Design and implementation of an exhaust system for a Citroën 2CV race car

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Abstract

This paper gives an overview of the design and production methods for a custom-made exhaust system for a Citroën 2CV race car. The goal of this new design is to improve the performance, especially the torque figures, at the critical shift points while maintaining a limited noise production. The design phase is done with simulations using ngspice software, where physical entities are represented by electrical ones. This electrical analogy included the exhaust tubes with variable lengths and the entire rotating engine with its intake manifold, combustion, crankshaft and camshaft profiles. After the simulations are done, the results are converted back from electrical to physical entities. The design resulted in a 30% increase in engine torque in the critical region where the engine operates in racing conditions. Equally important as the torque figures was the exhaust's noise production, which had to be limited to 95 decibels.

1 Introduction

The main purpose of the research stated in this paper is to improve the torque of the Citroën 2CV Race Car of the CQS Group Racing Team KU Leuven without producing more than 95dBA of noise. The aim is to increase performance of the race car during racing conditions, while retaining the maximum sound emission allowed by the circuit authorities by redesigning the entire exhaust system. This paper thus provides a solution for designing an exhaust system and simulating the gas flow and pressure waves in this system. It also offers a production method used to implement the obtained design [1].

1.1 Nature and scope of the problem

The race car, Pegasus, of CQS Group Racing Team is a Citroën 2CV, modified for participation in the Belgian 2CV racing cup [2]. The engine in this race car is a 652cc two cylinder boxer engine. Figure 1 demonstrates the engine torque as a function of its rotational speed. When the driver shifts to a higher gear at ± 7500 rotations per minute (RPM), the engine's rotational speed drops to 5000 RPM. The curve shows a torque decrease in this region which is highly undesirable for the car's acceleration (the red full line). Through dyno tests with several different engines but with the same intake and exhaust system as that of Pegasus, this shape of torque curve was obtained every time. Therefore the intake and exhaust system will be analyzed.

In order to maximize the acceleration potential of the race car, the engine's torque should be as high as possible, but even more desirable than this absolute maximum, is a flat torque curve. This is the theoretical optimum [3]. A practical solution for an approximation of this flat curve is the cancellation of the torque decrease to obtain a constant slope in the torque curve. This is illustrated by the blue dashed line in Figure 1. A new exhaust system will be designed in order to gain torque in the desired RPM-region and come closer to the optimal flat torque curve. Furthermore, the intake system will be analyzed to determine its influence on the torque curve.

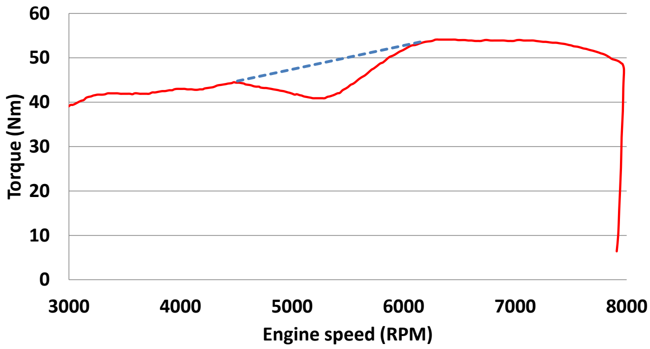

Figure 1: torque curve

1.2 Analysis of pressure and flow phenomena

The main supporting research, upon which this paper relies, is a PhD research using similar conversion of physical to electrical entities and an electrical engine equivalent [4]. The performance and noise emission levels depend on the flow characteristics and pressure levels in the exhaust system. First of all, the system has to be able to get all exhaust gasses out of the cylinder during the time interval when the exhaust valve is open. When all the exhaust gasses have been evacuated from the cylinder, flow resistance in the exhaust system should be minimal in order to guaranty a fluent gas flow. Exhaust gasses produce standing waves which can either aid or counteract the cylinder evacuation [5].

The gas flow and the pressure in the system are inseparable from each other. In order to evacuate as much exhaust gas from the cylinder, the pressure right behind the exhaust valve should be as low as possible. If the pressure behind the open exhaust valve exceeds the pressure in the cylinder, gas will flow back into the cylinder. This prevents fresh air-fuel mixture from entering the cylinder and thus limits efficiency [3].

The same relation goes for the pressure waves that arise when the exhaust valve is opened and an immediate pressure drop in the cylinder occurs. This pressure wave flows through the exhaust system at a speed equal to the sum of the gas flow speed and the sonic speed at exhaust gas temperature [6]. Where the tube suddenly increases in diameter, a fraction of the pressure wave reflects backwards inducing a pressure at the exhaust valve. This either aids the evacuation of exhaust gas or forces exhaust gas back into the cylinder, thus influencing the performance. The influence of this area increase depends on the relative value of the large area against the small area: a relative value between 0 and 1 results in a reflection fraction between 0 and 50% [5]. Whether the timing of the arrival of the pressure wave is ideal depends on the length of the entire exhaust system, the gas flow rate and the engine's rotational speed [5]. The RPM value for which a tube length has a positive effect on the exhaust gas evacuation is called the working point of that exhaust system.

2 Design

The design choice of the exhaust system is based on several criteria. First of all, there is the performance which has priority. Secondly, there is the noise emission which should not exceed 95dBA, measured at a distance of 1m at an angle of 45°. This criterion is strongly connected to the main criterion because of the fact that both compensate for each other. An exhaust system that offers optimal performance gives high noise emission results while a quiet exhaust system results in poor performance figures [5]. Thirdly, the exhaust system has to fit in the available space in the engine bay and under the side skirt. Lastly, the entire exhaust system should be as light as possible because of the fact that all additional weight is undesirable on a race car. These last two criteria are also connected because the limited space under the side skirts disallows the use of multiple larger silencers which add a lot of mass to the total exhaust system.

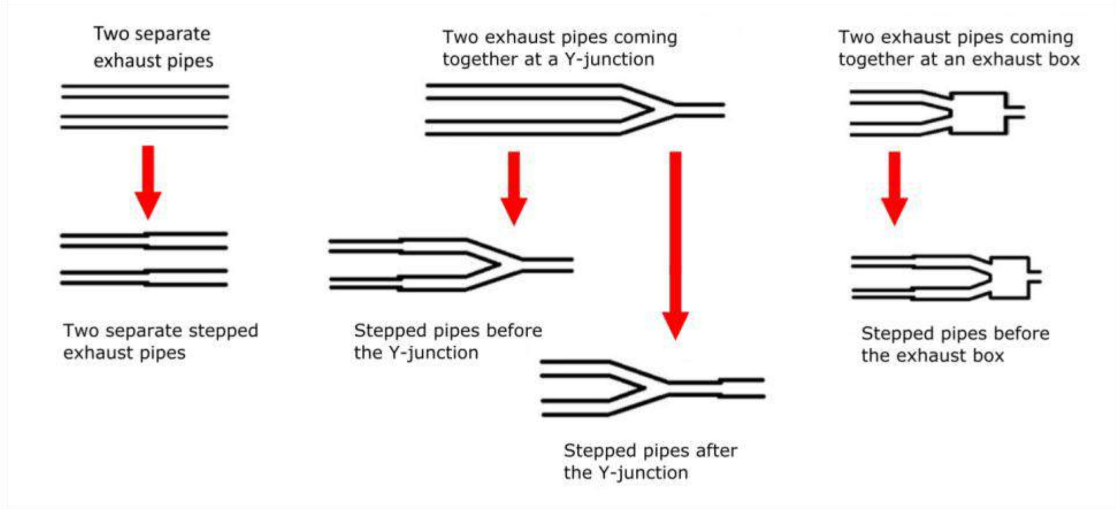

Figure 2: designs for the exhaust system

In Figure 2 the considered designs are shown. The research for an ideal compromise for these criteria began with the determination of four basic design principles for the two cylinder boxer engine: two separate exhaust tubes, two exhaust tubes coming together at a Y-junction, two exhaust tubes coming together at an exhaust box and the use of stepped tubes. These stepped tubes can be used at the primary tubes in combination with all three other design principles and were also combined with the secondary tube after the Y-junction. The primary and secondary parts of the exhaust systems are the parts before and after the confluence of tubes respectively.

The system with two separate exhaust tubes is very easy to produce and thus cost effective but each tube will need its own silencer which adds extra mass to the system. The second design incorporates a Y-junction. This design is the only design where the pressure waves and gas flows of the primary tubes can influence each other. Therefore, the position of the Y-junction is of major importance. One has to make sure that the position does not only generate a positive effect on the desired RPM value, but also does not generate negative effects on other meaningful RPM values [5]. This design is also the lightest and most compact due to the fact that only one silencer is needed, but it is more expensive to produce. The last exhaust system design consists of two primary exhaust tubes coming together in an exhaust box. This exhaust box ensures there are no interference effects between the two tubes by offering a buffer volume [5]. This means that the primary tubes act like two separate exhaust tubes. This architecture has already a silencing effect, but large space is required for this exhaust box and extra mass is added. All previously mentioned designs can be equipped with stepped tubes. These are used to add a second working point to the total system. All data necessary to simulate the engine and exhaust system were calculated using thermodynamics (Otto-cycle) and checked experimentally.

3 Development

3.1 Construction of the electrical scheme of the engine

To simulate the working of the engine and the pressure phenomena in the exhaust, the software ngspice is used. This is free electrical circuit simulation software and is easy to use. Physical entities are converted to electrical entities, based on the use of SI units. This way one pascal of pressure is set to be one volt in the electrical domain. Using SI units, physical entities can easily be converted to their electrical analogies.

Table 1 provides an overview of the used entities [4]. Parameter c is the speed of sound, dependent on temperature, l is the length of the tube, ρ is the gas density, S is the cross section of the tubes and R_a stands for the resistance of environmental gasses at the exhaust tip.

Physical entity	Electrical entity	Electrical formula
Pressure source	Voltage source	$V = p$
Tube	Transmission line	Z,C,L as described below
Valve	Switch	-
Flow resistance	Electrical resistance	$R = \frac{\rho \cdot c}{S}$
Volume	Capacitor	$C = \frac{V}{\rho \cdot c^2}$
Gas mass	Inductance	$L = \frac{\rho \cdot l}{S}$
Environmental gas mass at exhaust tip	Inductance	$L = \frac{r \cdot R_a}{c}$

Table 1: conversion from physical to electrical entities

To calculate the resistances, ρ and c are considered at room temperature for the intake, at 850K for primary exhaust tubes and at 615K for secondary tubes. The volume V in the formula of the capacitor is the volume of the cylinder with the piston at the considered height. In the simple scheme this volume is taken with the piston in bottom dead center (BDC). When the capacitor will vary in time, like explained further on, the capacitance will vary between the values which represent TDC and BDC, the top and bottom dead center respectively.

Figure 3 shows the elementary electrical equivalent of the two cylinder engine. This scheme is used as a simplification: phenomena as compression, combustion and intake are neglected. The scheme works as follows: in each of the two branches a capacitor is shown. This represents the cylinder volume. The switches left and right of the capacitor model the intake and exhaust valves. Because flow through a valve undergoes resistance, two resistances are placed in series with the switches. The switches are controlled by pulse sources which represent the camshaft with its exact timing. The voltage source at the left of the circuit models the gas pressure at the crankshaft angle where the exhaust valve opens, set at 2.98 bar as calculated in equation 9. This pressure is equal to 298000 Pa and therefore, the source voltage is 298 kV DC.

The resistors are calculated using the time constant τ = RC, in order to ensure proper filling of the capacitor. The capacitance is well known, and RC is considered to be 2 ms, the time it takes to fill the cylinder. This implies that the resistors have a value of 1 MΩ.

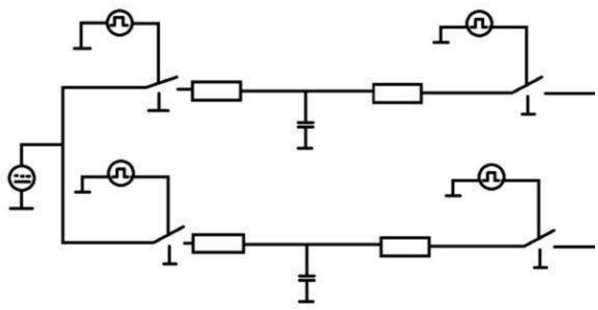

Figure 3: elementary equivalent scheme of the engine

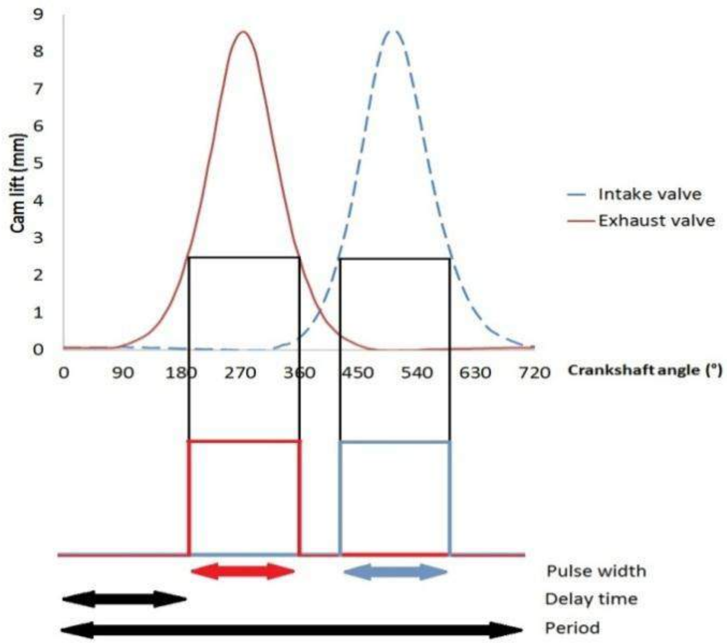

Figure 4: cam profile with pulse train parameters

In Figure 4, one can see how the derivation of the timing of the pulse trains that control the switches comes forth from the camshaft measurement. This is used to simulate an exhaust setup over the entire range of engine speeds: from 2500 RPM to 7800 RPM.

To make the engine model more accurate, compression, combustion and the intake system and intake stroke are introduced to the model. This more accurate scheme can be found in Figure 5. On the car Pegasus, the intakes are two separate short tubes with air filters. These are implemented by two transmission lines between the intake valves and the voltage source that now represents atmospheric pressure: 101kPa or 101kV. To simulate compression and expansion, the capacitor varies its capacitance sinusoidally in time, with respect to the previously determined camshaft timing represented by the pulse trains. Therefore, similar pulse train parameters are calculated from the camshaft profile: amplitude, frequency and offset. This offset is important to represent the crankshaft – camshaft timing.

Finally, combustion is added to the model. Physically, combustion means a quick pressure raise in the cylinder. This is modeled by a current source that injects a large current in a short pulse into the capacitor. This raises the voltage across the capacitor which simulates a pressure raise in the cylinder.

In Figure 6, the timing of all moving engine parts is shown. The exhaust valve is represented by the red curve (widely dashed), the intake valve by the blue curve (pointed), the crankshaft is shown by the sinusoidal green curve (narrowly dashed), and injection is shown as the pulsating purple curve (full).

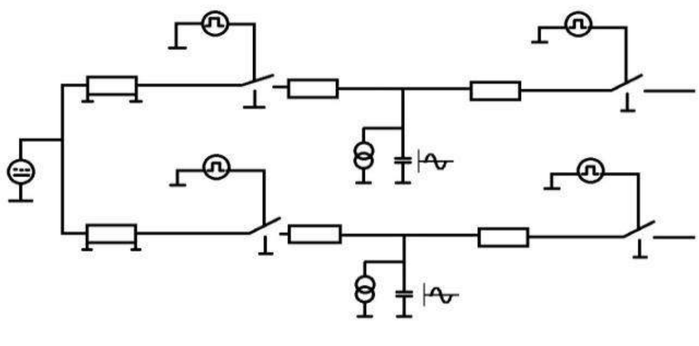

Figure 5: more elaborated equivalent scheme of the engine

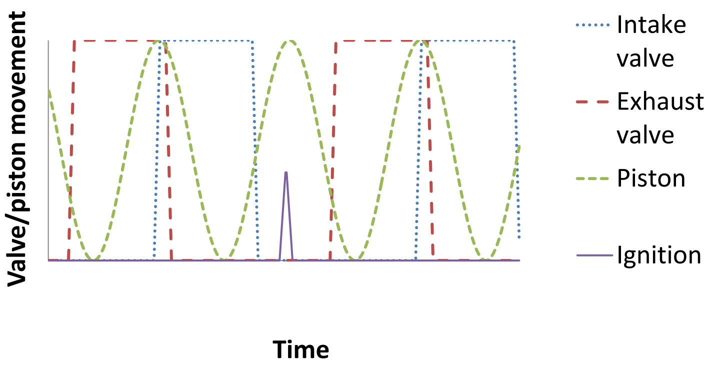

Figure 6: valve and ignition timing with respect to the crankshaft

3.2 Construction of the electrical scheme of the exhaust and intake

The different exhaust setups, explained in section 2, are introduced to the electrical scheme as a set of transmission lines. This can be seen in Figure 7. The introduction of a transmission line in the electrical scheme requires a set of parameters: its resistance to gas flow, R; its inductance, L; its capacitance, C and its length l. Since R, C and L are depending on the cross section of the pipes, a simple set of transmission lines can represent every possible combination of diameters, volumes and junctions. The R, L and C values of a transmission line, representing a tube, are calculated according to the formulas from Table 1. Similarly to the exhaust, the intake system is represented by two transmission lines with according R, C and L parameters.

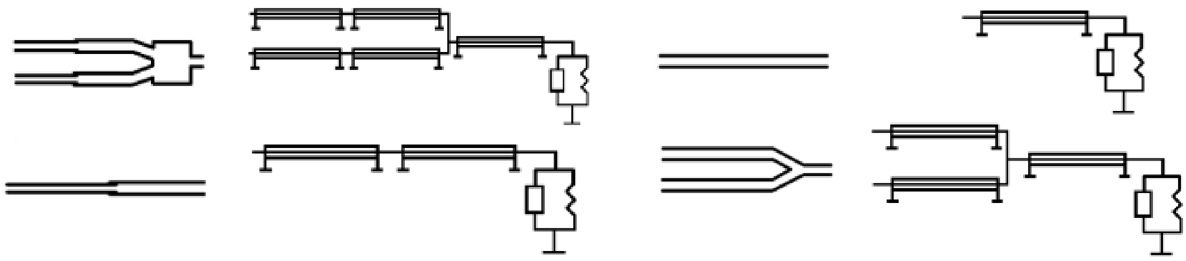

Figure 7: exhaust designs with electrical equivalents

3.3 Noise analysis

Not only the performance depends on the pressures, but the noise emission is also affected by these phenomena. When a pressure wave gets at the end of an open tube, a part of it reflects back to the source, but the other part gets emitted into the open air. The ways in which the pressure waves propagate in the open air depend on their frequencies. At high frequencies the pressure wave will propagate linearly, while at low frequencies the pressure waves will propagate spherically. Depending on the frequency and the amplitude the total sound intensity can be calculated and evaluated [7].

In order to calculate the noise production of the exhaust, the sound pressure needed to be calculated from the simulated pressure at the exhaust tip through

$$L_0 = 10 \cdot \log\left(\frac{{p_{RMS}}^2}{{p_0}^2}\right) \tag{1}$$

where p_{RMS} is the simulated exhaust pressure and p_0 is the reference pressure, which physically means the minimal pressure difference that the human ear can perceive, equal to 0.00002 Pa. This sound pressure L_0 is the sound at the exhaust tip, at a distance of one radius from the tube tip.

To simulate the emitted noise more accurately, a band pass filter is applied on the exhaust noise signal. This band pass filter represents the human perception of noise. The human ear can perceive frequencies in the range from 20 Hz to 20 kHz. Frequencies lower than 500 Hz are attenuated by -20 dB per decade, as are frequencies higher than 4 kHz. This simulates the human perception of sound and is illustrated in Figure 8. How the band pass filter is implemented electrically, is shown in Figure 9.

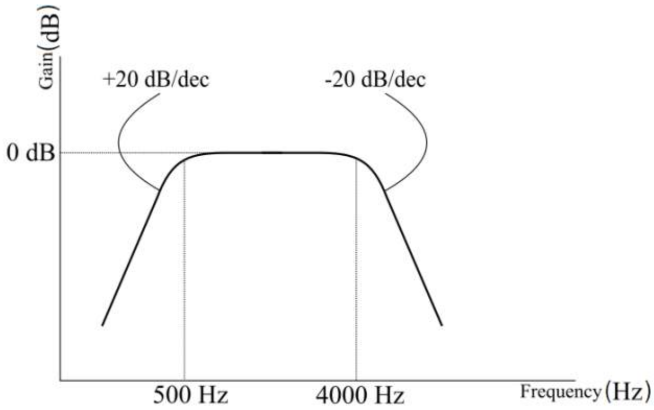

Figure 8: pass band filter to approach the human sound perception

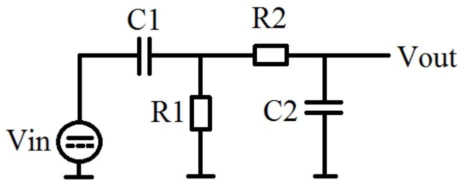

Figure 9: electrical band pass filter

4 Simulations

After completing the equivalent electric circuit that represents an exhaust system attached to the engine, transient simulations are conducted to obtain the pressure phenomena. Regions of interest are the pressure in the cylinder (the voltage across the capacitor), the pressure at the intake valve (the voltage right before the intake switch), the pressure behind the exhaust valve (the voltage right after the exhaust switch) and the pressure at the exhaust tip (the voltage right behind the last transmission line).

4.1 Design testing

In the first stage of the simulations, a simple tube was represented by one transmission line, connected to only one cylinder. The engine circuit was the most basic one, represented in Figure 3. A set of simulations is done using a tube length varying between 1.2 m and 2.5 m at an engine speed of 5400 RPM. The length with the best result i.e. with the lowest pressure in the cylinder, of the single tube exhaust is shown in Figure 10. The red step function (pointed line) represents the opened exhaust valve (high) and closed valve (low). This valve movement is simplified by a step function. The blue curve (widely dashed line) represents the pressure in the cylinder. During valve opening, the cylinder pressure drops. How low this pressure drops depends on the pressure in the exhaust manifold, right behind the exhaust valve. This pressure is represented by the green curve (narrowly dashed line) and strongly oscillates in the exhaust tube. The oscillation is caused by resonance of the reflecting pressure waves and the regular opening of the exhaust valve. This highly resonating exhaust delivers a good cylinder evacuation which results in good performances. Because of the relationship between noise and pressure, high pressure resonance causes the exhaust to emit more noise. The purple line (full line) represents the gas pressure at the exhaust tip resulting in the produced noise: 130 dBA without silencer.

The design with the Y-junction was the first that was simulated having different primary and secondary tube lengths. This type of exhaust proved to have two ideal tube lengths: one for the primary tube (having an ideal length of 1.58 m) and one for the total length of the exhaust system (ideally 2.26 m). Pressure waves reflect both at the Y-junction and at the end tip because of an increasing flow area and this is why there are two optimal lengths which delivered a good timing of the reflected waves to arrive at the exhaust valve right before closing. This can be seen in the graph in Figure 11. The red curve (dotted line) represents again the exhaust valve. The blue curve (widely dashed line) represents the pressure in the cylinder, which becomes – theoretically – negative, caused by low pressures (green curve or narrowly dashed line) arriving at the exhaust valve and therefore improving the suction effect of the exhaust. The purple curve (full line) represents the pressure at the exhaust tip. This particular setup emits a sound level of 129 dBA without silencer.

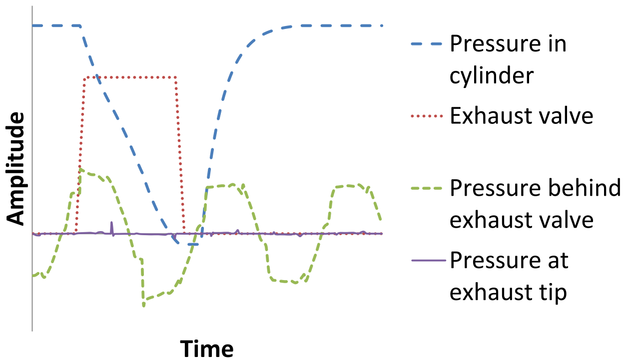

Figure 10: pressure phenomena at a simple tube

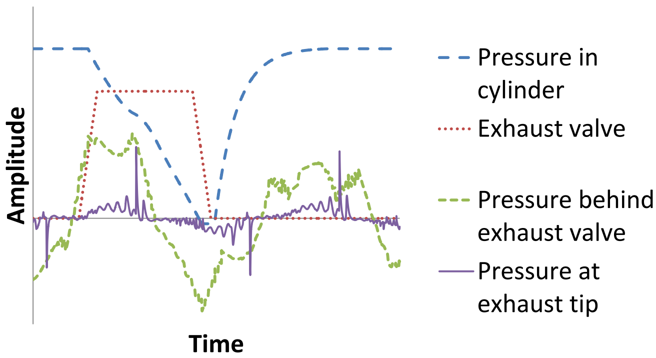

Figure 11: pressure phenomena with a Y-junction

The following exhaust types that were tested were the exhaust box with stepped tubes, two separate stepped tubes and the Y-junction with stepped tubes before and after the junction. All of these types had their well working tube lengths, but none of them performed better than the simple Y-junction. The purpose of an exhaust box is its neutralizing effect regarding wave reflections. Compared to the cylinder pressure (blue curve or widely dashed line) of the single tube, the lowest cylinder pressure at the moment the exhaust valve closes is higher. The neutralizing effect of the exhaust box can be seen in the green curve (narrowly dashed line) in Figure 13, which represents the pressure behind the exhaust valve. The amplitude of this pressure is twice as low compared to the pressure in the Y-junction design. Because pressure waves are attenuated by the exhaust box, this design emits less noise than the Y-junction design: 117 dBA without silencer.

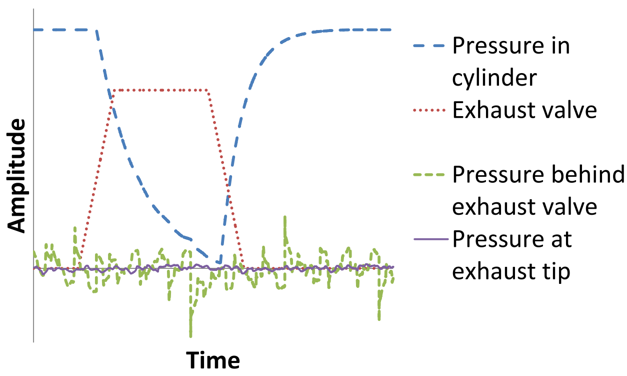

Figure 12: pressure phenomena with an exhaust box

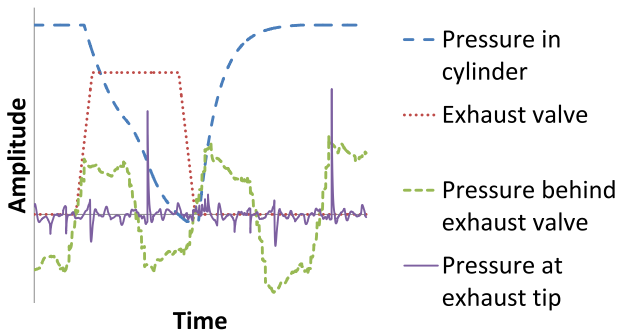

Figure 13: pressure phenomena with a stepped tube behind the Y-junction

The exhaust design using two separate stepped tubes gave similar results compared to the exhaust box design in terms of pressure remaining in the cylinder. However, because the diameter increase was too small to result in a second working point, the manifested reflections came from the end of the tube rather than from the area increase. When the area increase was bigger, the implementation space limit was exceeded and the design could not be implemented in the car. Therefore this design was not considered an option.

The Y-junction with stepped tubes appeared to have very few influence by delivering a very similar plot compared to the one of Figure 11 (the Y-junction exhaust): the reflection of pressure waves at the Y-junction caused the cylinder to evacuate decently while the dilatation delivered no better results. This plot is shown in Figure 13. During the simulations, a tube area increment of 25% was used, because this was realistic for the implementation afterwards. In order to fully use the reflecting effect, the tube area increment had to be larger than 25%, but this resulted in a bigger and heavier exhaust, and making the exhaust impossible to implement unless major modifications to the vehicle's bodywork were made. Therefore, the simple Y-junction design was chosen for further simulations with a more detailed engine representation.

4.2 Analysis for the entire engine speed range

The design for the exhaust with the simple Y-junction, proved to be efficient at an engine speed of 5400 RPM. Because a race engine runs through a broad range of speeds, this Y-design was tested over this full range: from 2500 to 7800 RPM. While doing so, the current problem, as stated in the introduction, where exhaust gasses are being pushed back into the cylinder, became apparent: at engine speeds right above 5400 RPM, the exhaust generated a pressure at the exhaust valve which was a lot higher than the pressure inside the cylinder. This was proven in the transient analyses where a large voltage behind the exhaust switch was generated at the moment that the exhaust switch opens (valve closes). During the simulations across the entire range of engine speeds, the remaining cylinder pressure at exhaust valve closing time was plotted as a function of engine speed. This resulted in Figure 14.

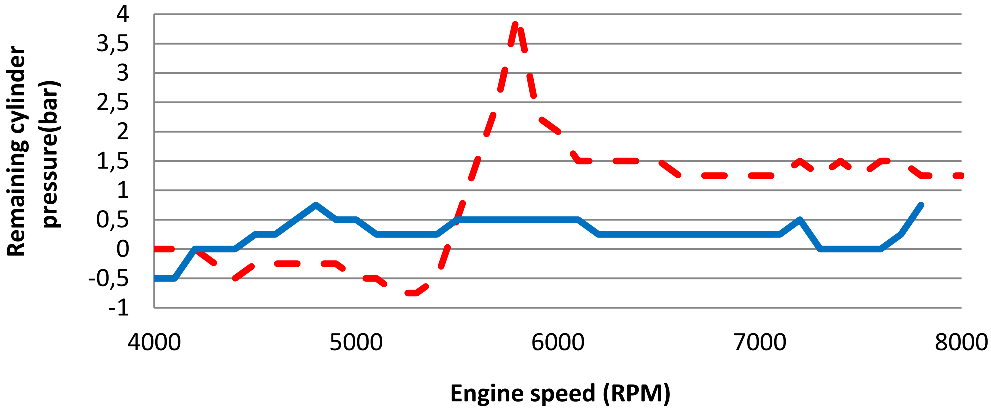

Figure 14: cylinder pressure at exhaust valve closing time as a function of engine speed

The red dashed curve represents the remaining pressure inside the cylinder at the moment the exhaust valve closes. From this graph, it appears that the exhaust delivers a positive effect below our target point of 5400 RPM, and performs worse behind this point. Therefore a new target point was taken at 7800 RPM, which delivered the blue full line from simulations across the engine speed range. Therefore, 7800 RPM became the new target point for future simulations.

4.3 Intake introduction to the system

After determining that the Y-junction design would be used to fine-tune the entire exhaust system, the intake system was evaluated. Firstly, the original intake system was simulated at the critical RPM value of 5400 RPM. The result of these simulations can be seen in Figure 15. There are two main phenomena visible: the actual suction of air into the cylinder and pressure wave reflections. As soon as the intake valve opens, the pressure at the intake valve drops because air flows from the intake system into the cylinder. The pressure in the cylinder rises as more air and fuel enter the combustion chamber. Right before the intake valve closes, pressure at the valve decreases, eventually reaching the same pressure value as in the cylinder. At this point, just below atmospheric pressure, the maximum pressure in the cylinder is reached. When the intake valve is closed, pressure oscillations are visible at the intake valve.

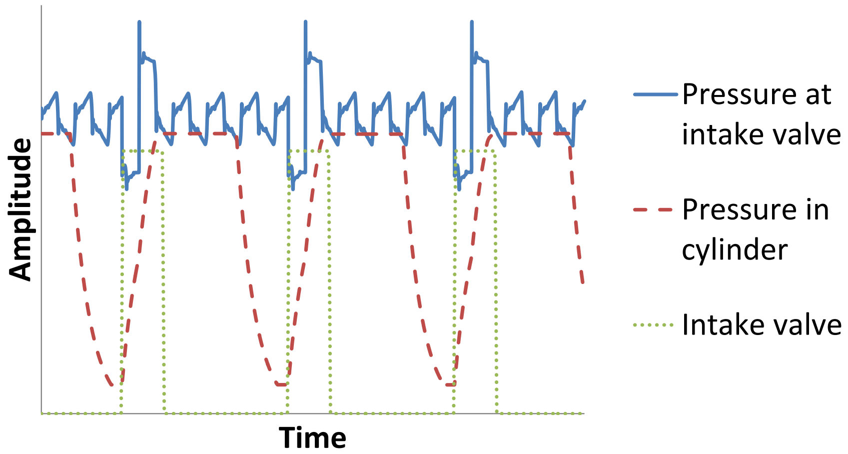

Figure 15: pressure phenomena at the intake

These oscillations are the result of pressure waves moving back and forth in the intake system caused by the regular opening and closing of the intake valve. It can be seen that the arrival of the wave's reflection allows the cylinder pressure to almost reach atmospheric pressure. When evaluating this intake system design for the whole engine speed range, it became clear that the cylinder pressure did not vary with the RPM value due to minimal oscillations. This means that the intake system did not cause the torque decrease and will not be changed.

4.4 Extension of the engine circuit

Once the exhaust design was determined, the next step was improving the entire electrical circuit in order to come closer to the simulation of the real situation. When it was proven that the intake system had no negative influence on the engine's performance, the electrical system was extended with the original intake.

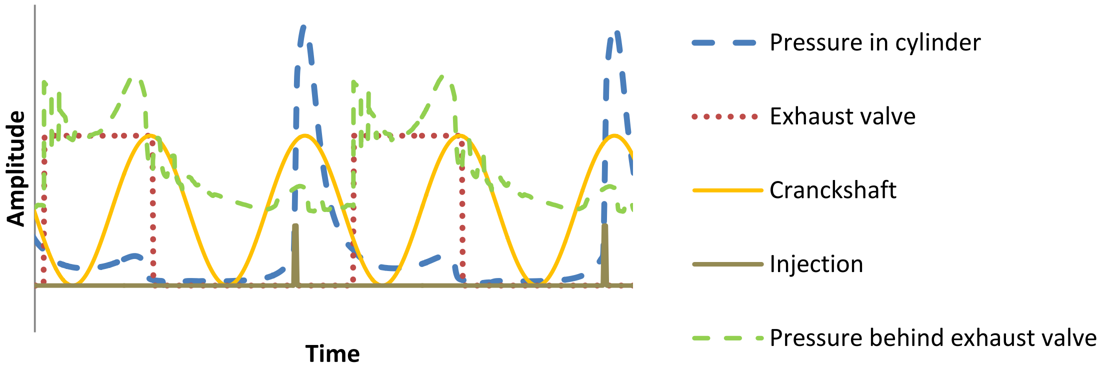

Figure 16: pressure phenomena for the tube with Y-junction and silencers in the extended electrical circuit

Secondly, all lossless transmission lines were substituted with lossy transmission lines. This improved the tube model because it allowed modeling not only resistance of the tube, but also the physical capacitive and inductive values based on the exhaust tube dimensions and gas temperatures.

Thirdly, the crankshaft movement was changed in two steps. The basic approximation from the elementary electrical scheme used a constant value for the capacitor, modeling the piston in a constant position in BDC. The first improvement was changing the capacitance i.e. the volume of the cylinder, suddenly from the maximum value to the minimum value according to a step function. The second improvement contained the implementation of a sinusoidal change, following the circulating movement of the crankshaft. This also allowed the modeling of the compression and expansion stroke to be more accurate. Then the ignition of the fuel-air mixture was added to the model in order to be able to incorporate the combustion. This is done by the current sources in Figure 5. Lastly, the pressure at the exhaust tip was set to atmospheric pressure.

The most important observation was the fact that the distinct phenomena caused by the pressure waves were less or not notable: most of the effects, like the strong pressure peaks caused by blow down, can still be seen, while other small effects, mostly pressure oscillations caused by reflections in the tubes, were overruled by capacitive, resistive and inductive effects. This is because all these changes in the circuit were a closer approximation of reality. This can be seen in the Figure 16.

4.5 Introduction of the silencer to the exhaust circuit

When the improved circuit had been verified and tested, the silencer was introduced (see Figure 17). In order to implement this silencer, two new transmission lines were needed at the end of the system. The silencer was modeled by one of these transmission lines because it can be seen as an exhaust tube with higher capacitive and inductive values and a lower resistance. After this silencer was fitted, the simulations were evaluated both concerning performance and the noise emissions. Because the silencer is an addition to the exhaust system design with Y-junction a silencing effect was noted and a total decrease of 11 dBA was achieved in the total noise production.

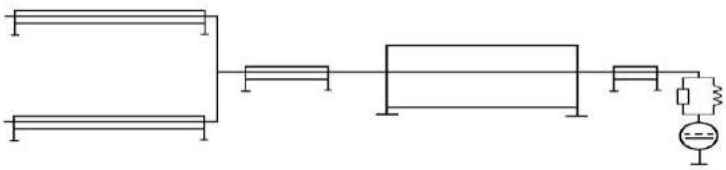

Figure 17: electrical scheme of the exhaust system with simple Y-junction and silencer

5 Prototyping

When all theoretical calculations were done, the implementation of the exhaust system was the next step. When implementing the design, one extra and crucial aspect was taken into account: flexibility in order to avoid damage when performing an engine swap during an endurance race.

Figure 18: wooden structure to produce the new exhaust with the old exhaust installed

First of all, a wooden structure (Figure 18) was constructed to indicate the space limits without having to use the Pegasus race car at the production site. This structure was attached to the engine to have a fixed reference point. Because the entire exhaust system consists of several parts, some of these parts can be of the shelve items while others are custom made.

The primary part of the exhaust system will be constructed using smooth tubes from the cylinder head until the Y-junction to ensure the exhaust gas can flow with the least resistance possible. These primary tubes will be bent into the correct shape in order to fit the whole system between the engine and the hood. The primary tube is connected to the engine using flanges. At the other side of the primary tubes, a welded Y-junction connects both primary tubes and makes sure the exhaust gas will flow into the secondary tube.

The second part of the exhaust system is connected to the first part using a flange. In order to guarantee an air-tight connection, an exhaust gasket is used. The flange used to connect both parts is welded to a short bended tube followed by a flex pipe. This flexible part has three different functions. First of all, it allows some maneuverability when performing an engine swap during a race. Also, by implementing this flexibility, the system allows some degree of freedom when changing the design of the engine or the engine packaging underneath the hood of the car. Last but not least, the flex pipe allows the primary and secondary parts of the exhaust system to vibrate separately, reducing the chance of cracking or breaking. This is a major advantage over a completely fixed exhaust system due to the high stiffness of the cars chassis and the large amount of vibrations that are coming from both the engine and the surface (the curbs in particular).

The flex pipe itself has a woven structure inside and outside. Thanks to this construction, the flow resistance of this type of flexible exhaust tube is much lower than that of flexed tubing which uses several small segments which slide over and under each other. The use of segments results in a rough, stepped pipe. These changes in diameter produce small pressure reflections and can act as small resistances to the gas flow.

The other side of the flex pipe is again welded to a smooth tube with the same diameter. The combination of these smooth tubes and the flex pipe form the secondary tube of the exhaust system. At the end of the secondary tube, the silencer is connected using a clamp. This is done because the current silencer is used for both cars of the CQS Group Racing Team and has to be interchangeable. At the very end of the exhaust system, a dB-killer is used. As the name suggests, this is a special exhaust tip used to silence the whole exhaust system some more with minimal loss of performance. This dB-killer is coupled to the silencer again with a clamp. To guarantee an air-tight connection, exhaust paste is used to seal the clamped connections. The whole exhaust system is supported using exhaust rubbers and bolts connected to the side of the body. The final design of the new exhaust system will have a similar configuration as the old design seen on Figure 18.

6 Validation

After all calculations and simulations were done, the results are validated. The simulations are validated using a frequency analysis while a second experiment will be performed in order to check the total performance of the exhaust system.

6.1 Frequency analysis

As a control for the correctness of the simulations, a frequency analysis is performed of the exhaust tip voltage, i.e. the noise the exhaust will produce. The result of the frequency analysis for the Y-junction design is shown in Figure 19. When taking a closer look at the lower frequencies, the engine rotational frequency must be visible in the spectrum. This closer look is shown in Figure 20 where a peak is visible at the frequency of 5400 min^{-1} or 90 s^{-1}.

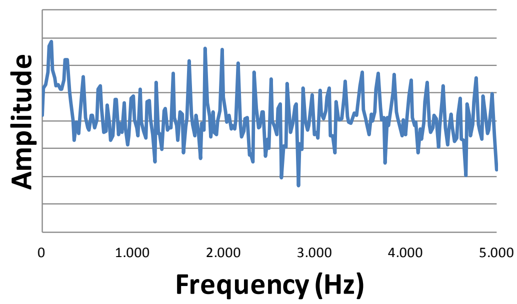

Figure 19: frequency plot

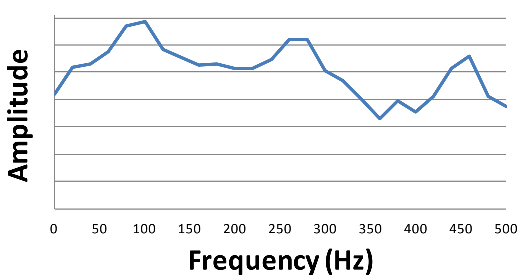

Figure 20: frequency plot focused on low frequencies

6.2 Dyno test

After fitting the car with the new exhaust system, a dyno test was performed. During this test, engine torque and emitted noise were measured. Figure 21 shows the new torque curve and compares it to the original one.

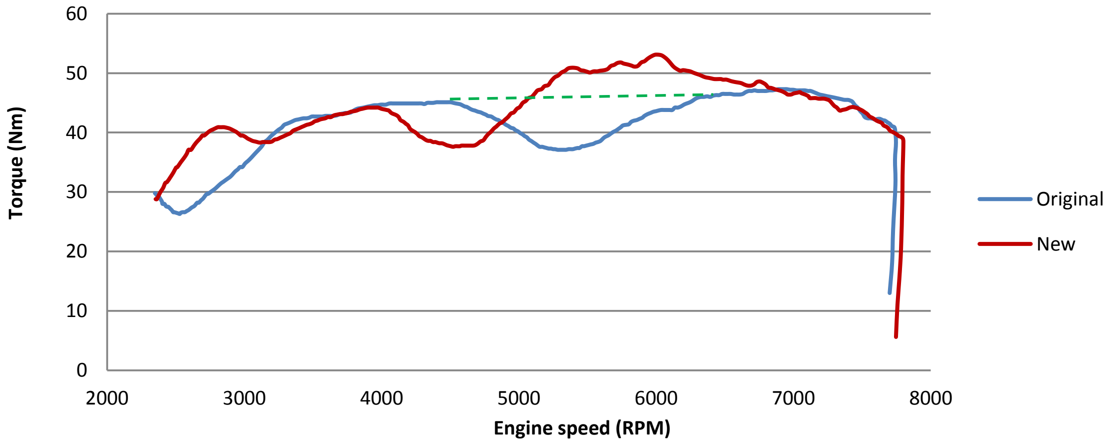

Figure 21: new versus original exhaust

One can see that the original torque decrease is not cancelled out, but it has shifted towards a lower RPM – range. In racing conditions, this new exhaust will perform better. When the driver shifts to a higher gear at an engine speed of 7500 RPM, the engine speed drops to 5000 RPM. At this engine speed, the torque is as high as if the original torque curve was flattened (green dashed line), which was the initial intention of this research. At engine speeds between 5000 and 7500 RPM, the new exhaust delivers an even higher or equally high torque than the original one. Since this is mainly the region where the engine operates during race conditions, the car can be expected to perform generally better (i.e. achieve shorter lap times).

The emitted noise was higher than the originally desired 95 dBA. It never crossed the 105 dBA limit though, which still allows the car to drive on Belgian circuits like Spa – Francorchamps and Jules Tacheny de Mettet, the main circuits visited by the CQS Group Racing Team KU Leuven.

7 Conclusion

This paper gives an overview of the design and production methods for an exhaust system for a Citroën 2CV race car. The design phase was done with simulations using ngspice software. This software makes it possible to simulate analog electrical circuits where inputs and variables were converted from physical entities to electrical entities. After the simulations were done, the results were converted back from electrical to physical entities. The simulations are verified through ngspice analyses and the final design's performance and noise emission will be evaluated during a dyno test. One can thus conclude that the use of an electrical simulation software can successfully be used to model flow and pressure systems in order to develop an exhaust system.

This paper used the two cylinder boxer engine of a Citroën 2CV race car to test the simulation method. The data that were collected using this simulation were verified with a prototype of the theoretically designed exhaust system by means of a dynamometer. The results show a shift of the torque decrease area from the original 5500 rpm to 4500rpm and a decreased width of the low torque area. The simulation results indicate the unbeneficial cylinder pressures occur at a lower RPM range and this range has become more narrow. The results on the dynamometer proved the correct interpretations of the acquired graphs and the small oscillations in the dyno graph can be explained by the imperfections of the exhaust prototype used for this test.

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