Iterative learning control for an active exhaust noise attenuation valve for internal combustion engines.

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Abstract

An active silencer has been developed to attenuate combustion engine exhaust noise. The silencer consists of an electrically controlled valve connected to a buffer volume. The pulsating flow from the engine is buffered in the volume. The valve connected to it is controlled such that only the mean flow passes to the atmosphere. This flow is free from fluctuations and consequently free of sound. An iterative learning controller has been developed to control the active valve. In the control algorithm, non-causal filters are implemented. These filters allow to control systems with high dynamics over a broad frequency range. The active silencer has been experimentally validated on a cold engine simulator, which generates realistic exhaust noise and gas flow using compressed air. The exhaust noise has been reduced in a frequency range starting from 5 Hz until 800 Hz. Depending on the rotational speed of the engine, the typical reductions ranges from 13 dB to 24 dB.

Introduction

The reduction of noise emission is an important issue in car development. The attenuation of exhaust noise using passive systems results in voluminous or high restrictive silencers. Active systems can attenuate the exhaust noise less restrictive with smaller volumes, particularly for the lower frequencies (typically below 200 Hz). A lot of academic research has been carried out, and the technology is now under investigation at several exhaust manufacturers (KEBA, Ricardo, Faurecia, Bosal, etc ...). The two different technologies under investigation are loudspeakers and valve systems.

The proposed approach in this paper is the combination of an electrically controlled valve and a buffer volume. The valve concept has much higher efficiency as a loudspeaker concept, particularly at very low engine revolutions. Basically, the engine behaves as a volume velocity source. The combusted gas is swept out the cylinder by the piston. These flow pulses charge the buffer volume. The control valve, directly connected to the buffer volume, controls the flow from the buffer volume to the atmosphere such that only the mean flow is passed. This flow is free of fluctuations and consequently free of noise.

The purpose of the presented simulation models is to develop the actuator for the silencer, to create a prototype. Then, an iterative learning controller (ILC) will be developed and implemented on the active silencer. The system will be able to attenuate the global exhaust noise with 13 dB to 23 dB in a frequency range until 1 kHz.

Global analog circuit.

The global model [1] is represented as an electrical analog circuit, as illustrated in figure 1.

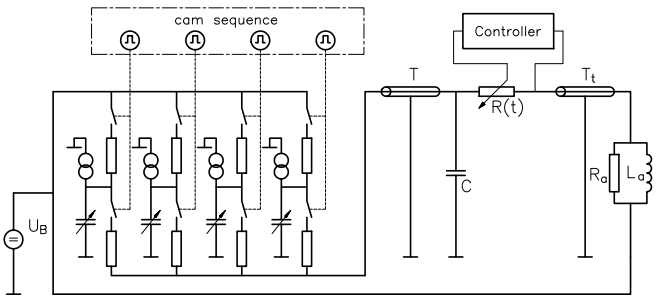

Figure 1: Electrical analog model of an engine equipped with the active exhaust system.

The left part is the engine model. The four capacitors represent the four engine cylinders, who's volume varies sinusoidal between maximum and dead volume. The combustion is simulated by charging the capacitor by a pulsing current source parallel over the capacitor. The upper set of switch-resistors represent the intake valves, the lower set the exhaust valves. The switches are actuated in the same sequence as the camshaft actuates the engine valves. The intake side is connected to a voltage source U_B representing the atmospheric pressure. The right part represents the active exhaust system. The silencer is connected to the engine via the duct represented by the transmission line T. The capacitor C represents the buffer volume and the variable resistor $R(t)$ the control valve. The transmission line T_t represents the tailpipe. The resistor-inductor combination R_a-L_a corresponds to the free air impedance. In simulation, a collocated feedback controller regulates the control valve using the pressure signal behind the valve. In practice, other control strategies need to be applied to handle the time delay between the valve action and its effect in the error sensor.

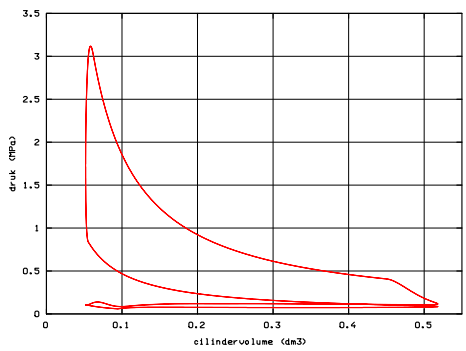

Figure 2: Simulated indicator diagram.

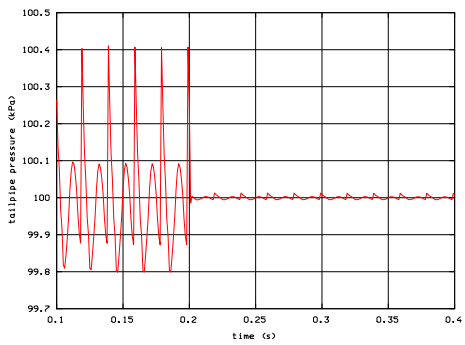

Figure 3: Tailpipe pressure. Control starts at 0.2 s.

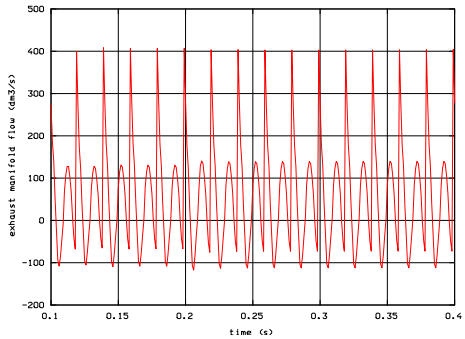

Figure 4: Gas flow through the exhaust manifold.

The simulation results are displayed in figures 2, 3 and 4. In figure 2, the engine indicator diagram is presented. This diagram has no direct physical significance, because it is an isothermal simulation. Only the remaining pressure at the exhaust valve opening time point is deterministic for the exhaust noise. In figure 3, the pressure in the tailpipe is presented. The controller is activated at 0.2 s. Figure 4 represents the gas flow from the engine exhaust to the active silencer. The gas flow is not affected by the controller action. The gas flow will be used as input data for the detailed active exhaust model.

During this simulation stage, the back pressure to the engine, the exhaust system resistance. etc ... are investigated to optimize the active exhaust system configuration.

Active exhaust system circuit.

Once an optimal configuration is found, the control valve itself has to be developed. The electrical analog circuit, displayed in figure 5, embeds the necessary elements how to construct the control valve.

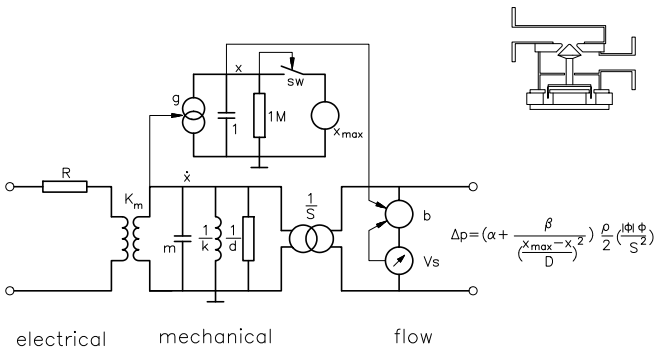

Figure 5: Electrical analog model for the voice coil driven control valve.

The control valve consists of a conical valve head which regulates an orifice opening. The valve head is driven by a voice coil in a permanent magnet. The voice coil resistance is displayed in the electrical part. The magnet assembly transforms the electrical current in mechanical force by the transformer K_m. The force acts on the valve head mass m suspended on a spring k with damping d. The resulting velocity $\dot{x}$ is integrated to obtain the valve head displacement x. The displacement is limited by $x_{\max}$, where the valve is completely closed. The gas pressure drop over the valve is a function of the valve head displacement and the flow through the valve. The valve resistance characteristic is displayed in figure 6. The pressure drop Δp is generated by the voltage source b. The generated pressure acts also on the valve head. This force is coupled back to the mechanical circuit by the gyrator S, representing the valve orifice surface.

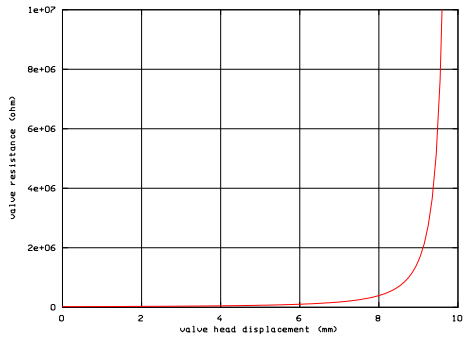

Figure 6: Control valve flow resistance as function of valve head displacement.

At the gas flow terminals of this circuit, the buffer volume capacitor and a volume velocity source, generating the disturbing gas flow presented in figure 4, is connected. At the electrical terminals, a collocated feedback controller is connected. The acoustic flow through the valve is used as error signal.

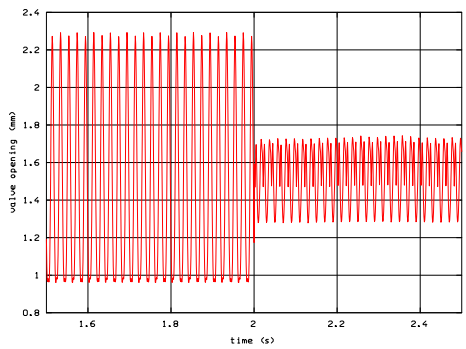

Figure 7: Valve head displacement. The control starts at 2 s.

From these simulation results, the mechanical dimensions of the control valve can be determined. Figure 7 presents the valve head displacement. The controller starts at 2 s. The displacement of the valve head under control remains below 0.5 mm around a mean opening of 1.5 mm. The voice coil stroke and optimal position is a direct consequence of the valve displacement. The electrical current to drive the voice coil is displayed in figure 8. This current depends on the drive magnet, the moving mass and the displacement. The DC-current can be adjusted by pretensioning the suspension spring of the

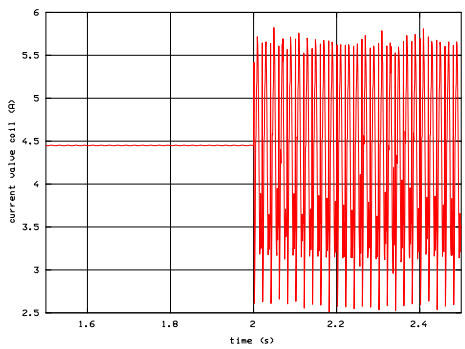

Figure 8: Current through the control valve voice coil.

valve head. For this valve, the DC-current is 4.5 A, the AC-current is $1.5\,\mathrm{A_{RMS}}$. This results in 90 W total power, wherein 80 W is needed to position the valve against the backpressure in the exhaust system. The resulting attenuation of the flow pulsation through the tailpipe is presented in figure 9.

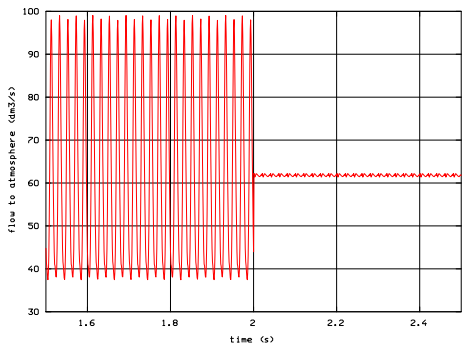

Figure 9: Attenuation of the gas flow pulsation in the tailpipe.

Design of the controller.

An experimental setup, presented in figure 10 has been created with an active silencer developed using the results of the simulation. As controller, an iterative learning controller (ILC) using non-causal control filters [2] has been developed and implemented. The scheme of the controller is presented in figure 11.

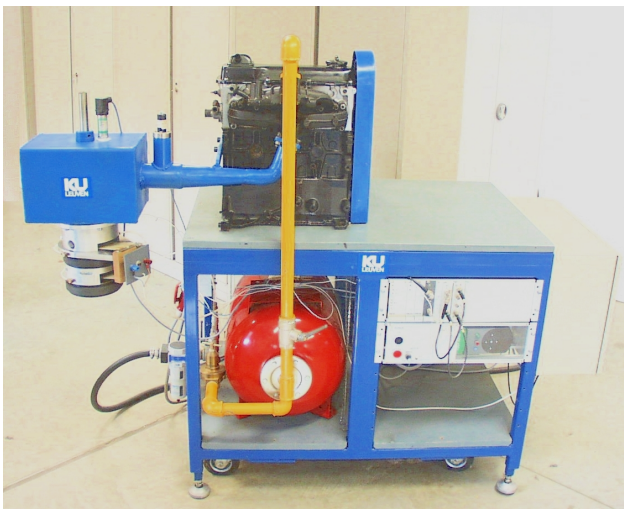

Figure 10: Active exhaust system experiments on a cold engine simulator.

The ILC-controller consists of a memory $E^{-s\,T_p}$ wherein

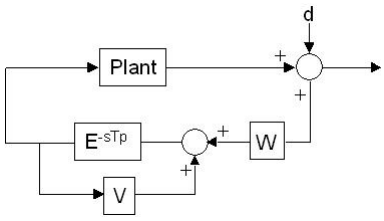

Figure 11: Scheme of the iterative learning controller (ILC).

the signal to cancel the noise pulse is stored. When the pulse d occurs, the controller sends the contents of the memory to the noise canceling actuator. The memory of the controller will be reloaded with the previous memory contents filtered by the filter V. The residual noise, measured with a pressure sensor, will be filtered with the learning filter W and is added to the contents of the memory. When the next pulse occurs, the cycle repeats.

The ILC-controller controls the next event using information from the previous events. It is only suitable for repetitive events. As the events occur in the future, it is possible to use a non-causal filter as learning filter W. For the filter V, a constant $V < 1$ is sufficient. A non-causal filter can combine phase lead with amplitude drop, which is impossible for causal filters such as used in feedback control.

The most important design parameters are the convergence criterion and the performance criterion. The convergence criterion:

$$||V - P\,W|| < 1 \tag{1}$$

wherein P is the physical plant transfer function and V and W the control filters transfer functions, has to be fullfilled to obtain a stable controller. In the Nyquist plane, the loop of the transfer function $W\,P$ has to remain within the unit circle with centre point $(V, 0)$, as illustrated in figure 15. The performance criterion:

$$E_\infty = \frac{V-1}{1-V+P\,W}\,d \tag{2}$$

expresses the maximum attenuation E_∞ of the disturbance d after convergence of the controller after a number of repetitive events in terms of the control filters V, W and the plant P.

As the engine produces similar exhaust pressure pulses, an ILC-controller is applicable. First, the plant transfer function P is measured between the output pressure behind the actuator valve of the active silencer and the input voltage to the actuator valve power amplifier. The result is presented in figure 12 in amplitude and phase. Due to the dynamics of the valve and the piping around, a continuous drop of the phase occurs which makes feedback control with a bandwidth above 100 Hz very difficult.

In ILC-control, a non-causal filter with a continous phase lead is designed. The resulting non-causal W filter is presented in figure 13. The V-filter is choosen to be constant, $V = 0.98$. The transfer function $W\,P$ is presented in figure 14. The frequency region wherein

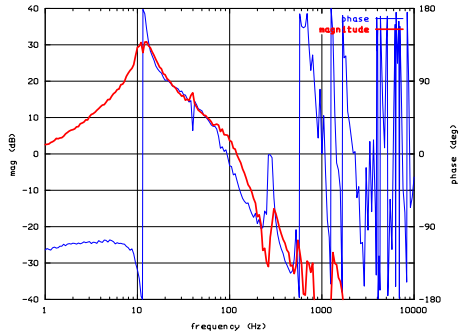

Figure 12: Plant transfer function P.

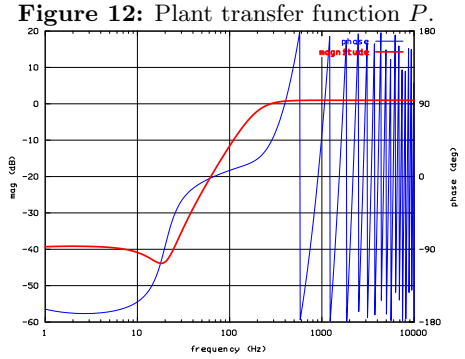

Figure 13: Learning filter transfer function W.

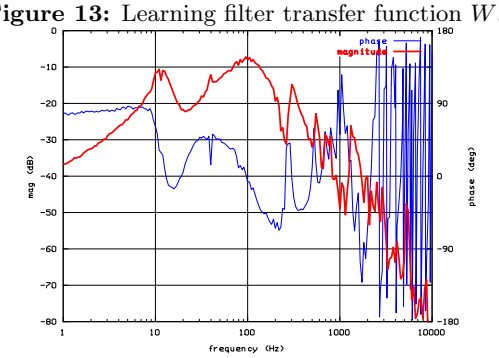

Figure 14: Transfer function $W\ P$

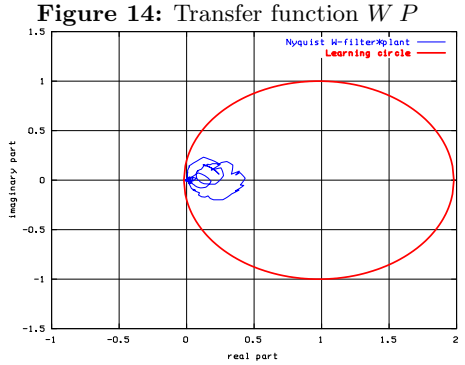

Figure 15: Transfer function $W\ P$ inside the unit learning circle in Nyquist plane.

reduction of noise can be achived is the region wherein the phase remains between -90 and 90 degrees. In this case, this region will be until about 1 kHz. When plotting the transfer function in the Nyquist plane, as displayed in figure 15, the loop of the transfer function remains in the unit circle with centre point $(0.98, 0)$. Consequently, the controller will be stable. The maximum reduction of the exhaust noise, determined using expression (2), is presented in figure 16. Reduction will be obtained until 1 kHz, with maxima higher than 20 dB around $10\,Hz$ and 100 Hz. Figure 17 displays the resulting attenuation of the noise at the exhaust opening after convergence of the controller.

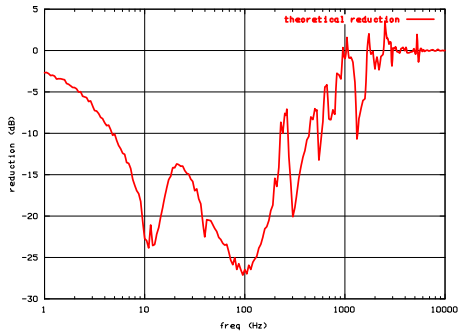

Figure 16: Maximum achieveble reduction E_∞/d after convergence of the controller.

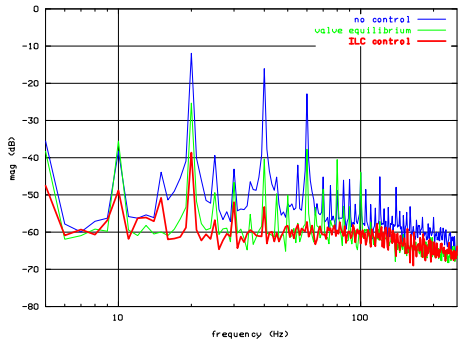

Figure 17: Reduction of the noise at the exhaust outlet.

The cold engine runs at 600 rpm, generating harmonics at 20 Hz, 40 Hz, 60 Hz, etc . . . . When applying a DC-current trough the attenuation valve, an attenuation of about 15 dB occurs at the three first harmonics due to the valve resistance. The valve introduces a backpressure of 10 kPa in the exhaust manifold. When activating the ILC-controller, an additional 15 dB reduction of all harmonics occur, without increase of the backpressure. The global reduction of the exhaust noise amounts 23 dB. In the engine speed range until 1600 rpm, the global noise reduction was always above 13 dB.

Conclusion.

An active exhaust silencer has been developed using electrical analog circuits. In the global circuit, the active exhaust system can be optimized. This simulation provides the necessary data for the simulation model wherein the actual control valve itself is developed. From these simulation results, a prototype active exhaust can be generated. An iterative learing controller has been developed for the active silencer. The exhaust noise has been reduced with 13 to 24 dB in a frequency range until 1 kHz.

References

[1] René Boonen, Paul Sas, Design of an active exhaust attenuating valve for internal combustion engines, proc. of the ISMA2002 conference, Leuven, Belgium, **1** (2002), 33-42 (*also available at* `www.isma-isaac.be`)

[2] G. Pinte, R. Boonen, W. Desmet, Paul Sas, Active structural acoustic control of repetitive impact noise, Journal of Sound and Vibration, **319**, (2009), 768-794.