Design of an active exhaust attenuating valve for internal combustion engines.

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Abstract

An active silencer to attenuate internal combustion engine exhaust noise is developed. The silencer consists of an electrically controlled valve connected to a buffer volume. The pulsating flow from the engine is buffered in the volume and the valve resistance is continuously controlled such that only the mean flow passes to the atmosphere. This flow is free of fluctuations and consequently free of sound. The design of the active silencer is carried out using electrical analog circuits. First, the interaction between the active silencer and the engine will be studied using an analog circuit including the combustion engine and a linearized active silencer. Then, a detailed valve model is built in a separate electrical analog circuit. It includes the electrical, the mechanical and the flow-dynamic properties of the actuator valve. The actuator valve concept is then simulated, from which a prototype can be constructed. The active silencer has been tested on a cold engine simulator. This device generates realistic exhaust noise with the associated gas flow using compressed air. The silencer can attenuate pulsations from engines at very low revolution speed, without passive elements preconnected between the engine and the active silencer. This is not possible using loudspeaker based active silencers.

1 Introduction

The internal combustion engine has found its way into a broad spectrum of applications, wherein transportation has far the largest share. The success of the internal combustion engine has also created problems associated with it, such as air pollution and environmental noise.

During the past decades, legislation has come forward which stringently reduces air pollutants, greenhouse gases and noise emission. Today, the noise emission limits are set to 74 dBA for cars and 80 dBA for heavy vehicles [1]. For cars, the emission of CO_2 is limited to 190 g/km (EU III-directive), and will be further reduced to 170 g/km (EU IV) in 2003 and 140 g/km (EU V) in 2008. Although these directives are not related at first sight, they do have their implications on the exhaust system development. It will be necessary to develop exhaust systems with minimum back pressure to the engine, to maximize engine efficiency and this without loss of noise attenuation performance. A way out to this problem is to develop active exhaust systems.

Much research on active noise cancellation in ducts is carried out in the recent years and numerous patents have been generated. Loudspeaker systems are successfully applied and commercial available in ventilation channels [2]. Loudspeaker systems are also developed for stationary diesel engines, for example by Detroit Diesel Corporation [3] and the KEBA AIS^{TM}-system. For cars, active loudspeaker systems are developed for six or more cylinder engines [4]. For four-cylinder engines, loudspeaker systems suffer from problems as the low sound generating efficiency and reliability in the extreme conditions of an engine exhaust.

Applying a controllable valve in the exhaust duct is a more robust concept. The restricting element of the valve can be small and rigidly constructed. It can be exposed to the exhaust gas directly.

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. As consequence of the gas flow, the valve generates a pressure drop over it. Active noise cancellation is achieved, when the volume velocity behind the valve is kept constant.

In figure 1, the first graph demonstrates how the 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 vary-

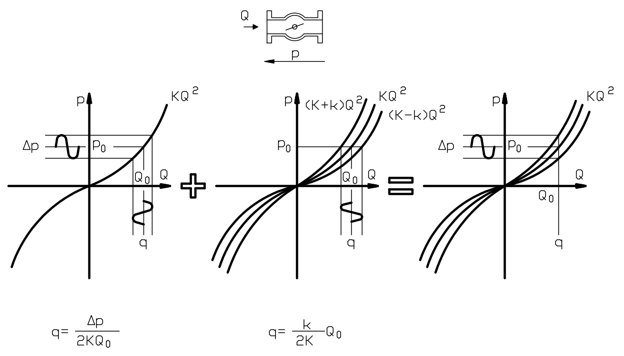

Figure 1: Basic principle using a valve to control the flow through a duct. The symbols in the figure mean: Q_0 is mean fluid flow, q is flow fluctuation, p_0 is mean pressure, Δp is the pressure fluctuation, K is the valve constant which expresses the relation between pressure and flow, k is the variation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

ing the valve resistance. Superposition of both effects results in a constant volume flow, shown in the third graph.

Based on this concept, a noise attenuating device is developed by LEA (University of Poitiers, France) [5, 6], using a butterfly valve, and is tested on an internal combustion engine test rig. The test rig consists of an 1350 cm^3 four cylinder car engine equipped with an hydraulic brake. In the exhaust line two passive mufflers are preconnected to the actuator valve. As control strategy, a x-LMS feedforward and a feedback control system are tested. In the feedforward case, the three first harmonics are reduced by 20 to 30 dB. In the feedback case, the first harmonic is reduced by 20 dB, the second harmonic by 5 dB.

The research work presented in this paper aims to develop an active exhaust silencer based on a valve concept, capable to handle the acoustical power of the pressure pulsation of the engine exhaust without the requirement to install silencers prior to the active device. The generated back pressure must be low. After the active device, a small passive silencer can be installed to attenuate the high frequency noise.

2 Development of the active silencer

2.1 Principle

Theoretically, inserting 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 exhaust system, whatever the pressure becomes in it. Capacitive elements can be introduced using ducts or volumes. When the valve is inserted in an exhaust duct of an engine without additional capacitive elements, a high back pressure will be generated or only a low noise attenuation will result.

The most simple system able to control of a volume velocity source using a controlled valve 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 [7] shown in figure 2 below. The impedance Z is the impedance of the tail pipe and the open air radiator. The flow from the source will split over the capacitor C and the time dependent resistor $R(t)$. Now, the controller has to vary the valve resistance during time, such that the fluctuating flow through the resistance becomes zero.

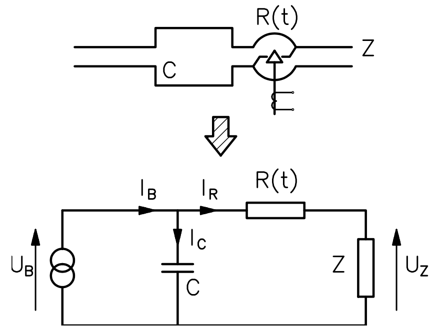

Figure 2: The most simple scheme of an active control valve on an internal combustion engine, approximated as a volume velocity source.

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

$$U_B = \frac{1}{C}\int I_C dt = I_R\,(R(t) + Z) \qquad (1)$$

Differentiating (1) results in:

$$I_C\,\frac{1}{C} = \frac{dI_R}{dt}\,R(t) + I_R\,\frac{dR(t)}{dt} + Z\,\frac{dI_R}{dt} \qquad (2)$$

Constant flow through the valve opening implies:

$$\frac{dI_R}{dt} = 0 \qquad (3)$$

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

$$R(t) = R_0 + \int \frac{1}{C}\,\frac{I_C}{I_R}\,dt \qquad (4)$$

wherein R_0 is the initial valve resistance. The same result can be achieved by a controller which minimizes the pressure U_Z after the valve:

$$\frac{dU_Z}{dt} = Z\,\frac{dI_R}{dt} = 0 \tag{5}$$

This simple consideration has two important consequences. First, by balancing the volume-valve combination, it is always possible to control the flow of any 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 can be optimized to obtain minimum back pressure to the engine, resulting in a higher engine efficiency.

2.2 Electrical equivalent model for the engine and the silencer

The simple model, presented in figure 2, 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 a duct is connected between the engine and the active silencer. A tail pipe is connected behind the silencer. The resulting circuit is presented in figure 3.

The left part of the circuit is the engine model. The four variable capacitors represent the four engine cylinders, who's volume varies sinusoidal between maximum and dead volume. 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 cam shaft operates the engine valves. The four short transmission lines behind the exhaust valve resistors represent the four exhaust conduits between the cylinder ports and the exhaust manifold junction. The intake side is connected to a voltage source U_B representing the atmospheric pressure and equals 100 kV. The combustion is simulated by charging the cylinder capacitor by a pulsing current source parallel over the capacitor. The charge time point corresponds to the ignition time point.

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 tail pipe and the resistor-inductor combination R_a and L_a corresponds to the spherical radiator impedance. In simulation, a collocated feedback controller conducts the control valve

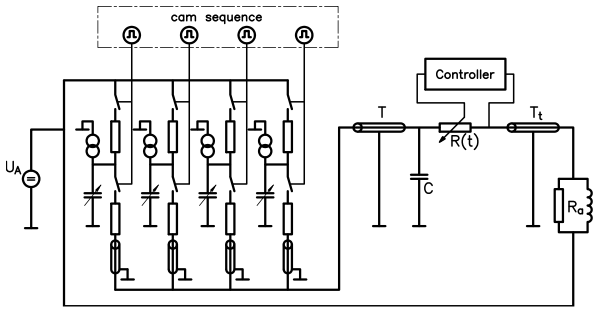

Figure 3: Electrical analog circuit for an internal combustion engine equipped with an active noise control valve.

resistance. This controller is only needed to determine the active silencer properties. In practice, other control strategies must be applied to handle the time delay between the valve action and its effect in the error sensor.

In principle, the back pressure to the engine can be set to any desired value by choosing the appropriate buffer volume capacitor C. In practice, it will result in a compromise between available space for the active silencer and acceptable back pressure.

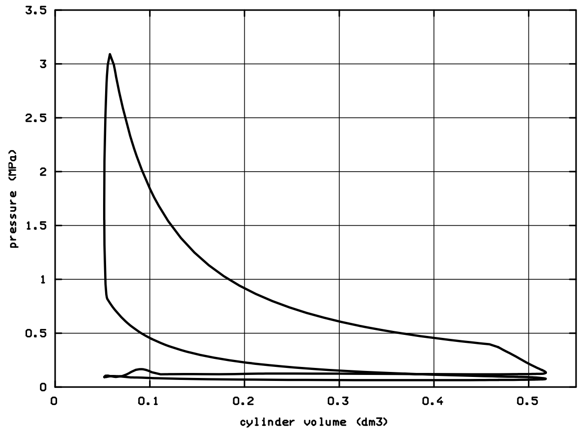

Figure 4: Simulated indicator diagram from the circuit presented in figure 3.

2.3 Simulation results

In the electrical circuit of figure 3, the engine parameters of a 2000 cm^3 engine are introduced. The ducts have a diameter of 60 mm. The duct between the engine and the silencer is 500 mm long, the tail pipe 700 mm long. The back pressure is set to 10 kPa, resulting in a buffer volume of 12 dm^3. The initial actuator valve resistance is set to 200 kΩ (1 Ω = 1 Pa s/m^3). The feedback control gain is set to 10^8.

In figure 4, the simulated engine indicator diagram

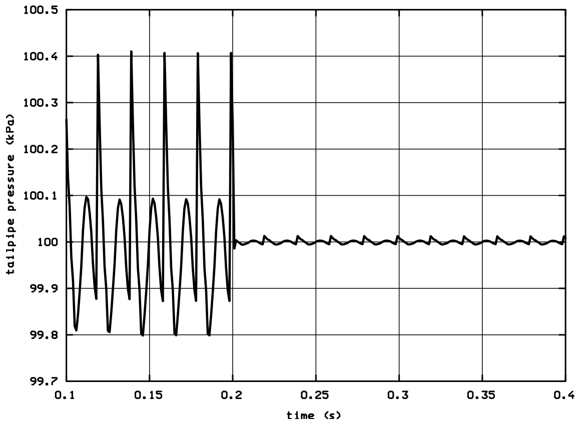

Figure 5: Simulated tail pipe absolute pressure. The controller starts at 0.2 s.

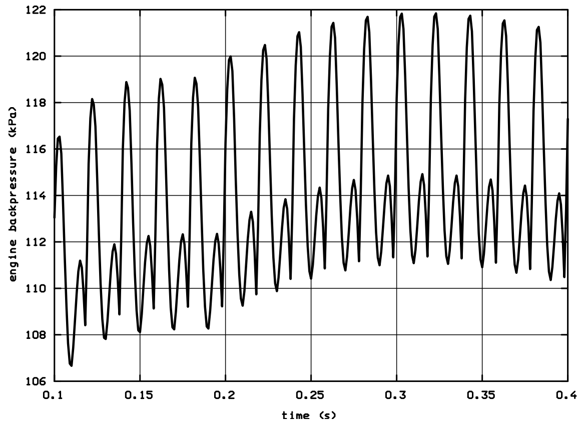

Figure 7: Simulated absolute back pressure to the engine. The controller starts at 0.2 s.

is presented. Actually, this diagram has no physical significance, it is an ”isothermal” simulation. Only the pressure at the exhaust valve opening time point must have its correct value, which is deterministic for the exhaust noise. The exhaust pressure in the tail pipe is displayed in figure 5. The controller is activated at 0.2 s. Activating the controller at 0.2 s does not effect the gas flow from the engine, as indicated in figure 6. The engine behaves as a volume velocity source. In figure 7, the absolute back pressure to the engine is displayed. After the transient switching on the controller, the back pressure returns to the presetted value of 10 kPa.

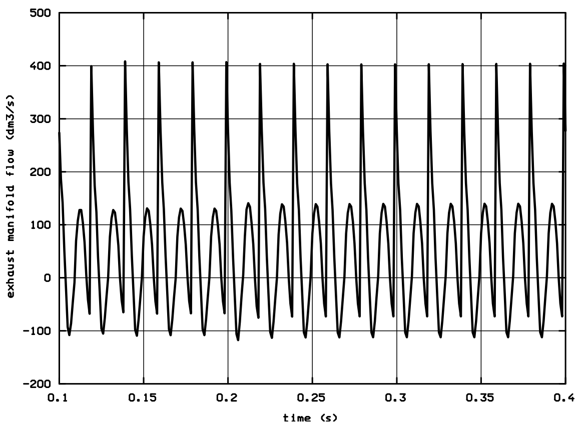

Figure 6: Simulated gas flow through the exhaust manifold.

2.4 Electrical analog model for the actuator valve.

The construction of the actuator valve is presented in figure 8. The valve has a conical head and is driven by a voice coil in a permanent magnetic circuit. The valve has a monotonic increasing valve resistance in terms of head displacement. This resistance is presented in figure 10. This is an advantage compared to for example a butterfly valve, which has a sign reversal in its resistance characteristic derivative in terms of position angle.

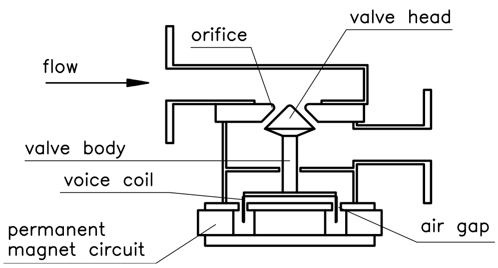

Figure 8: Scheme of the voice coil driven actuator valve.

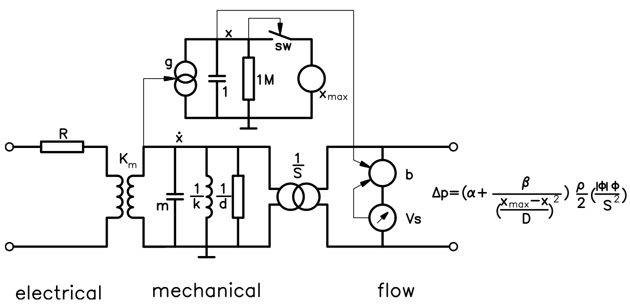

Figure 9: Detailed electrical analog circuit for the voice coil driven actuator valve presented in figure 10.

Figure 9 presents the electrical analog circuit of the actuator valve presented in figure 8. This circuit is typical for the construction of this valve. Another construction, for instance a butterfly valve, would result in a totally different electrical analog circuit.

The left side of the circuit consists of the voice coil electrical resistance R. The transformer K_m transforms the current into force by the voice coil in the permanent magnetic circuit. The force acts on the mass m of the moving valve body, i.e. the motor coil and the valve head. The valve body is suspended by a suspension spring k and is damped by the damper d. The damper can be constructed as a short circuit coil in the magnetic circuit. As mechanical "voltage", the valve head velocity $\dot{x}$ appears. The valve head flow resistance is dependent to the valve head displacement, consequently, the velocity has to be integrated. The integration is carried out by the controlled current source g, which current is proportional with the applied velocity. The integrated value appears as a voltage over the capacitor with value 1. The voltage source $x_{\max}$ corresponds to the distance between the zero position and the closed position of the valve. The zero position is the position where the suspension spring exerts zero force. By determination of the value $x_{\max}$, the pretensioning of the suspension spring can be taken into account. When the closing position is reached, the switch sw closes and the integration stops. The resistor of 1 M smoothes the current when the switch is activated, but its value is sufficiently high to not distort the integration. The pressure source b uses the displacement x and the volume velocity Φ measured in V_s to produce the pressure drop Δp [8]. The resulting pressure exercises a force on the valve head through the orifice area S. This force is fed back to the mechanical circuit through the gyrator $1/S$.

2.5 Simulation results.

To simulate the valve, the buffer volume capacitor has to be connected parallel over the input nodes at the flow-dynamic side. The engine as source will be replaced by a volume velocity source, to save computational effort. The volume velocity source is also connected in parallel over the flow input nodes. The source generates the flow corresponding to the engine exhaust, which is presented in figure 6. At the electrical nodes, a proportional feedback controller is connected. This controller generates an electrical current proportional to the alternating part of the flow V_s as error signal.

The values of the components of the actuator valve are: $R = 4\,\Omega$, $K_m = 16$ N/A, $m = 96 \cdot 10^{-3}$ kg, $k = 12 \cdot 10^3$ N/m, $d = 125$ Ns/m and $S = 916 \cdot 10^{-6}$ m^2.

Figure 11 presents the simulated gas flow through the tail pipe. The controller is activated at 2 s. The non-linear valve resistance does not influence the attenuation performance significantly. The valve head displacement is presented in figure 12. The distance between the head and the orifice is about 1.5 mm and the stroke in control amounts 0.4 mm. These are valuable data to design the voice coil and the magnet.

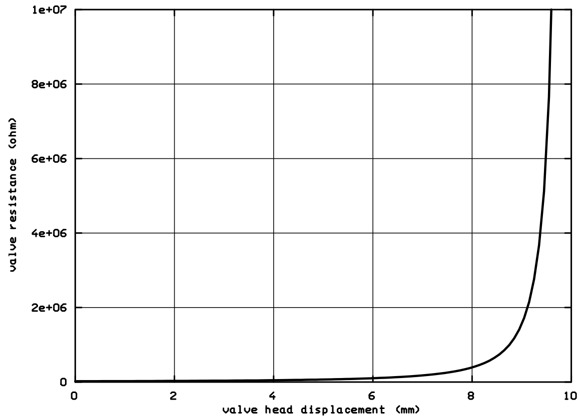

Figure 10: Actuator valve flow resistance in terms of valve head displacement for a conical head valve. The valve is closed at 10 mm.

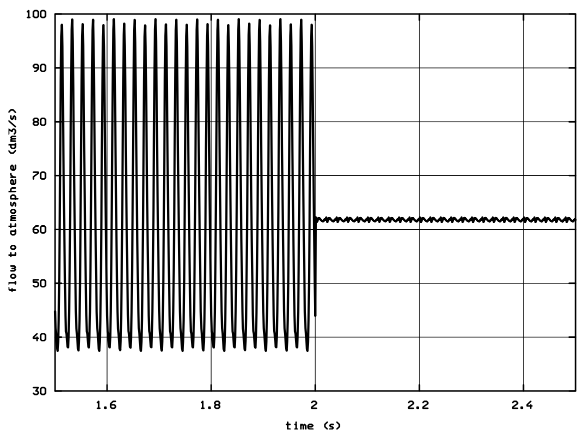

Figure 11: Attenuation of the gas flow pulsation through the tail pipe. The controller starts at 2 s.

The current through the voice coil is displayed in figure 13. The associated electrical power consumption of the valve amounts about 78 W to compensate the back pressure. The attenuation of the sound itself requires only 5 W power. If necessary, the constant force as consequence of the DC-current of 4.4 A can be compensated by placing a pretensioned spring parallel to the motor. This reduces the 78 W power consumption to theoretically zero.

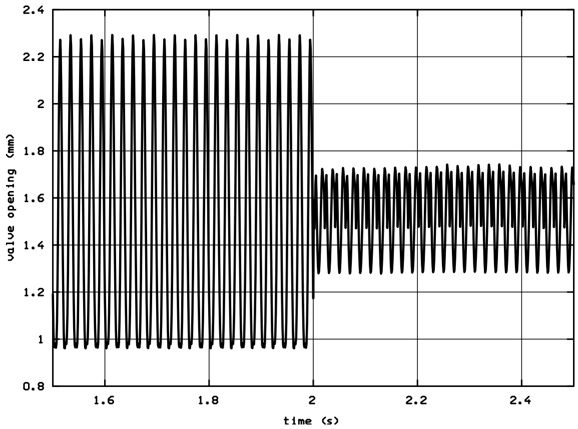

Figure 12: Actuator valve head displacement. The controller starts at 2 s.

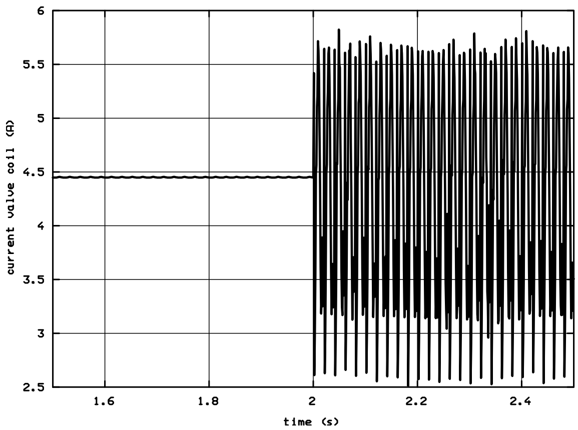

Figure 13: Current through the actuator valve voice coil. The controller starts at 2 s.

3 Experimental results

3.1 Cold engine simulator

The development of the active silencer has been carried out on a cold engine simulator. This device produces realistic exhaust noise using compressed air and permits to carry out acoustical and fluid-dynamic experiments with new concepts of exhaust systems without taking precautions against the hot and corrosive environment when directly testing on an operational internal combustion engine.

The exhaust noise of an engine is generated during the exhaust cycle. When the exhaust valve opens, the remaining pressure at the end of the expansion cycle discharges to the exhaust manifold pressure level. Then, the remaining gas is scavenged by the piston. The terms ”blowdown” and ”displacement” are used to denote these two phases of the exhaust cycle. The blowdown phase is typically very short in duration compared to the whole exhaust cycle and is responsible for the exhaust noise. Typically, the exhaust cycle starts 40^{o} to 60^{o} crank angle before the bottom dead center. Until about the bottom dead center, the gases are discharged due to the pressure difference between the cylinder and the exhaust manifold. The cylinder volume change due to the piston movement around the bottom dead center remains small, about 10% to 15%. Therefore, the blowdown pulse can be approximated as a discharge of a constant volume over the exhaust valve resistance. The displacement phase happens mainly at atmospheric level and ends when the exhaust valve closes at the top dead center. The noise generated during the displacement phase can be neglected compared to the blowdown phase. The cold engine simulator will generate only the blowdown pressure pulse.

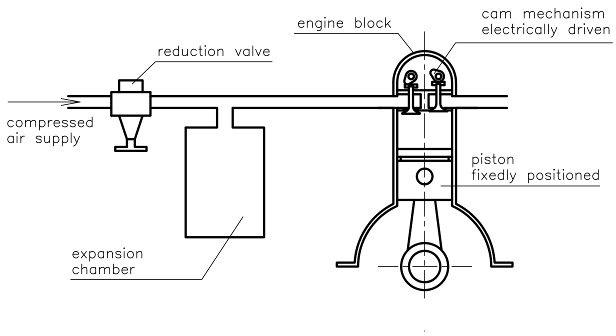

Figure 14: Working scheme of the cold engine simulator.

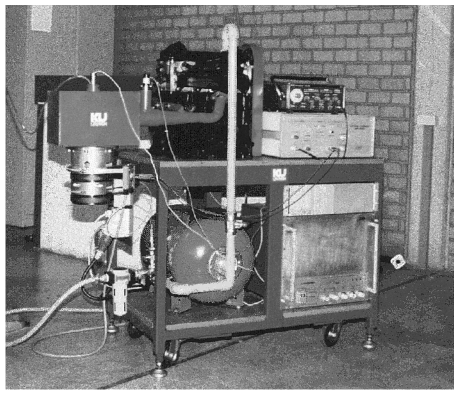

Figure 15: The cold engine simulator equipped with the active exhaust silencer.

3.2 Working principle.

The working scheme of the cold engine simulator is presented in figure 14. It consists of a regular engine block whose pistons are fixed at the bottom dead center. The intake collector is connected via an expansion vessel and a pressure reduction valve to a conventional pressurized air supply network. The supplied pressure corresponds to the remaining pressure at the end of the engine expansion cycle at the time point when the exhaust valve opens. The cam mechanism of the engine block is driven by an electric motor, which speed can be set on a frequency converter. During the inlet stroke, the cylinder charges at the intake manifold pressure level. When the outlet valve opens, the cylinder discharges and the pressure pulse enters the exhaust. This discharge corresponds to the blowdown phase of an engine exhaust cycle. These pressure pulses from the simulator exhibit a similar behaviour as these of a regular combustion engine. The presented cold engine simulator in the research work has been built using a Volkwagen 1600 cm^3 engine.

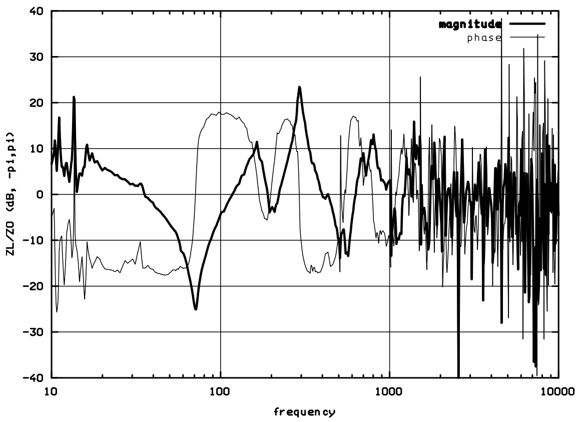

Figure 16: Exhaust acoustical impedances of the cold engine simulator.

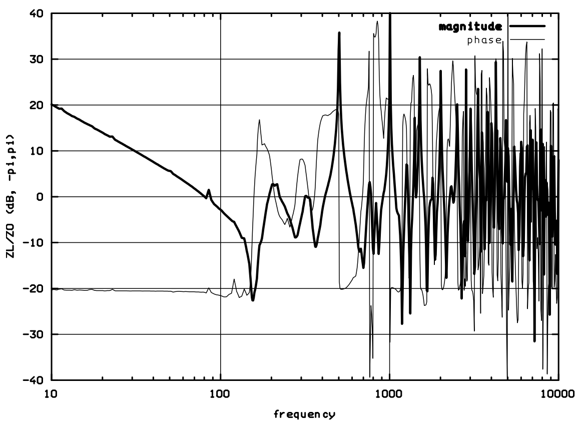

Figure 17: Exhaust acoustical impedances of the operational combustion engine.

To prove that the acoustical characteristics of the cold engine simulator are equivalent to these of a regular engine, their acoustical impedance and source spectrum are compared. At the same time, the electrical analog circuits will also be validated by determining their acoustical impedance and source spectrum.

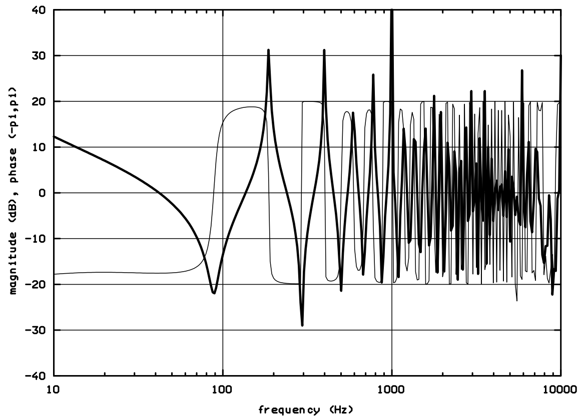

Figure 18: Exhaust acoustical impedances simulated from the engine part of the electrical analog circuit in figure 3 (linearized).

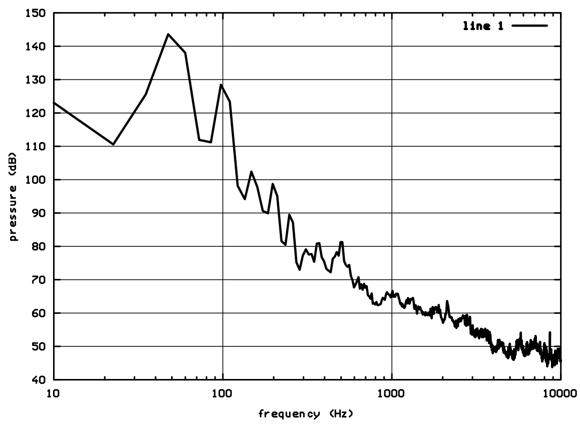

Figure 19: Exhaust acoustical source spectrum of the cold engine simulator.

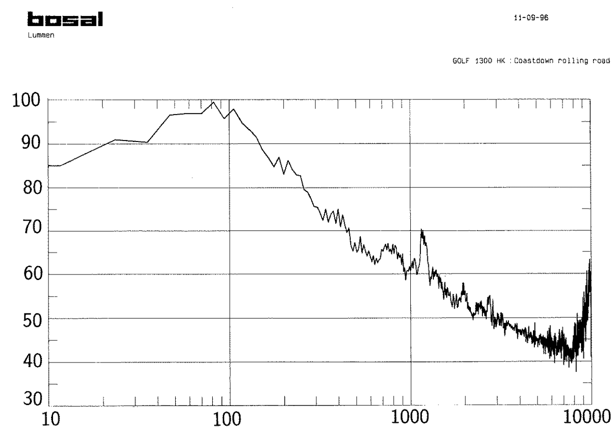

Figure 20: Exhaust acoustical source spectrum of an operational combustion engine.

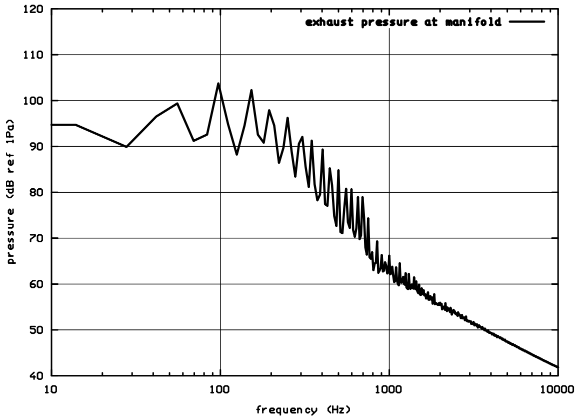

Figure 21: Exhaust acoustical source spectrum of the simulation of the engine part of the electrical analog circuit in figure 3.

3.3 Exhaust acoustical impedance.

Figures 17 and 16 present the acoustical impedances at the engine exhaust manifold. The acoustical impedances are measured according to the ISO/FDIS 10543-2 procedure [9]. The method uses the transfer function between two microphones positioned in a waveguide connected to the engine exhaust. From this transfer function, the acoustic impedance is calculated after deriving the reflection coefficient from the measured transfer function. To prevent the engine to disturb the reflection coefficient during the measurements, the engine is run with an electric motor and the intake is sealed.

Figure 16 presents the acoustical impedance of the cold engine simulator, running at 1000 rpm. The reference impedance (0 dB) corresponds to the connected waveguide characteristic impedance $Z_0 = 347$ kΩ. The center measurement presents the acoustical impedance of an 747 cm^3 renault engine. In figure 18, the simulated acoustical impedance of the engine part in the linearized electrical circuit of figure 3 is presented. The linearization implies: (1) the switches are not operated, i.e. three switches are always open and one is closed; (2) the cylinders capacitors are fixed to the volume with the piston in the middle position.

The three impedance curves are similar. The -20 dB/decade line below the first antiresonance is determined by the cylinder and the manifold volume. The antiresonance itself is the Helmholtz resonance between the cylinder volume and the air mass in the manifold. All the subsequent resonances are internal manifold resonances. The phase of the acoustical impedance ranges between -90^o and $+90^o$, i.e. the impedance ranges between capacitive and inductive.

3.4 Exhaust source spectrum.

Figure 19 and 20 present the sound pressure spectra measured inside the duct behind the manifold junction on the cold engine simulator and the operational combustion engine respectively. Figure 21 presents the sound pressure spectrum simulated at the manifold junction of the electrical analog circuit presented in figure 3. The absolute level is dependent on the cylinder pressure at the exhaust valve opening time point and the cylinder volume, the exhaust valve resistance and the connected impedance. The pressure level decays -20 dB/decade at high frequencies.

Concluding, the cold engine simulator exhibits the same behaviour as a real engine with regard to acoustic phenomena.

3.5 Active silencer.

An active silencer, based on the actuator valve developed during the previous section, has been built and tested. A photograph of the cold engine simulator equipped with the active silencer is shown in figure 14.

Two types of controllers are tested, namely a x-LMS adaptive feedforward controller and a feedback controller. The feedforward controller was commercially available at Digisonix. The feedback controller was developed during the research project and implemented in analog electronic hardware. Reference [10] provides more information how the feedback controller is developed. The bandwidth of the feedback controller is about 100 Hz. Both control algorithms have similar performance. The presented results are obtained using feedback control.

3.6 Attenuating performance.

Figure 22 presents the tail pipe pressure during activating the controller. The cylinder pressure at the exhaust opening time point equals 400 kPa. The controller is activated at 0.6 s. The low frequency pulsation is removed from the exhaust noise. The tail pipe pressure spectrum with controller off and on is presented in figure 23. The silencer is capable to attenuate very low frequency noise, which is extremely difficult using loudspeaker systems. The pulsation frequency is 10 Hz (300 rpm). The SPL reduction between control on an off amounts 13 dBL and after A-weighting 4 dBA.

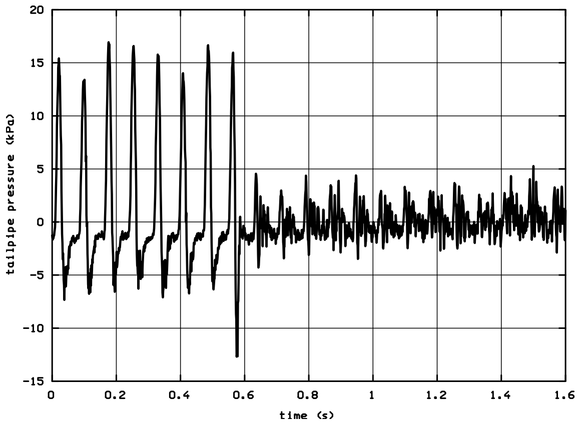

Figure 22: Tail pipe pressure during time. The feedback controller is activated at 0.6 s.

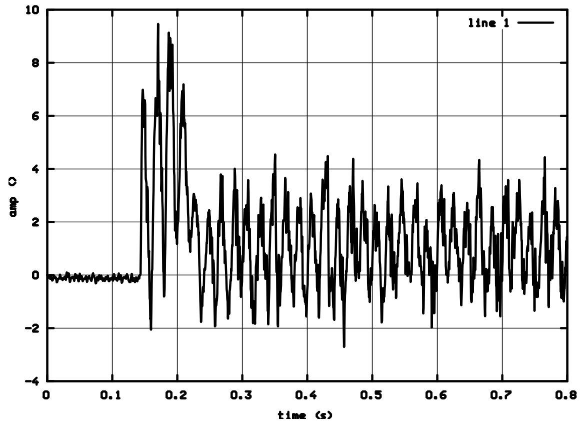

Figure 24: Current through the actuator valve voice coil. The controller starts at 0.15 s.

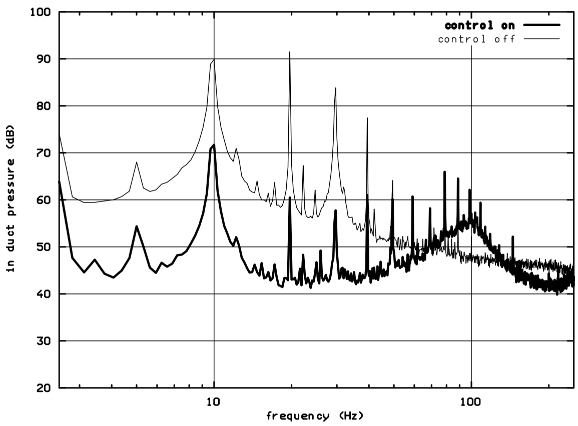

Figure 23: Acoustic tail pipe pressure spectrum with feedback control on and off.

3.7 Electrical power consumption.

In figure 24, the measurement of the current through the voice coil of the actuator valve while activating the controller is displayed. The controller starts at 0.15 s. The voice coil resistance is $3.6\ \Omega$. The associated power consumption of the valve to attenuate the exhaust noise amounts about 4.5 W, which is consistent with the simulated power.

3.8 Back pressure to the engine

In figure 25, the mean back pressure to the engine exhaust is displayed. The back pressure is measured using a piezo-resistive sensor and is filtered through a 1 Hz second order low pass filter. The back pressure is decreased from 17 kPa to 2 kPa during the measurement by decreasing the DC-current through the actuator valve voice coil. Meanwhile, the acoustic pressure level in the tail pipe is monitored.

As equation (4) states, the sound attenuating performance is independent of the back pressure, as long the initial resistance R_0 remains high enough such the total resistance $R(t)$ does not become negative. By decreasing the DC–current through the actuator valve voice coil, the initial resistance decreases. As soon the back pressure sinks below 10 kPa, $R(t)$ should become negative, which is physically not possible. As result, the noise attenuation capability decreases dramatically. The back pressure value of 10 kPa corresponds the value whereupon the silencer was developed. In principle, the back pressure can be minimized for each engine operation point by an additional controller to obtain maximum engine efficiency.

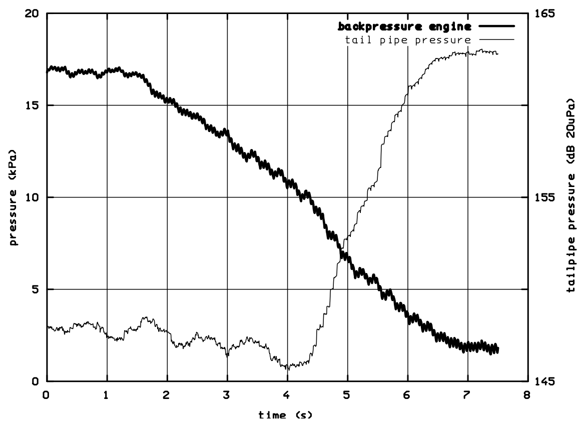

Figure 25: Tail pipe rms-pressure while the engine back pressure sinks from 17 kPa until 2 kPa.

4 Conclusion

An active silencer for internal combustion engine exhaust noise is developed, using an actuator valve in conjunction with a buffer volume, capable to attenuate the low frequency components of the exhaust noise without preconnected passive silencers.

This research project leads to the following conclusions:

1. Any exhaust noise from a reciprocating engine can be attenuated by balancing the buffer volume and the engine back pressure. The back pressure can be chosen and then the buffer volume can be determined (or vice versa).
2. An electrical analog circuit is proposed to simulate the active silencer on a combustion engine. The acoustic impedance and the source spectrum of the modeled engine are similar to these of a real engine.
3. The actuator valve itself is simulated using an electrical analog circuit, wherein the electrical, mechanical and fluid-dynamic properties are modeled in detail. From these results, a prototype actuator valve can be constructed.
4. A cold engine simulator is developed which generates realistic exhaust noise using compressed air. It allows acoustic and fluid-dynamic experiments on new concepts of exhaust systems, without taking precautions against hot corrosive gases. The exhaust acoustic impedance and the source spectrum are similar to a real engine.
5. The active silencer has been built and tested on the cold engine simulator. The noise attenuation amounts 13 dBL (4 dBA), consuming about 5 W electrical power and causing 10 kPa back pressure to the engine.

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