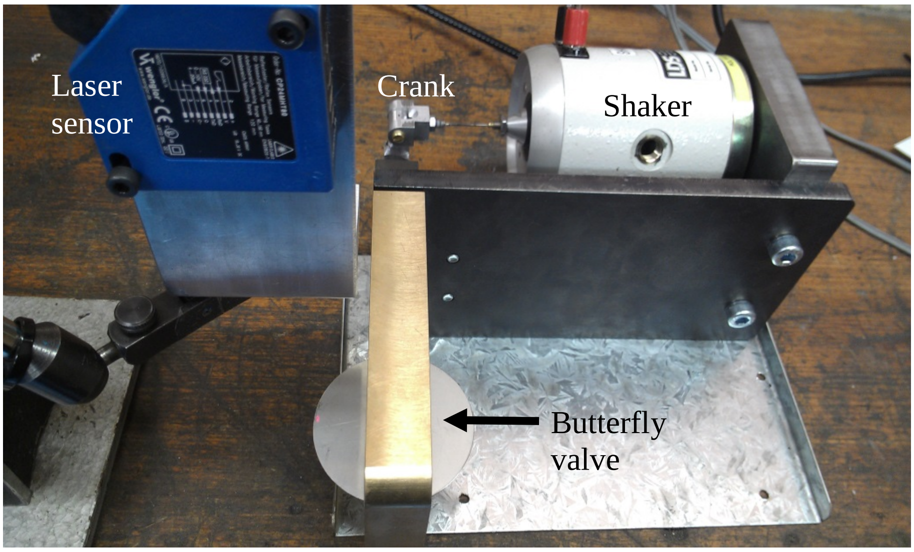

Figure 4. Experimental setup of the butterfly valve rig with linear actuation system and laser displacement sensor used for development testing

3. Mechanical Design

3.1. Design requirements

A main portion of the work involves the mechanical design of the active exhaust silencer that provides the acoustic attenuation of tonal exhaust noise. The active silencer has a number of requirements to fulfil:

- Acoustic attenuation of tonal noise for varying engine speed and load.
- The ability to withstand exposure high temperature engine exhaust gases.
- Prevent leakage of exhaust gases.

Three successive designs were developed as part of a design and test programme. Their main features are as follows:

Design 1 - the initial concept design, comprised of the following specifications

- Mild steel butterfly body, 62mm diameter butterfly valve, 1.0mm stainless steel
- Steel body to support drive shaft
- Crank Arm (L_a) 30mm

Design 2 was modified to operate under thermal stress, mainly by matching the material coefficients of expansion. Its main features were:

- Brass butterfly body, 62mm diameter butterfly valve, 1.0mm stainless steel
- Brass body to support drive shaft
- Crank Arm (L_a) 30mm
- Brass bush to support shaft near crank arm

Design 3 was created to maximise valve displacement over the proposed frequency range of operation, by reducing friction and tuning torsional stiffness to adjust the system's resonance. It comprised the following specifications

- Stainless steel butterfly body, 59mm diameter butterfly valve, 1.0mm stainless steel
- External bearing carriers thermally insulated from butterfly body
- Ceramic hybrid bearings
- Crank Arm (L_a) 10mm
- Adjustable torsional springs to increase resonance frequency of system

4. Tests and Results

A brief description of the evaluation tests carried out on the prototypes is provided in this section, along with key results.

4.1. Thermal expansion tests

Tests were conducted to ensure that the thermal expansion of the active valve assembly when exposed to high temperature exhaust gas would not cause the moving parts to seize. The test involved installing the valve into the engine exhaust duct and the engine was operated under load until the temperature of the valve block reached 350^{o}C. The valve was rotated manually to check whether it had seized. The valve block was then allowed to cool to room temperature and was then manually rotated to ensure that the effects the heating and cooling cycle did not adversely affect the valve system. Design 1 failed the tests, as differences in coefficients of thermal expansion caused the shaft to seize at high temperatures, or after cooling. After a test with brass bush inserts in Design 1, it was decided to rebuild the entire valve body in brass. Brass material was selected as it has a coefficient of thermal expansion that is greater than stainless steel, which is used for the shaft, and it also has a low coefficient of friction. After these changes were made the device (Design 2) passed the thermal test. The design was further improved (Design 3) by using ceramic bearings in bearing housings that were thermally insulated from the valve body. The use of these high temperature bearings reduced the friction on the shaft compared with the brass bush bearings.

4.2. Valve displacement tests

Tests were conducted on a bench setup, where each valve was clamped into position and a Wenglor CP24MHT80 distance photoelectric sensor was pointed at the edge of the valve as seen in Figure 4. The driving signal voltage and power amplifier gain were constant; the frequency was varied between 50Hz and 150Hz. The valve angular displacement (ϑ_o) was calculated from the displacement measured by the laser sensor.

The results from tests on Design 2 had strong harmonic distortions that were attributed to shaft bending, which was reduced with additional support points on the shaft close to the rocker arm. The displacement of the valve at various frequencies is shown in Figure 5. The response of the system in Figure 5 is characteristic of a single degree of freedom (SDOF) mass-spring system in the inertial range, above the resonance frequency. It was estimated that the resonance frequency was approximately 45 Hz. The inertia of the system comprises the moving mass of the shaker armature, stinger rod coupling between the shaker and the crank arm, the rotating shaft and butterfly valve. Stiffness is only provided by the shaker's suspension. In order to increase the valve displacement over the frequency range of interest of 100-120 Hz, additional stiffness was introduced in the system by means of a torsional spring attached to the valve shaft. The torsional spring was a piece of spring steel wire that was clamped at each end using small drill chucks and was twisted along its axis. The addition of the torsional spring and ceramic bearings led to Design 3. The dynamic response of Design 3 was tested and it was possible to tune the resonance of the system by adjusting the clamped of the spring wire, which adjusts the torsional stiffness. Figure 6 shows the response of the system when its resonance frequency was tuned to 90 Hz.

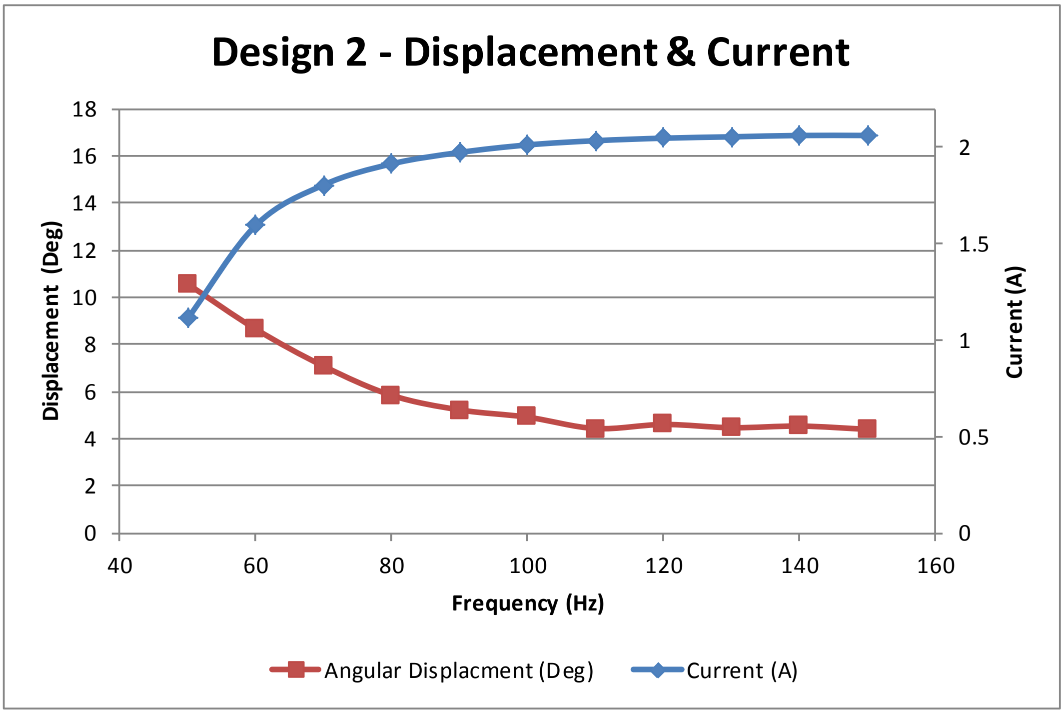

Figure 5. Design 2: Displacement and current draw as a function of frequency for a fixed excitation voltage

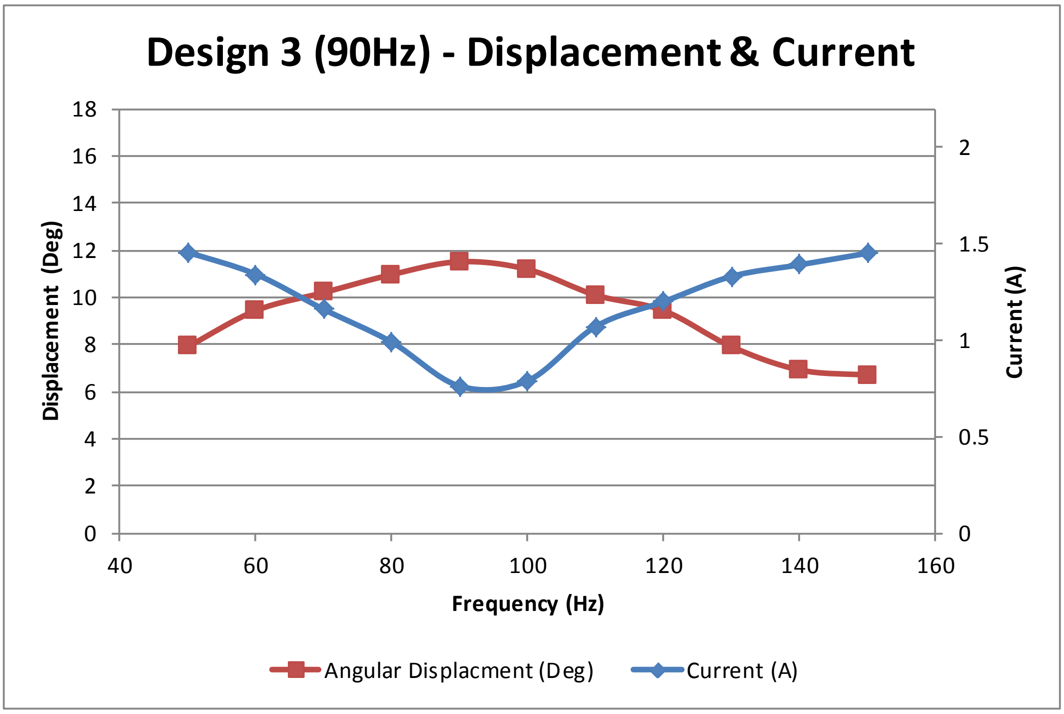

Figure 6. Design 3, tuned to 90 Hz: Displacement and current drawn as a function of frequency for a fixed excitation voltage

Using Equation (1), an engine operating condition of 2000RPM corresponds to a cylinder firing frequency of 100Hz for a 6-cylinder engine. At a frequency of 100Hz, Figure 5 shows that the current drawn by Design 2 is approximately 2 amps, whereas for Design 3 the current drawn, shown in Figure 6, is 0.75 amps. Further, comparison of the angular displacement at 100Hz for Designs 2 and 3shown in Figure 5 and Figure 6, is 5° and 11°, respectively. Hence, Design 3 that was tuned to a resonance frequency of 90 Hz, has twice the angular displacement using half the amount of current compared with Design 2. The valve in Design 3 can operate with a greater angular displacement using less current compared with Design 2. The advantage of using a valve that can oscillate with a large angular displacement means that it can exert high back pressure, without needing to set the zero position of the valve offset from horizontal.

4.3. Engine tests

Tests were conducted in the engine test cell at the School of Mechanical Engineering, The University of Adelaide on a 3.5 litre, V6 Mitsubishi 6G74M petrol engine that was loaded by a Schenck eddy current dynamometer with a Dyne Systems controller, as shown in Figure 7. This engine type was used in Mitsubishi Magna passenger vehicles. The catalytic converter was removed to increase the space available between the engine and the vertical exhaust pipe, and an active exhaust silencer was installed. The tachometer signal was generated by means of a magnetic pick up sensing the passing of four magnetic bolt heads screwed into the dynamometer shaft end plate.

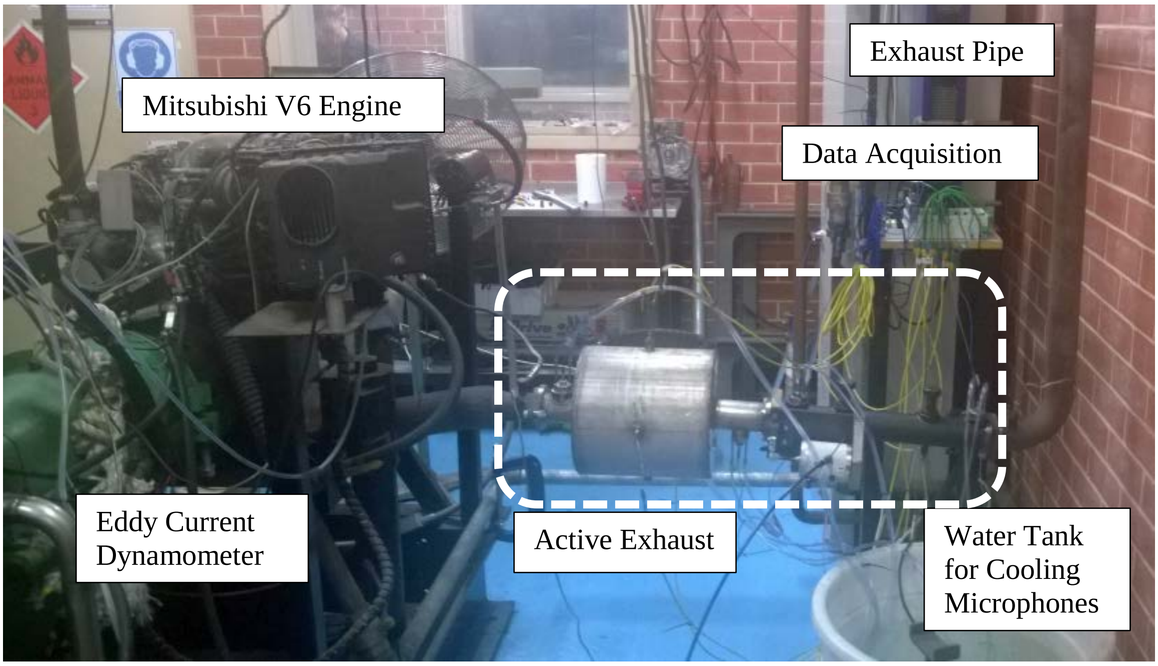

Figure 7. Engine testing setup

The results obtained with Design 3, tuned to the resonance frequency of 90 Hz, are presented in this section for various mean valve angles, where 0° is fully open in the horizontal position and 90° is fully closed. The reason why only Design 3 was tested on the engine was because this prototype had the greatest range of motion of the valve. When the valve is installed in the exhaust, the valve angle at rest is set to a particular value so that the valve obstructs the flow and causes a slight increase in the back pressure on the engine. By using a design that has a large range of the oscillating angle, permits the angle at rest to be small, and therefore reduces the flow obstruction. However, due to the limited angular displacements achieved with the valve so far, it was necessary to set the valve angle at rest greater 0° in order to conduct active noise control to reduce the tonal noise. Three valve angles at rest were chosen for testing: 46°, 56°, and 66°. The initial testing was conducted at an angle of 46°, then was increased by 10°, which is less than the maximum angular displacement of the valve of 12° as shown in Figure 6. The results presented in Table 2 show that at the 46° valve angle, no noise reduction achieved as the valve didn't restrict the mass flow rate sufficiently. The valve was tested at angles of 56° and 66°, at two engine speeds, 2000 rpm and 2125 rpm, with a constant load of 150 Nm. After conducting the test where the angle was 66° it was decided not to conduct further tests with greater valve angles as there was little additional benefit in noise reduction, but the exhaust back pressure increased significantly.

Time averaged engine back pressures were found to not vary significantly when active control was tuned on. Their values measured in these four configurations are reported in Table 1. A block FFT LMS [7] algorithm was implemented on a dSPACE ds1104 platform for these tests. The parameters of the active noise controller were set such that the initial convergence coefficient was adjusted at regular intervals to reach the target attenuation value of 10dB at the cylinder firing frequency. The engine was operated in the set configuration until exhaust gas temperatures stabilised. The controller was then turned on approximately 30s after the start of the recording. Results obtained in the four configurations of interest are reported in Figure 8 to Figure 11, in the form of a time history of the tonal sound pressure level at the firing frequency in the left graph, and control voltage generated by the controller and fed to the power amplifier in the right graph, the latter being representative of the control effort.

Table 1. Engine back pressure measured in the test configurations

Valve Angle	**2000 rpm**	**2125 rpm**
56°	6.5 kPa	7.7 kPa
66°	14.0 kPa	17.2 kPa

These results show that increasing the valve angle or the engine regime increases control authority, causing the controller to converge faster, and to achieve the noise reduction target with a reduced control effort. This is illustrated in Figure 11, which shows the stabilised noise level reductions and corresponding control voltage and current in rms values. Table 2 shows that for the angle of 46°, the maximum driving current of 2.8A was exceeded before any significant noise reduction could be measured, indicating that the system was unable to achieve sufficient angular displacement.

Table 2. Controller performance for the four test configurations

		Noise Reduction (dB)	**Back Pressure (Pa)**	**Control Voltage (V)**	**Max Current Draw (A)**
Angle 66 Deg	2000 RPM	12.3	13994	0.19	0.84
	2125 RPM	14.1	17185	0.15	0.61
Angle 56 Deg	2000 RPM	12.9	6473	0.37	1.50
	2125 RPM	13.1	7693	0.32	1.30
Angle 46 Deg	2000 RPM	0	< 3000	-	> 2.8
	2125 RPM	0	< 3000	-	> 2.8

5. Conclusions

This paper presents the steps taken to design and demonstrate an oscillating butterfly valve actuator for active exhaust noise control on internal combustion engines. Careful design was required to generate the torque required for, and overcome the dynamic loads resulting from, the oscillatory motion of the automotive size butterfly valve at the frequencies required for the proposed application. Numerous solutions were envisaged, and the most advanced to date consists of an inertial shaker driving a rocker arm attached to the valve shaft.

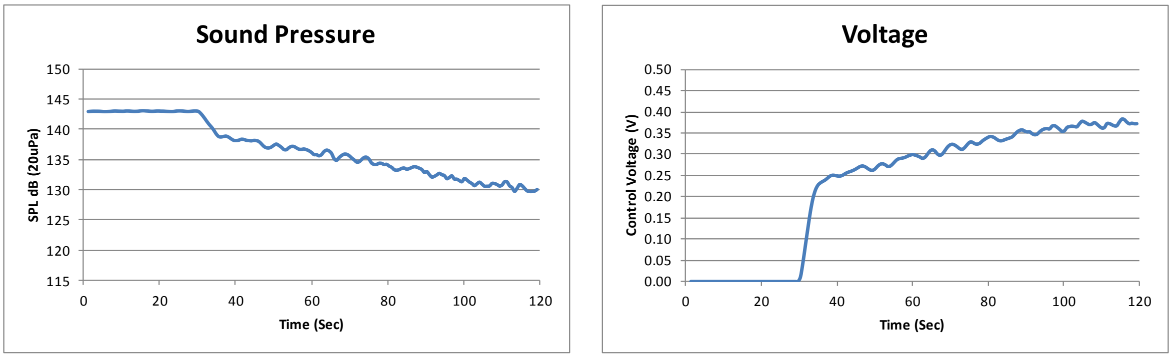

Figure 8. Valve set to 56 degrees and engine run at 2000 RPM at 150 Nm:
control starts at approximately 30s; left graph: tone SPL; right graph: control rms voltage

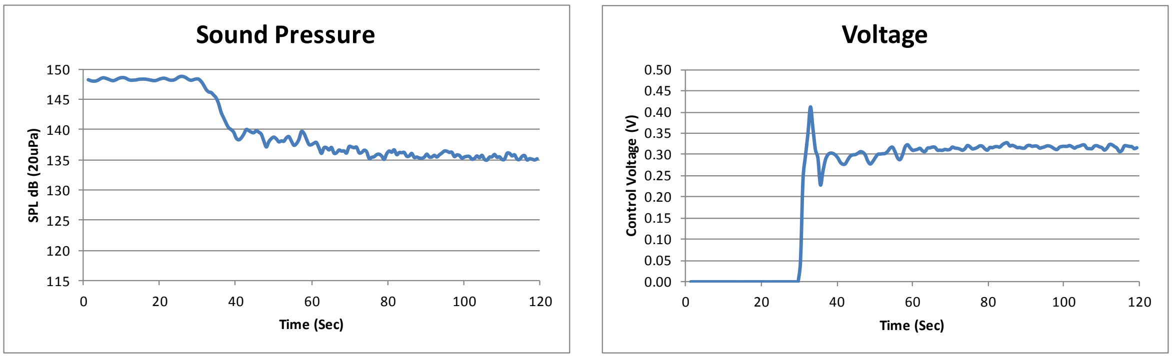

Figure 9. Valve set to 56 degrees and engine run at 2125 RPM at 150 Nm
control starts at approximately 30s; left graph: tone SPL; right graph: control rms voltage

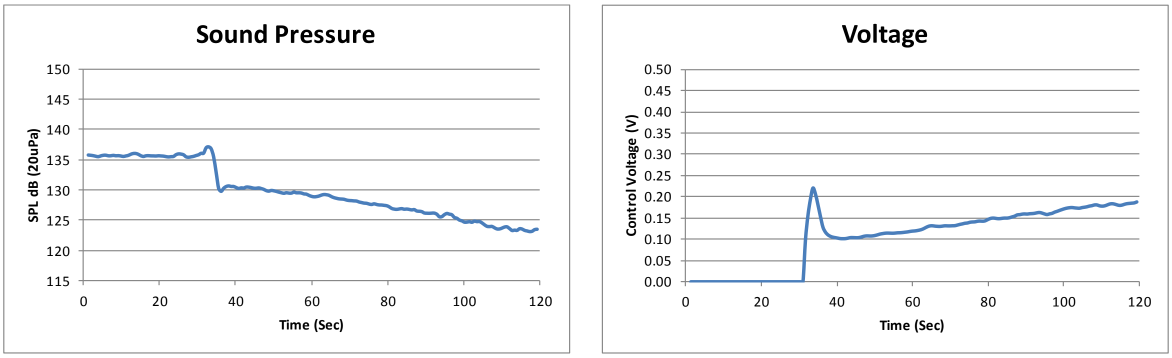

Figure 10. Valve set to 66 degrees and engine run at 2000 RPM at 150 Nm
control starts at approximately 30s; left graph: tone SPL; right graph: control rms voltage.

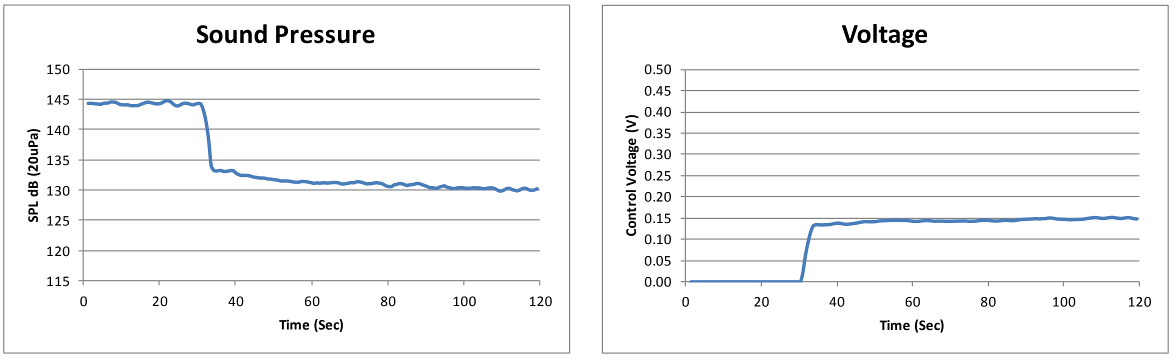

Figure 11. Valve set to 66 degrees and engine run at 2125 RPM at 150 Nm
control starts at approximately 30s; left graph: tone SPL; right graph: control rms voltage

One of the most significant design risks was on the survivability and durability of the system in the hot corrosive environment of an internal combustion engine exhaust system. This risk was managed by choosing materials with matching coefficients of thermal expansion, and eventually using ceramic bearings and stainless steel housing.

Another source of difficulty lies in the requirement that the translation motion of the shaker be converted into valve rotation, at relatively high frequencies, where the system needs to be stiff. This causes significant stress and accelerated fatigue, eventually leading to material failure. The mechanical performance of the system was also limited by inertial loads and the limitation of the electrodynamic shaker, and it was found that the system's performance could be greatly enhanced by introducing additional stiffness, in the form of a torsional spring mounted on the valve shaft.

The results presented here show that the mechanical design changes to the initial system have greatly improved the overall system performance, and it was successfully demonstrated in a practical application on an automotive engine.

As shown in the results, it is possible to reduce the fundamental noise of an exhaust by more than 10 dB with an oscillating butterfly valve design as seen in Figure 8 to Figure 11. The results show that there is a direct relationship between the angle of the valve and the back pressure and the control voltage required to obtain control. More work is required to determine the optimum valve angle required for a particular operating speed range.

Acknowledgments

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