ACTIVE FLOW CONTROL OF THE EXHAUST NOISE FROM INTERNAL COMBUSTION PISTON ENGINE

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ABSTRACT

The concept of reducing piston engine exhaust noise by controlling the flow through the exhaust line has been investigated theoretically and demonstrated experimentally in previous work. While this work demonstrated a successful application to a medium size diesel engine in fixed operational conditions, the issue of back pressure was not addressed. Back pressure can be mitigated by the interposition of a buffer volume, the size of which is a determining factor in system performance, particularly when it is designed to operate over a broad range of engine speed and loading. This paper presents a simplified model of an active flow control exhaust system, which is then applied using a 16 litres, 8 cylinder diesel engine, in order to estimate back pressure, control valve loading and dimension its key components in such a way that back-pressure is maintained below a specified threshold over a broad range of speed and loading.

Keywords: Engines, Active noise control

1. INTRODUCTION

Hardouin et al. (1) developed a control system using a butterfly valve and a bend-limited frequency domain adaptive feedback controller, which they demonstrated on a pulsed low experimental set-up, using downstream pressure as the error signal. Subsequently, Carme and his co-workers (2) (3) also implemented a butterfly valve system, this time driven by a filtered reference LMS algorithm. They demonstrated good success at a single speed on a piston pump and a 11 litres, 6 cylinders diesel engine, but pointed out that the secondary path was strongly dependent on the exhaust gas flow speed, and suggested the implementation of a lookup table to account for speed variations. Boonen and Sas (4) developed an electrodynamically driven globe valve to regulate the flow out of an engine cylinder where the piston is kept at bottom dead centre, and air flow is forced through the valves, which are operated by a motor driven cam mechanism. They obtained good results at low frequencies by implementing a feedback strategy, using a controller with 100 Hz bandwidth. Using a simple electrical analogy, they determine that the required flow resistance can take any positive value, and the engine back pressure can be reduced by increasing the volume between the engine and the control valve. In most realistic situations, however, the back pressure will be minimised by using the largest possible buffer volume within the spatial constraints of the application. Maintaining the back pressure at the minimum positive value is of great importance for practical applications, but was not discussed in the literature reviewed above. The purpose of this paper is to determine the basics of the control process, and to produce an estimate of the range of back pressures that can be expected from a realistic situation, using a medium size diesel engine typically used for road transport or power generation.

After a brief description of the physical layout of the control system and the presentation of its simplified governing equations, the mass flow rate of an example diesel engine is synthesised from its published performance data, and subsequently used in time dependent simulations of the active flow control process applied to that engine, using a somewhat idealised sensing strategy. Finally, results are reported in terms of sound pressure level, back pressure and valve displacement.

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2. MODEL OF THE SYSTEM

The following text describes the development of a mathematical model that is used to

- Develop an approximate model of the mass flow rate produced by the engine.
- Develop an approach that can be used to control the mass flow fluctuations at the exhaust outlet, and hence will reduce the exhaust noise.
- Predict the change in back-pressure on the engine.
- Predict the noise reduction by the active noise control system, in relation to its cost in terms of back pressure.

Figure 1 shows a schematic of the active exhaust silencer. A fluctuating mass flow $q(t)$ enters the silencer (from the left in Figure 1). There are pressure losses along the exhaust duct between the outlet of the engine manifold and the active exhaust silencer. The buffer volume upstream of the control valve has a volume V_s. At the outlet of the buffer volume is an oscillating butterfly valve that is used to regulate the mass flow rate out of the system. Downstream of the active butterfly valve is another section of silencer and the exhaust pipe that also has friction losses associated with it.

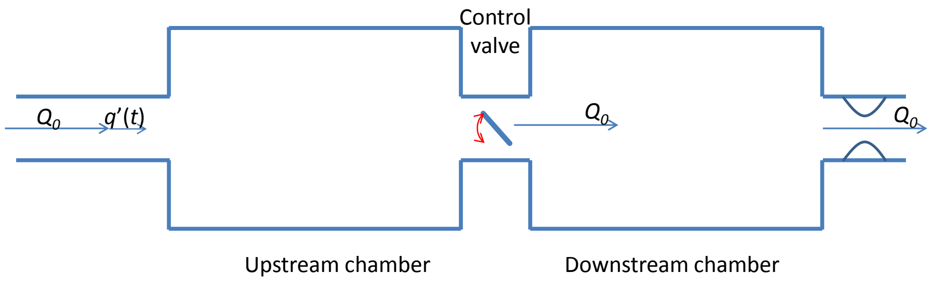

Figure 1: Schematic of the active exhaust silencer.

2.1 FLOW CONTROL

The mass flow rate is expressed as the sum of its mean value Q_0 and its fluctuation $q'(t)$:

$$q(t) = Q_0 + q'(t). \tag{1}$$

At steady state, and neglecting the effect of temperature fluctuations, the law of conservation of mass dictates that the constant flow rate achieved by the flow regulator be Q_0.

The mass of air stored in the buffer volume V_s is time dependent. The volume V_s is fixed, and the pressure and density within it vary. At a minimum, the air pressure and density in this upstream volume must be such that the required continuous flow rate Q_0 is maintained out of the fully open butterfly valve, through the flow resistance offered by the exhaust line downstream of the valve. This loss is modelled as a loss coefficient C_{d0}. in the following equation describing the minimum pressure P_{min} in the downstream chamber with reference to the atmospheric pressure P_{atm}.

$$P_{min} = P_{atm} + \frac{Q_0^2}{2\rho S^2} C_{d0} = P_{atm} + \frac{Q_0^2 V_s}{2M_{min} S^2} C_{d0}\,, \tag{2}$$

where S is the cross section area of the downstream exhaust pipe.

Using the perfect gas law, P_{min} can also be expressed as a function of the minimum mass of air M_{min} stored in the volume V_s at the temperature T_{min}

$$P_{min} = \frac{M_{min}}{m_{air}} \frac{R}{V_s} T_{min}\,, \tag{3}$$

where R=8.31 J.mol^{-1}.K^{-1} is the ideal gas constant and m_{air} is the molar mass of air. Substituting this expression in the above equation (2) leads to a second degree polynomial equation that can be solved for M_{min}.

The corresponding pressure inside the buffer volume can then be expressed as

$$P_{min} = \frac{1}{2}\left(P_{atm} + \sqrt{P_{atm}^2 + 2\frac{RT_{min}}{m_{air}}\frac{Q_0^2}{S^2}C_{d0}}\right). \tag{4}$$

The value of this minimum storage pressure increases with exhaust gas temperature, mean mass flow rate, and minimum pressure loss coefficient of the fully open control valve and downstream exhaust line.

At any instant t when the controller is operating, the mass of gas stored in V_s is:

$$M(t) = M_{min} + \int_{t_{min}}^{t} q'(\tau)d\tau\,, \tag{5}$$

where $M(t_{min})=M_{min}$. The variation in mass is determined by the variation of the incoming mass flow rate relative to its average.

The gas density inside the volume can be calculated. The instantaneous pressure and temperature variations at each instant are derived from the isentropic relations for an ideal gas.

$$P(t) = P_{min}\left(\frac{M(t)}{M_{min}}\right)^{\gamma}, \tag{6}$$

where $\gamma=C_p/C_v$ is the ratio of specific heats.

The assumption of an isentropic (adiabatic and reversible) process for the stored volume of gas is inaccurate, as heat exchange does occur, but will be used for this simplified model.

The above equations are now used to simulate the behaviour of an active flow control system.

2.2 ACTIVE CONTROL SYSTEM

A diesel engine is a device that generates a time varying mass flow rate, which is created by the successive exhaust gas pulses out of each cylinder. The aim of the proposed controller is to regulate the mass flow rate out of the exhaust pipe to a constant value, as shown in Figure 2, and hence minimise the fluctuating component of the mass flow, which constitutes the monopole source at the exhaust outlet.

It is necessary to store gas upstream of the butterfly valve in a buffer volume such that it acts as a capacitor between the fluctuating mass flow rate released by the engine and the steady mass flow rate to be discharged from the exhaust outlet. The exhaust system is modelled as shown in Figure 1, where a buffer volume is placed between a time-dependent source of mass flow and a butterfly (control) valve. In the interest of modelling the system as a series of discrete elements, the volume of air stored in the exhaust duct outside of the buffer volume is neglected, although in reality the volume of the upstream exhaust pipe between the buffer volume and the exhaust manifold contributes to the total buffer volume, with non-negligible friction losses occurring along the entire exhaust line. The flow losses through the system are assumed to be concentrated at two discrete points: upstream of the buffer volume and downstream of the valve. For the purpose of this preliminary analysis, all system dimensions are assumed to be much smaller than an acoustic wavelength in the frequency range of interest.

The magnitude of the pressure variation in the storage volume is determined by the gas temperature, and the size of the available volume V_s, which needs to be maximised in order to minimise pressure fluctuations upstream of the control valve and back pressure on the engine.

The additional back-pressure introduced by the control system is estimated by comparing the mean pressure upstream of the butterfly valve, with and without the butterfly valve.

Although the goal of the controller is to minimise the fluctuating component of the mass flow that is discharged from the exhaust, the algorithm is more complicated. One can imagine that if the butterfly were fully closed, there would be no mass flow discharged from the exhaust and although the mass flow would be minimised (zero), the engine would not function. Instead, the goal of the controller is to

- minimise the fluctuations in the mass flow rate downstream of the control valve, and
- minimise the build-up of pressure in the buffer volume.

Hence, the cost function for the controller cannot be designed exclusively around an error signal such as the pressure measured downstream of the control valve - it must also regulate the mean pressure in the buffer volume, particularly to prevent it from building up.

In order to gain an understanding of the back pressure introduced by the control system as well at valve displacements, the simplistic active controller shown in Figure 2 was also implemented in the simulations:

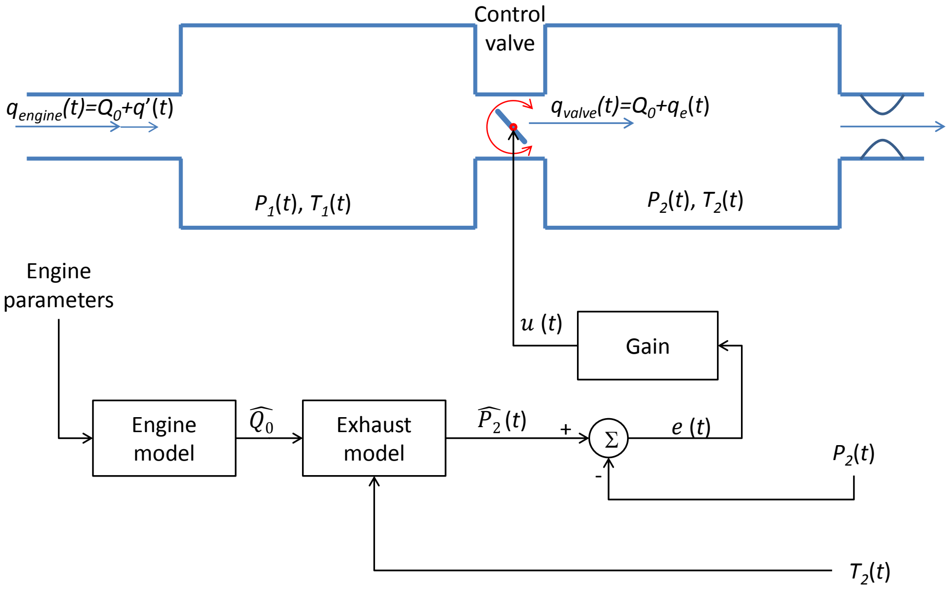

Figure 2: Active controller used in the simulations
to predict the exhaust back pressure and angle of the butterfly valve.

The controller includes a representative model of the plant that estimates the mean mass flow rate $\widehat{Q_0}$ based on measured engine parameters. Using this estimate and the gas temperature downstream of the control valve, a model of the exhaust line is used to estimate the mean pressure $\widehat{P_2}$ that would occur in the downstream chamber if mass flow rate fluctuations were totally cancelled.

Figure 2 illustrates a feedback architecture where the valve angle is adjusted in proportion to the difference $e(t)$ between this estimated value and the measured value of $P_2(t)$. It is recognised that the actual architecture of the controller will be different to that presented here, which is used in the present case to determine the typical buffer volume and backpressures that can be expected in a practical application.

3. APPLICATION TO A DIESEL ENGINE

The mathematical model described above was used to analyse the expected response of the engine that will be used as the demonstration test platform for this project - a Mercedes-Benz diesel engine model: OM 502 LA, 15.93 litres, 8 cylinders, 90° V, inter-cooled twin turbo. This engine will be used for the subsequent experimental programme.

Based on the engine specifications, the exhaust pulse of a single cylinder was modelled, and its shape was determined to include both the blow down and displacement phases of the exhaust as per the measurements reported by Tabaczynski et al. (1) on a single cylinder spark ignition engine. The exhaust pulse from each cylinder is composed of a short, sharp peak corresponding to the release of gas as the exhaust valves open shortly before the bottom dead centre, followed by a longer and weaker contribution as the piston pushes the remaining gas out of the piston, which stops when the exhaust valves close at the top dead centre (real engines have slightly different timings). In a four stroke engine, the time-dependent mass flow rate of exhaust gas produced by the engine is the sum of the exhaust pulses associated with each cylinder over two crankshaft rotations, with delays associated with the cylinder firing sequence and propagation down the manifold.

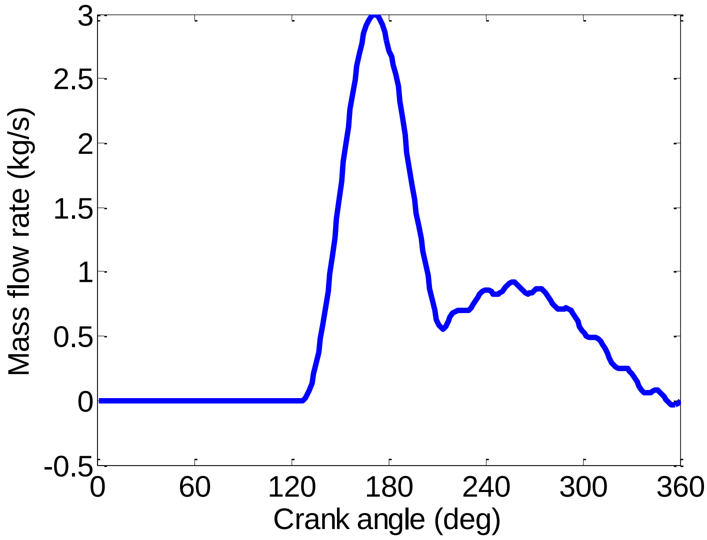

Figure 3: Mass flow rate from one piston versus crank angle at 1800 rpm.

The gas unsteady mass flow rate out of the engine is modelled as a regular succession of such pulses whose magnitude, frequency, duration and overlap are determined by engine and operational parameters. The shape and magnitude of this function determines the mass of gas that needs to be stored upstream of the regulating valve, which is calculated in the next section.

For purposes of illustration, simulations were carried out based on a butterfly valve commercially available from the SOMAS VSS series. It is understood that this may not be a suitable product for the application due to the potentially excessive inertia of the disk, but this choice was made to use representative valve characteristics in order to produce realistic estimates of the back pressure introduced by the proposed control system.

The results presented below are based on the following parameters:

- Upstream and downstream expansion chamber volumes of 50 litres, (the upstream expansion chamber is used as the buffer volume)
- Engine operating at rated power and 1800 rpm in standard atmospheric conditions (6),
- The butterfly valve performance data are those of the 150mm SOMAS VSS model (7).

All results are presented over a quarter of engine revolution. Using the exhaust pulse function shown in Figure 3, the mass flow rate produced by the engine is shown in Figure 4, its mean value at rated power is 0.68 kg/s.

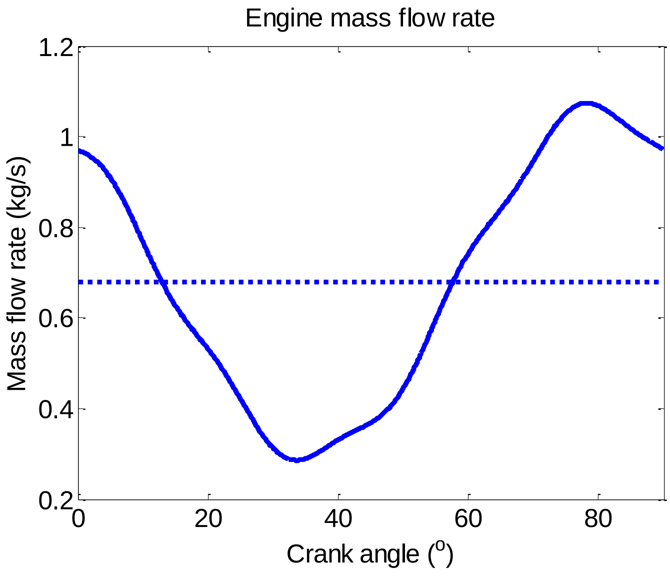

Figure 4 Mass flow rate of exhaust gas produced by the engine at rated power over a quarter of engine revolution. The straight line represents the corresponding mean value.

Based on this model of the exhaust gas mass flow rate, results from a simulation carried out over a period of 10s, where active flow control starts 1 second into the simulation, are shown below. The evolution of the sound pressure level on either side of the butterfly valve is shown in Figure 5, and the corresponding mean static pressures are plotted in Figure 6.

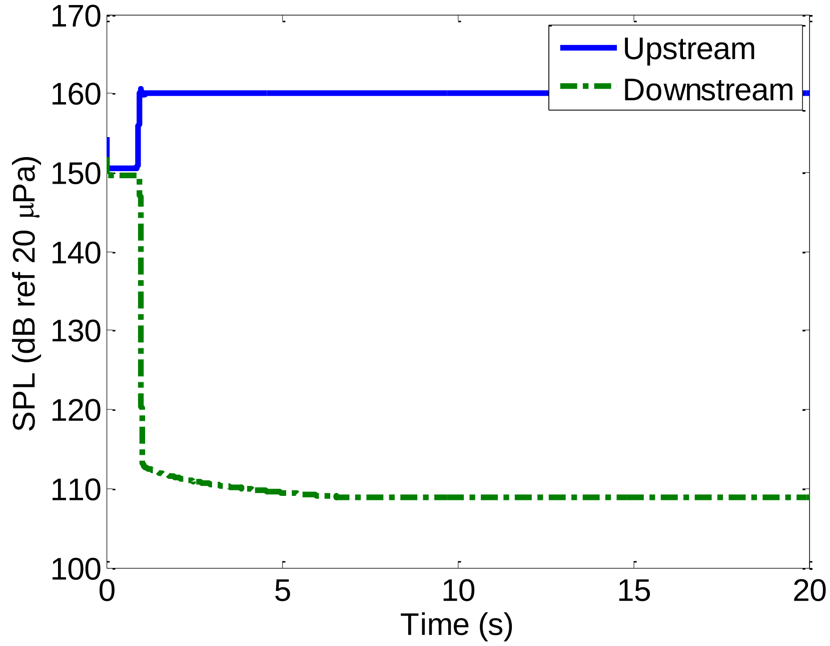

Figure 5 Time series of the sound pressure level inside the exhaust system on either side of the control valve.

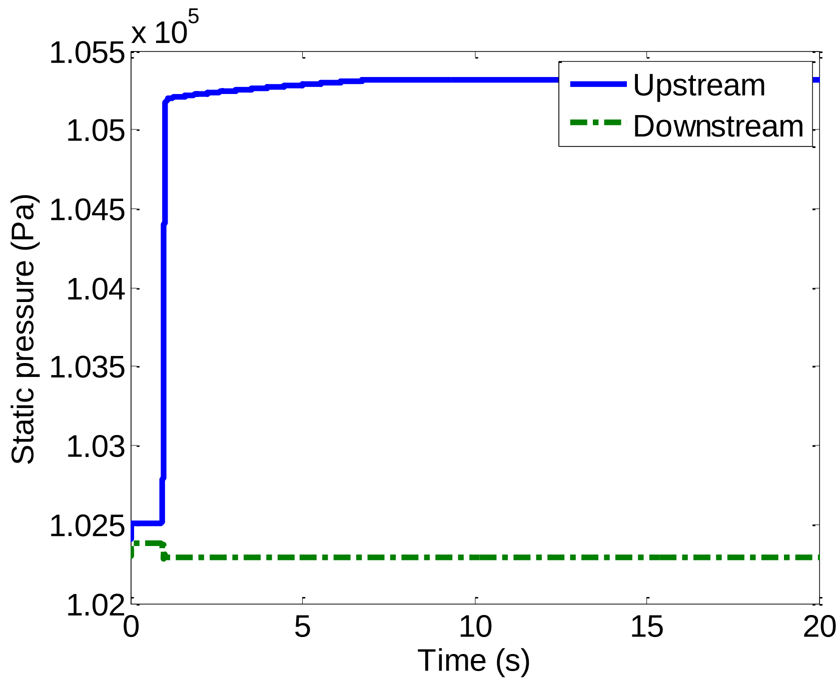

Figure 6 Static pressure inside the exhaust system on either side of the control valve.

When active control starts, the sound pressure downstream of the valve decreases by approximately 40 dB, while it increases by 10 dB in the upstream chamber, where the static pressure increases by slightly less than 3 kPa. The pressure drop prior to control is associated with the residual pressure drop introduced by the fully open valve. These results with a 50 litre buffer volume show that the simulated controller introduces an additional back pressure of approximately 3 kPa (0.03 atm) at rated power for this engine, as illustrated in Figure 7. As a first approximation, the back pressure is inversely proportional to the volume of the upstream exhaust chamber, and the back pressure can therefore be reduced by increasing this volume.

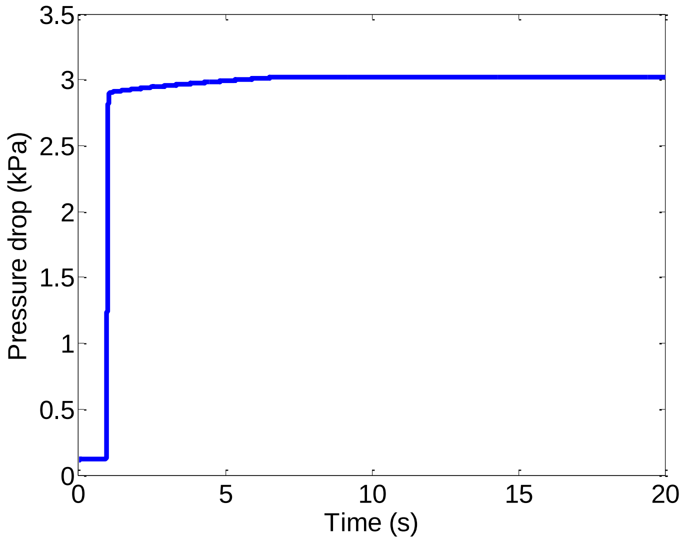

Figure 7 Static pressure drop across the control valve.

The time-varying pressure calculated with and without flow control in the upstream exhaust chamber is plotted in Figure 8, and compared with the pressure derived from the theoretical model presented previously.

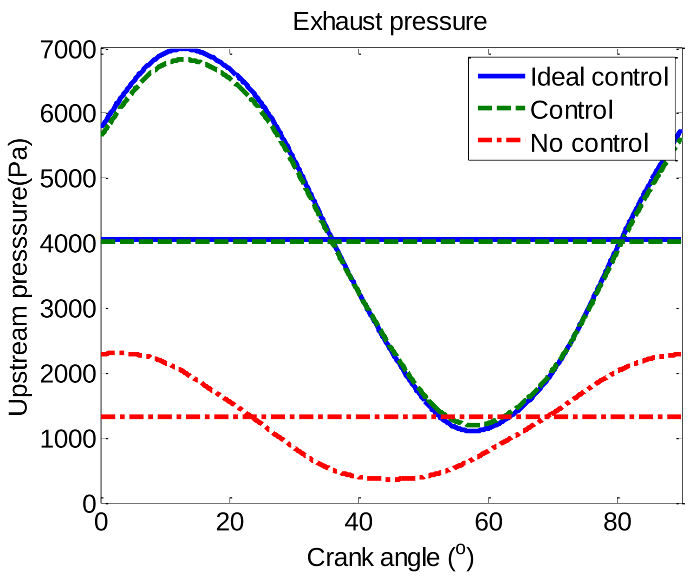

Figure 8 pressure calculated in the upstream exhaust chamber over a quarter of engine revolution.
— blue: ideal control; --- green: controller simulations; -·-· red: no control.
The straight lines represent the corresponding mean values.
The atmospheric pressure is subtracted from the results.

Figure 9 shows the pressure calculated in the downstream exhaust chamber, with and without control. In these simulations, the controller is quite successful at cancelling pressure fluctuations downstream of the valve, and maintaining a pressure required to evacuate the exhaust gases at a constant mass flow rate. It should be noted here that these results were obtained based on the idealistic assumption that the controller can maintain an accurate model of the mean mass flow rate and losses in the downstream section of the exhaust line. A practical application would certainly require an alternative sensing strategy to estimate the exhaust mean and downstream instantaneous flow rates, but the purpose of this paper is to estimate the additional back pressure induced by an idealised exhaust flow control system on the operation of the engine.

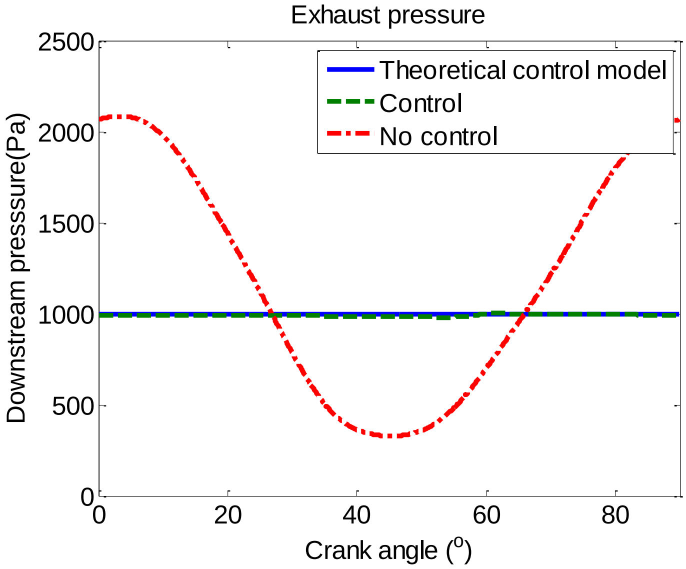

Figure 9 Pressure calculated in the downstream exhaust chamber
over a quarter of engine revolution.
— blue: ideal control; --- green: controller simulations; -·-· red: no control.
The atmospheric pressure is subtracted from the results.

The power spectral densities of the pressure fluctuation in the downstream exhaust chamber are shown in Figure 10. In this simulation, the fundamental tone and first few harmonics are strongly attenuated, while the higher harmonics are amplified. The harmonic content of the actual engine will be different from that of the simulations here, and the severity of harmonic amplification will be critical to the success of the experimental application. The main motivation for putting the control valve between the two chambers of the exhaust silencer is to passively attenuate any higher harmonics that may be amplified when active flow control is operating, along with aerodynamic noise from the valve.

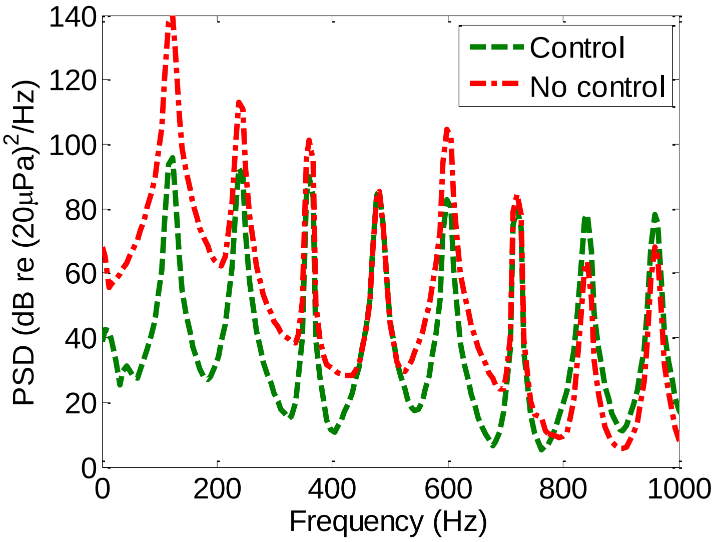

Figure 10 Power spectral densities of the pressure fluctuation in the downstream exhaust chamber.
--- Green: controller simulations; -·-· red: no control.

The angular displacement of the butterfly valve during control is shown in Figure 11. In these simulations, the amplitude of its oscillations is approximately 45 degrees, between 25 and 70 degrees.

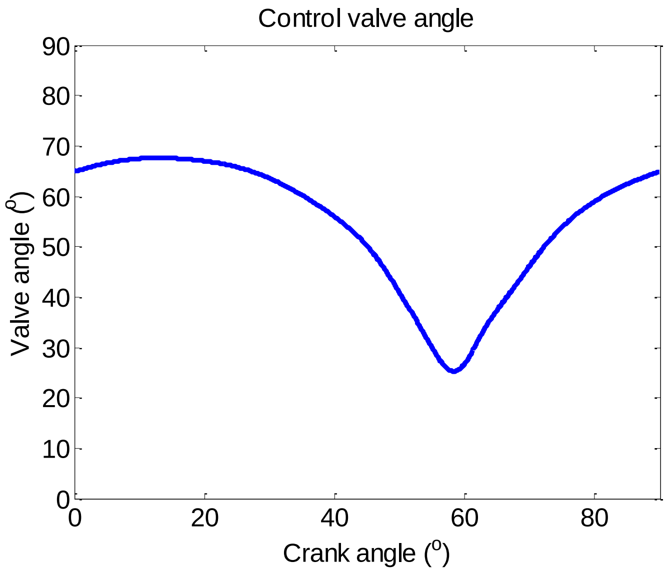

Figure 11 Angular rotation of the butterfly valve over one-quarter of an engine revolution.

4. CONCLUSION

The concept of active flow regulation to minimise exhaust noise has been demonstrated in previous work, using a buffer volume and a regulating valve inserted in the exhaust line. The purpose of the work conducted here is to estimate the performance of an active exhaust silencer for parameters such as exhaust back-pressure and angular rotation of the butterfly valve, and expected noise reduction, when this technology is applied to a medium size diesel engine.

Time dependent simulations of a somewhat idealised controller were applied to the model of a 16-litre diesel engine. Using performance data of a commercially available butterfly valve, it was shown that by using a buffer volume of 50 litres in the exhaust line between the engine and the control valve, cancellation of the exhaust noise at the cylinder firing frequency would result in an increase of the back-pressure by approximately 3 kPa (0.03 atm.) at rated power for this 420 kW engine. In subsequent experiments on the demonstration test platform, this buffer volume will be accommodated by the proposed two-stage expansion chamber silencer and upstream exhaust duct, and hence no additional volume would be required. The back pressure increase due to the installation of the active exhaust silencer can be reduced by increasing the size of the buffer volume, which is an important consideration for the subsequent optimisation of the system against engine operational requirements.

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