Development of an active exhaust silencer for internal combustion engines

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Abstract

A silencer to attenuate engine exhaust noise using active control methods is developed. The device consists of an electrically driven valve, combined with a buffer volume, which is connected to the exhaust outlet. Using the mean flow through the valve and the pressure fluctuations in the volume, the valve regulates the flow in such a way that only the mean flow passes through the exhaust outlet. The fluctuations of the flow are temporally buffered in the volume.

To carry out optimization and validation experiments, a cold engine simulator is developed. This device generates realistic exhaust noise and the matching gas flow using compressed air. The simulator allows quick and reliable acoustic and fluid dynamic experiments on exhaust prototypes.

The active silencer is capable to reduce the exhaust noise from 91 dBA to 78 dBA after the tail pipe outlet, with a back pressure of 3 kPa to the engine.

1. Introduction

This research project aims at the attenuation of engine exhaust noise using active control methods. The resulting system should take less volume and should have a positive energy balance in comparison with a fully passive system.

The reduction of external noise is nowadays one of the important issues in car development. The legislators are lowering the noise emission standards continuously in the different countries. For example, the pass by noise level for passenger cars is reduced from maximum 77 dBA to 74 dBA in 1995. Also, in engine development, multi valve technology is becoming common to increase the engine efficiency by lowering in- and outlet valve resistance. As consequence, exhaust system manufacturers must reach higher attenuation levels in combination with lower flow resistance.

Much research on active noise cancellation in ducts is carried out in the recent years. Loudspeaker systems are successfully applied and commercial available in ventilation channels [15]. A loudspeaker setup is developed for stationary six-cylinder diesel engines by Detroit Diesel Corporation [13]. A problem using loudspeakers is the low sound generating efficiency and reliability in the extreme conditions of an engine exhaust.

To increase sound generation performance, a loudspeaker with high diaphragm displacement, suitable for the low frequency range, is under development at the technical university of Dresden [19]. If compressed air is available, an electro-pneumatic loudspeaker is also an option [2 (page 49)],[18].

Applying a controllable valve in the exhaust duct is a more robust concept. Good results are achieved mainly in the low frequency range, mostly on fan setups [8],[10]. On internal combustion engines, the sound power generated by the control sound source remains a problem.

In this project, the concept of a controllable valve in the exhaust is used. A solution to the sound power problem is achieved by mounting a buffer volume between the engine exhaust and the control valve. Idealized, the engine behaves as a volume velocity source. It is not possible to control the flow of a volume velocity source, if no capacitive elements are present between the source outlet and the active valve. The capacity of a duct is low, consequently the performance of the valve must be high. By adding an additional volume between exhaust and valve, the required noise generation performance of the valve is reduced. By balancing the volume and the valve performance, it is always possible to realize active noise attenuation for a combustion engine.

The dimensioning of the active silencer is based on acoustic simulations using electrical analogies [1]. An electrical equivalent circuit is developed which consists of partial circuits for the engine, the active exhaust system and the controller. The simulation predicts a noise level reduction of approximately 27 dB, directly after the controlled valve.

A main problem when conducting experiments is the availability of a representative sound source. Directly testing on an internal combustion engine is difficult because it is necessary to take precautions against the hot corrosive exhaust gases. Carrying out experiments on loudspeaker and fan setups results in unreliable data, because the acoustical circumstances on these setups differ considerably from these of an operating engine. Therefore, a cold engine simulator has been developed which generates the engine sound with the matching gas flow in a realistic way using compressed air [16],[17]. More generally, the simulator is applicable for the study of sound behaviour and flow phenomena in any kind of exhaust system. The simulator has been validated by comparing experimental data with numerical simulations and experiments on a real combustion engine. These comparisons demonstrate that the cold engine simulator approximates the exhaust noise of an internal combustion engine very well.

The experimental setup consists of an electrically driven valve, in series with a volume, which is connected to the exhaust outlet of the cold engine simulator. The control signal for the valve is calculated from the pressure in the buffer volume and the flow through the exhaust duct. The valve is controlled such that only the mean flow passes the valve opening. The flow fluctuations are temporally stored in the volume. The control algorithm is a feed forward ”inverse plant” algorithm, without adaptation. This results in a fast algorithm, but it requires the active silencer to follow the theoretical proposed plant without major deviations in the desired frequency range. This approach results in a noise reduction from 91 dBA to 78 dBA, measured 10 cm out of the axis of the tail pipe opening, nevertheless, the ”inverse plant” approach limits the noise reduction capability considerably. In the near future, feed back and adaptive control algorithms will be implemented on the controller and tested.

2. The active silencer

2.1 Principle

To control the flow in the exhaust duct, a valve is placed in the flow. The resistance of the valve is continuously variable by applying an external signal. It is assumed that the valve is purely resistive, it isn’t capable to store energy from the gas flow. The principle of using a valve to control duct flow is illustrated in figure 1.

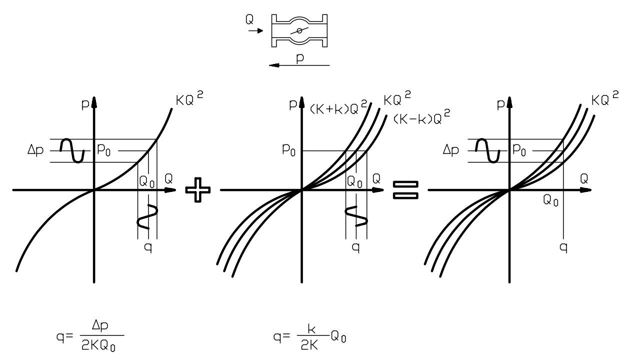

figure 1: Basic principle of using a valve controlling duct flow. The meaning of the symbols in this figure is: Q_0 is mean fluid flow, q is flow fluctuation, p_0 is mean pressure, Δp is the pressure fluctuation, K is the valve constant which expresses the relation between pressure and flow, k is the variation on K.

A valve placed in a volume flow causes a pressure drop over the valve. The principle of the active noise cancellation valve is, to obtain a constant volume flow despite a fluctuating pressure, by varying the valve resistance.

In figure 1, the first graph demonstrates how sound pressure influences the fluid flow via the valve resistance characteristic. The second graph demonstrates how the opposite fluctuating flow is generated from the mean pressure drop over the valve by varying the valve resistance. Superposition of both effects results in a constant volume flow, shown in the third graph.

Mounting an active valve in the flow of a volume velocity source, like a combustion engine, has no effect if no capacitive elements are present between the source and the valve. The volume velocity source forces a prescribed flow through the system, whatever the pressure becomes in the system. Capacitive elements can be introduced using ducts or volumes.

The most simple system for active control of a volume velocity source is presented in figure 2. The engine acts as a volume velocity source. At the exhaust, a volume with capacity C and a regulating valve with variable resistance $R(t)$ is connected. The translation of the physical system results in the electrical equivalent circuit shown in figure 2. The flow from the source will split over the capacity and the time dependent resistor. Now, the controller has to vary in time the valve resistance, such that the fluctuation flow through the resistance becomes zero.

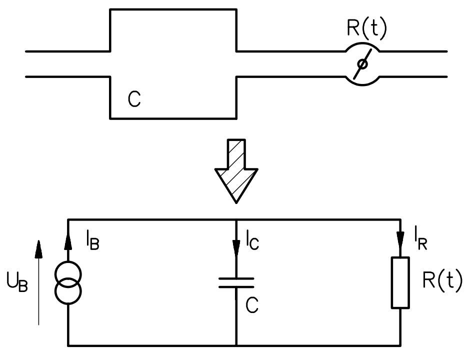

figure 2: The most simple scheme of active control of an engine using a valve.

The variation of the valve resistance is obtained from the electrical equivalent circuit:

$$U_B = \frac{1}{C} \int I_C \mathrm{d}\,t = I_R\, R(t)$$

$$I_C\, \frac{1}{C} = \frac{\mathrm{d}\, I_R}{\mathrm{d}\, t}\, R(t) + I_R\, \frac{\mathrm{d}\, R(t)}{\mathrm{d}\, t}$$

Constant flow through the valve opening means:

$$\frac{\mathrm{d}\, I_R}{\mathrm{d}\, t} = 0$$

The controller has to vary the valve resistance during time according:

$$R(t) = R_0 + \int \frac{1}{C}\, \frac{I_C}{I_R}\, \mathrm{d}\, t \qquad (1)$$

Two important conclusions follows from this simple consideration: First, by balancing the volume-valve combination, it is always possible to control the flow of a volume velocity source. Second, the resistance R_0 can be chosen freely with the only restriction that the resistance $R(t)$ remains always positive. The resistance R_0 must be optimized to obtain minimum energy loss of the engine.

2.2 Expanded electrical equivalent model

The model above looks not very realistic for an engine exhaust system, therefore the model will be expanded. The volume velocity source will be replaced by an engine model, and an exhaust duct is connected between the engine and the active silencer. The controller uses expression (1) as control algorithm. A scheme of this system is presented in figure 3.

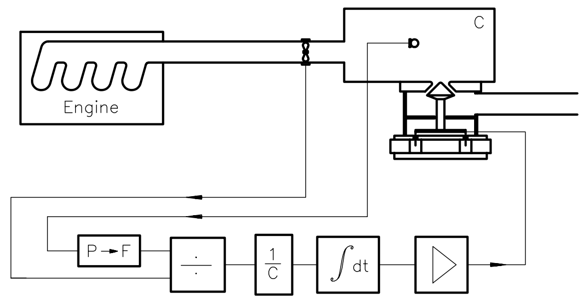

figure 3: Scheme of an internal combustion engine with an active exhaust noise cancellation device.

The electrical equivalent circuit of the system shown in figure 3 is presented in figure 4.

The exhaust of the engine is considered as a series of discharges of constant volumes. When the outlet valve of a combustion engine opens, the remaining cylinder pressure drops in a few milliseconds to atmospheric pressure. The piston is at its lower dead point and the cylinder volume changes only 10 á 15 % during the escape time of the pressure pulse.

In the electrical equivalent model, the four capacitors represent the engine cylinders. The voltage of the source U_B equals the remaining pressure at the end of the expansion stroke of the engine. The capacitors are charged via the upper set of switch–resistor combinations, which represent the inlet valves of the engine. The capacitors are discharged via the lower set of switch–resistor combinations, which represent the outlet valves. The switches are operated in the same sequence as the cam shaft operates the engine valves. The pulses enter the exhaust duct, represented by the transmission line T. The capacitor C and the variable resistor $R(t)$ forms the equivalent of the active silencer. The controller uses the current through the transmission line and the voltage over the capacitor to generate the control signal for the valve $R(t)$, using expression (1) as control algorithm. The inductor-resistor combination, representing the acoustic impedance of free air, closes the circuit.

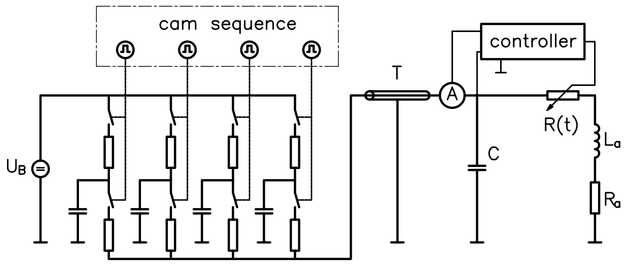

figure 4: Electrical analog circuit for a four cylinder engine with an active noise cancellation device.

The simulation is carried out in time domain using a 1000 cc engine, a duct of 1 m length and 10^{-3}m^2 cross-section, a buffer volume of $12 \cdot 10^{-3}\text{m}^3$ and a control valve resistance which can vary between 50kΩ and 300kΩ.

The simulation result is presented in figure 5. Both lines represent the pressure directly after the control valve, before the outlet of the exhaust tail pipe. Before activating the controller, the pressure, represented by "line 1", corresponds with a sound pressure level of 153 dB in the tail pipe. When the controller is activated, the pressure reduces to a sound pressure level of 126 dB.

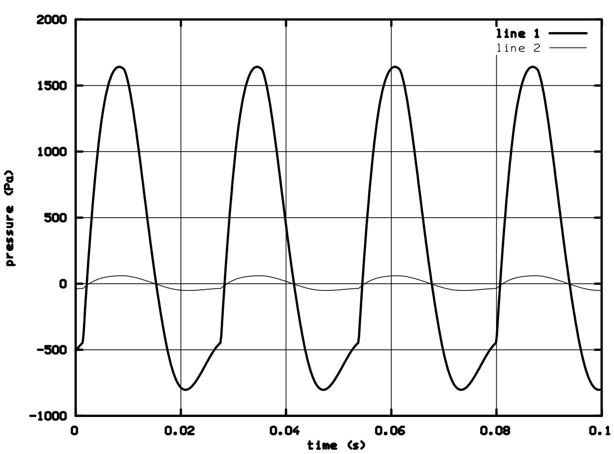

figure 5: Simulation result of the circuit presented in figure 4. "line 1" is the pressure at the outlet opening without control, "line 2" is the pressure with control.

In frequency domain, the active control results in a shift of the noise spectrum with 27 dB downwards. This is illustrated in figure 6.

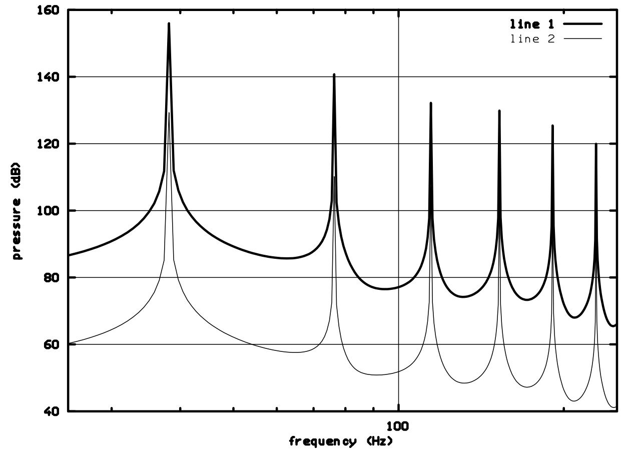

figure 6: Frequency spectrum of the time signal presented in figure 5. "line 1" is the pressure without control, "line 2" is the pressure with control.

3. Experimental results

3.1 The cold engine simulator

To carry out experiments with the active silencer, a cold engine simulator is developed which generates realistic engine noise and the matching gas flow using compressed air. It permits to experiment with new concepts of exhaust systems, without taking precautions against the hot corrosive gases of a real engine.

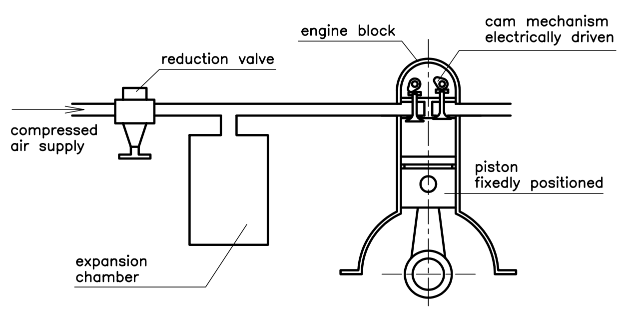

figure 7: Working scheme of the cold engine simulator.

The assumption that the pressure pulse can be considered as a discharge of a constant volume gas forms also the basis for the cold engine simulator. A scheme of the engine simulator is presented in figure 7. It consists of a regular engine block whose pistons are fixed at their lower dead points. The inlet collector is connected via an expansion vessel and a pressure reduction valve to a normal pressurized air supply network. The cam mechanism of the engine block is driven by an electric motor. The supplied pressure at the inlet collector is equal to the pressure in the cylinder of an operational combustion engine at the end of the expansion stage. During the inlet stage, the cylinder charges at the same pressure level as applied at the inlet. When the outlet opens, the cylinder discharges and the pressure pulse enters the exhaust. The discharge takes a few milliseconds. These pressure pulses are similar to these of a real combustion engine.

To validate the cold engine simulator, pressure measurements are carried out on the simulator and a real combustion engine. On the exhaust, a straight duct is connected of 7 m length and $10^{-3}\mathrm{m}^2$ cross-section. The crank rotation speed is 300 rpm. The applied pressure at the inlet collector equals 200 kPa.

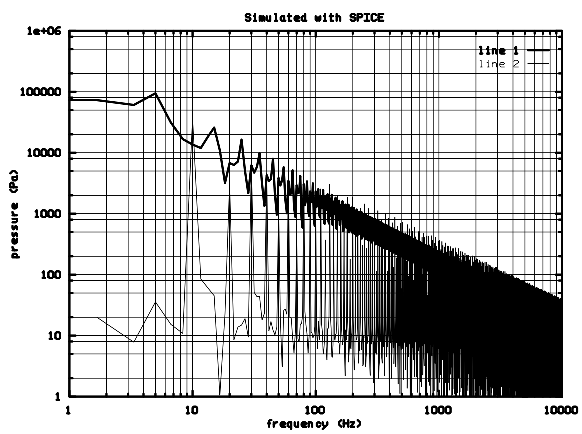

figure 8: Frequency spectrum of cylinder pressure (”line 1”) and manifold pressure (”line 2”) simulated from the electrical equivalent circuit.

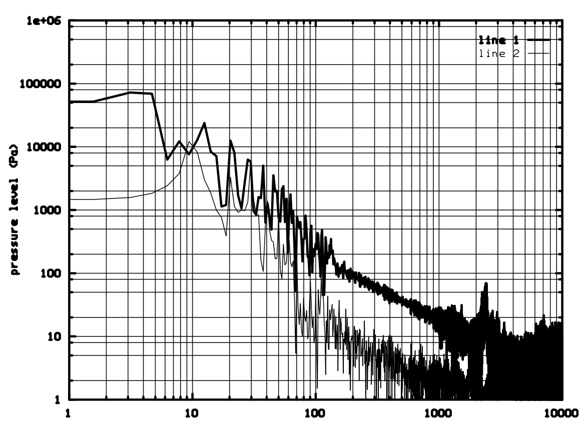

figure 9: Frequency spectrum of cylinder pressure (”line 1”) and manifold pressure (”line 2”), measured on the cold engine simulator.

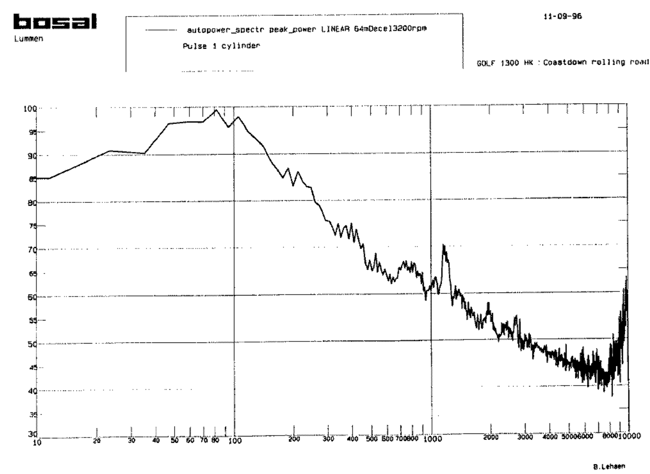

figure 10: Measurement of frequency spectrum of the manifold pressure of an automobile combustion engine, connected to a straight exhaust duct. (Source: BOSAL Research)

Figure 8 shows the simulated pressures of the electrical equivalent circuit of the engine in figure 4, except that only a transmission line and the open air impedance are connected to the engine model. The peaks, appearing at firing frequency, of both cylinder pressure (”line 1”) and manifold pressure (”line 2”) show a -20 dB/decade decay. This behaviour is caused by the capacitance of the cylinder volumes.

Figure 9 shows the measurement of the cylinder and manifold pressure on the cold engine simulator. The measurements are taken at the same conditions as used in the simulation. ”line 1” is the cylinder pressure and ”line 2” is the pressure in the outlet duct, 20 cm after the outlet valves.

The -20 dB/decade decay is dominantly present. After 200 Hz, the higher frequencies show more damping as in the simulation. These damping effects are introduced by many causes. The most important ones are: the finite opening times of the outlet valves; the saturation of the flow in the valve opening; the difference in sound speed in the duct in forward and backward direction, due to the mean gas flow through the duct.

The measurement, shown in figure 10, is the frequency spectrum of the manifold pressure of a real internal combustion engine. Here also, the -20 dB/decade slope is dominantly present, as in the two previous figures. The cold engine simulator These figures demonstrate that the cold engine simulator and the internal combustion engine exhibit a very similar behaviour. It demonstrates also the usefulness of the equivalent circuit for the engine.

3.2 The active silencer

Experiments with the active silencer are carried out on the cold engine simulator. A picture of the setup in shown in figure 11. The electrically controllable valve is mounted on the buffer volume. As control signals, a piezo-resistive sensor measures the mean pressure in the volume, and is filtered by an analog 1 Hz second order low pass filter. The alternating (acoustic) pressure in the volume is measured by a piezo-capacitive sensor and is filtered by an analog 2000 Hz second order low pass filter. The mean flow through the outlet duct is measured by a small turbine and is also filtered by an analog 1 Hz second order low pass filter.

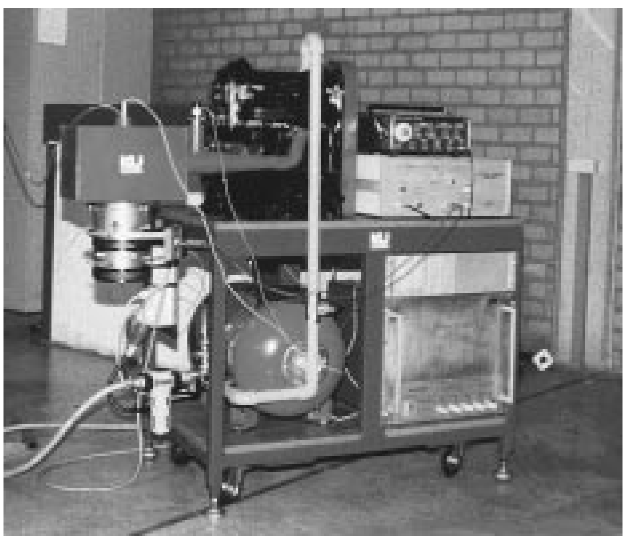

figure 11: The experimental setup of the active silenceR on the exhaust of the cold engine simulator.

As figure 3 suggests, a voice coil in a magnet actuates the valve head to control the valve opening. The voice coil is powered by a current amplifier, with a maximum electrical power of 500 W. Before the current amplifier, a preamplifier buffers and filters the output signal from the controller. The filter is also an analog 2000 Hz second order low pass filter. By choosing these second order filters, the phase lag at 500 Hz is not higher than 45^o after passing all in- and output filters. As consequence, the valve should be able to attenuate pulses of at least 2 ms pulse duration.

At low frequencies (below 17 Hz), the applied force on the voice coil is almost equal to the pressure drop over the valve times the valve opening area. At higher frequencies, the inertia of the valve motor becomes dominant and the force determines the acceleration of the valve head.

The controller is split in two parts. A low bandwidth PI-controller realizes the set point resistance R_0, where around the active noise cancellation is carried out, using the piezo-resistive sensor. This controller uses the linear relationship between the applied force and the resulting pressure.

The active noise controller constructs the control signal using the flow through the duct and the pressure in the volume, based on equation (1). The input signals are taken from the piezo-capacitive sensor and the flow sensor. This controller takes the valve motor inertia into account to determine the resulting valve head displacement. This controller is not active in the low frequency range by implementing a 10 Hz first order high pass filter on the controller input signal.

The sum of the signals of both controllers is applied to the preamplifier of the current amplifier. Both controllers are implemented in *Z*-domain on a C40 digital signal processor of Texas Instruments, using a sample frequency of 16 kHz.

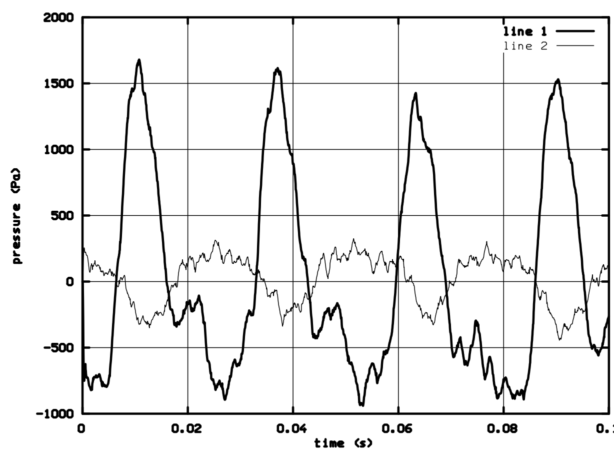

figure 12: Measurement of the pressure after the control valve in the outlet duct. ”line 1” is the pressure without control, ”line 2” is the pressure with control.

In figure 12, the result of the control is presented. The pressures are measured in the tail pipe after the control valve using a pressure sensor. The pressure ”line 1” without control corresponds with the original sound pressure level of 151 dB. By activating the control, the pressure ”line 2” reduces to a sound pressure level of 138 dB. The realized noise reduction in the tail pipe amounts 13 dB. The same reduction is measured with a sound level meter 10 cm out of the axis of the outlet of the exhaust tail pipe. The radiated noise reduces with 13 dBA from 91 dBA to 78 dBA by activating the controller.

The frequency spectrum in figure 13 of the pressure in the tail pipe presented in figure 12 shifts 13 dB downwards, by activating the controller.

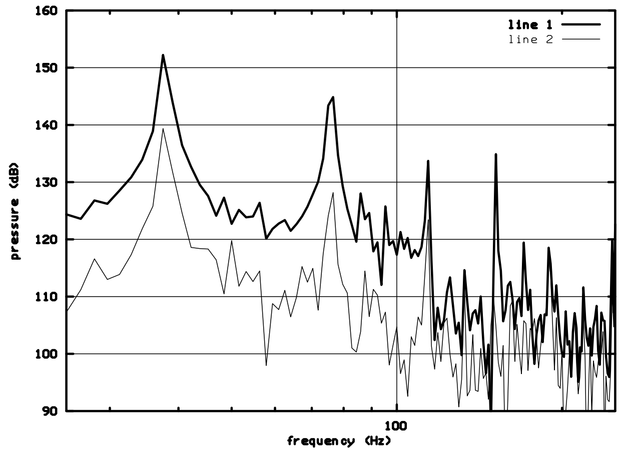

figure 13: Frequency spectrum of the time signals displayed in figure 12. ”line 1” is the pressure without control, ”line 2” is the pressure with control.

The controller uses a linearized model of the valve resistance as function of the valve opening. The valve resistance depends quadraticly as function of the valve opening and this effect results in a distortion of the pressure drop over the valve. This limits the sound reduction performance of the active muffler. In the near future, feed back and adaptive feed forward control strategies will be implemented and tested, to minimize the influence of the quadratic valve resistance.

4. Future investigations

Before creating an active silencer suitable for a real internal combustion engine, energy consumption and dimensions have to be optimized. The dimensions of the valve can be reduced considerably. This results also in a lower electrical energy consumption. Reducing the buffer volume dimensions results in a higher back pressure for the engine and consequently in a higher energy consumption from the exhaust gas flow. The buffer volume and the valve have to be balanced to obtain minimum energy consumption in an active silencer in realistic dimensions, suitable for passenger car combustion engines.

5. Conclusion

The actual state of this research project permits to formulate following conclusions:

- By preconnecting an active controlled valve in an exhaust duct by a buffer volume, it becomes possible to attenuate any engine exhaust noise.
- The cold engine simulator produces realistic engine exhaust noise and gas flow using compressed air. It allows quick and reliable experiments with new concepts of exhaust systems.
- The active silencer reduces the engine exhaust noise with 13 dBA using a feed forward ”inverse plant” algorithm. More attenuation can be reached using algorithms less sensitive to errors in the plant model.

6. Acknowledgment

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References

1. Leo L. Beranek, *Acoustics* , Mc Graw-Hill, (1954)
2. M. Heckl, H. A. Müller, *Taschenbuch der Technische Akustiek* , Springer Verlag, (1975)
3. D.D. Davis,Jr., G.M. Stokes, D. Moore and G.L. Stevens,Jr., *Theoretical and experimental investigation of mufflers with comments on engine exhaust design* , Report 1192, Langley Aeronautical Laboratory, (1954)
4. D.E. Baxa (editor), *Noise control in internal combustion engines* , John Wiley & Sons, (1982)
5. M. G. Prasard, *A four load method for evaluation of acoustical source impedance in a duct* , Journal of Sound and Vibration, **114**(2),347-356(1987)
6. J. Kergomard, A. Garcia, G. Tagui and J. P. Dalmont, *Analysis of higher order mode effects in an expansion chamber using modal theory and equivalent electrical circuits* , Journal of Sound and Vibration, **12**(3),457-475(1989)

7. S. D. Snyder, C. H. Hansen, *Active noise control in ducts: some physical insights* , Journal Acoust. Soc. Am., **86**(1),184,(1989)
8. S. M. Hutchins, J. Cherqui, *Active Exhaust Noise Cancellation* , ISATA 91 conference, (May 1991)
9. H. R. Hall, F. W. Brendt, R. J. Bernhard, *Active control of radiated sound from ducts* , Transactions of the ASME, **114**, (July 1992)
10. L. Hardouin, P. Micheau, J. Tartarin, L. Laumonier, *An anti-pulsatory device used as an active noise control system in a duct* , acta acoustica, **1**, 189-198 (1993)
11. Hans Bodén and Mats Åbom, *Modeling of fluid machines as sources of sound in duct and pipe systems* , acta acoustica, **3**, 594-560 (1995)
12. A. Selamet, N.S. Dickey, *A time domain computational simulation of acoustic silencers* , Journal of vibration and acoustics, **117**, 323-331, (July 1995)
13. Everett Arnold and Warner Frazer, *Development of a prototype active muffler for the Detroit diesel 6V-92 TA industrial engine* , Detroit Diesel Corporation
14. G. P. Blair, J. A. Spechko, *Sound pressure levels generated by internal combustion engines exhaust systems* , SAE 720155
15. J. Tichy, *Applications for active control of sound and vibration* , Noise News International, 73-86, (June 1996)
16. R. Boonen, P. Sas, *Determination of acoustic impedance of an IC-engine with validation on a cold engine simulator for exhaust noise* , Proceedings of the ISMA21 conference, Leuven,**1** ,389-400, (1996)
17. R. Boonen, P. Sas, *Design of an active noise cancellation device for internal combustion engines* , Proceedings of the ACTIVE97 conference, Budapest, 499-512, (1997)
18. L.A. Blondel, S.J. Elliot, *Subsonic compressed air sources for active noise control* , Proceedings of the ACTIVE97 conference, Budapest, 619-634, (1997)
19. Torsten Obier, Günther Pfeifer, *An alternative source for high sound pressures and low frequencies* , Proceedings of the ACTIVE97 conference, Budapest, 689-698, (1997)