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The Application of a Wave Action Design Technique with Minimal Cost on a Turbocharged Engine Equipped with Water Cooled Charge Air Cooler Aimed for Energy Management

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Abstract

The vast majority of today's cars remain powered by gasoline or Diesel engines. In the European market, Diesel cars accounts for 55% of all new registrations while gasoline cars for 44%. All the other technologies such as hybrids, fully electric or natural gas and ethanol powered cars combine to make up the remaining 1%. Despite the publicity on these highly anticipated alternative technologies, statistics indicate that their usage remains marginal when compared to more 'traditional' powertrains. The internal combustion engine remains the principal way of transportation, at least for the foreseeable years. That being said, undesirable emissions (HC, CO, NOx, PM) are today's major concern because of their negative impact on air quality and global warming. The latest Euro 6 standard limits the CO emissions on a gasoline engine to 68% and the PM (particulate matter) on a Diesel engine to 96% lower than those established in 1992. CO2 emissions which are directly proportional to consumption rates are recently added to the list with a target of 95g/Km for 2020.

The main trend for solving these issues is through engine downsizing with high rates of turbocharging accompanied by fuel injection control, after-treatment and system integration. Techniques that increment the energy saving costs. The ICCT (International Council on Clean Transportation) estimates that the cost of taking a 4-cylinder 1.5L Diesel engine from no emission controls to the Euro 6 standard is around US$1400. A major technique to gain control of the engine processes and achieve high performance and low emission levels is through air handling depicted by air intake tuning, turbocharging tuning and charge air cooling.

This article briefly describes the air intake line of 4-cylinder 1.5L turbocharged Diesel engine and investigates the role of wave action phenomena on its intake line. The goal here is to achieve lower consumption and better performance by finding the optimal intake geometry. The latter comprises of a compressor wheel, a water cooled charge air cooler prototype (WCAC) and connecting pipes. The wave action effects were first explained and highlighted on a dynamic flow bench at "Mann+Hummel" using a frequency modeling approach which enabled to find the optimal length of piping upstream and downstream of the WCAC. Next this length was experimentally

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verified by mounting the different length on an engine test bench and registering pressure wave amplitudes. Finally the pressure recordings were fed to engine simulation software GT-Power™ for a transient low speed simulation mimicking urban driving conditions and specific fuel consumption data were recovered.
The result is a proposed air intake line for a turbocharged engine with a water cooled charge air cooler design choice. The proposed changes do not add any engine costs, but invests in significant R&D and retooling.

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1. Introduction

The attention of the automotive community today is focused on limiting emissions to meet the ever more stringent regulations without the detriment of drivability and driver comfort. Suppliers are forced to anticipate the regulation by investing in new part development and manufacturing processes.

One important aspect of engine design that has a first order influence on engine performance is the breathing of air of the engine [1]. This is depicted by the air intake line that has the role of 'feeding' air to the intake valves. Given the reciprocating nature of an engine, the movement of the intake valves creates pressure pulsations that travel along the piping of the intake and reflect back to the valves. The pressure levels are thus highly unsteady and compressible in nature and are of first order influence on combustion processes and exhaust emissions. This phenomenon is referred to as wave action at the intake [2] and has been the subject of study of many authors. The goal is to induce high pressure levels just before intake valve closing [3] and benefit from the wave action to acoustically charge the intake. The air intake line became more and more sophisticated over the years, with the addition of new elements to cope with the overall tendency of downsizing the engine. Downsizing is the car industry's response to meet the regulation and reduce fuel consumption. This is done by reducing the displacement of the engine and accessing lower operational speeds with an increase in efficiency. And in order to preserve drivability, the downsized engines are equipped with a matched turbocharger unit and a charge air cooler. The latter cools the air leaving the compressor thus compensating for the decrease of air density induced by the overpressure, this is crucial for performance targets, NOx emissions and knock prevention. From a thermodynamic point of view, the charge air cooler introduces a temperature gradient and a pressure drop. This paper will concentrate on understanding the wave action at the intake line of a modern Diesel turbocharged engine, notably the position of the charge air cooler relative to the entire intake line arrangement and its influence on certain engine parameters such as torque response and fuel consumption. Historically this effect of wave action was passed over [4][5] in favor of more direct pressure charging where the compressor and turbine control would produce the desired boost levels. This was done to facilitate the downsizing of engines. However, the turbocharger is not always capable of delivering the desired performance, this is particularly true at low engine speeds and loads. At these speeds, the enthalpy of the exhaust gases is not sufficient enough to turn the turbine wheel and thus the compressor provides no or minimal power. This is of great importance for today's commuters that spend a great deal of their time in urban driving conditions, under low speed operation (below 2000rpm) and partial load. This means that poor turbocharger efficiency under these conditions would be detrimental to fuel consumption and transient low end torque response, which is a driver's preference. Banisoleiman *et al*. [6] indicated that matching the turbocharger for full load efficiency often leads to inadequate boost pressures at low engine speeds. This is why wave action and acoustic tuning effects have become interesting for a turbocharged engine essentially for this low speed region where the turbocharger is inactive and the engine is fairly similar to a naturally aspirated one with comparable displacement.

In order to establish acoustic tuning, a reflection of the propagating waves must take place, this reflection is then correctly timed to the opening and closing of the intake valves. It would be hard to establish such a reflection at the main runners [2] as it is normally the case with naturally aspirated petrol engine for this would require impractically long runners [7] given the desired low speed tuning. Despite such limitation, Watson [8] and Sato *et al.* [9] showed that is possible to tune the intake air line for the case of a turbocharged Diesel engine. It will be shown in this paper that a reflection could be established at the interface of the charge air cooler and thus acoustic supercharging at these low speeds is possible. This will be done on a dynamic bench that creates an impulse excitation of mass flow at the entrance of any intake element and the consequent reflection is measured. Chalet *et al.* [10] provide a description of the bench as well as the measurement technique of a transfer function [11] that characterizes the frequency response of the geometry mounted on the bench. This transfer function models the dynamic behavior of the pressure waves at the intake by reproducing the unsteady and compressible effects of pressure. It was used to reproduce an engine simulation in a coupled routine with GT-Power^{TM} for a single cylinder engine [12] and a four cylinder engine [13]. In this paper, such a measurement technique is used, where a new water cooled charge air cooler is mounted on the dynamic bench. Water cooled charge air coolers have a multitude of benefits over traditional air cooled ones, they have a reduced pressure drop thanks to their smaller volume, better heat exchange properties, smaller packaging…

In this work, the charge air cooler was mounted on the dynamic bench in a multitude of arrangements. The length between the charge air cooler and the intake valves is investigated. A series of pipe lengths were tested and a final one is chosen. The objective is to optimize the volumetric efficiency of the engine at an operating speed of 1250RPM. This is a critical operating speed for which optimization would yield the best results. At this speed the turbocharger provides no boost pressure because the exhaust gases are not yet hot enough, so any possible acoustic tuning effects would have the greatest potential around this operating point. Next the series of pipe lengths are mounted on an engine test bench and the method is validated on engine, i.e. the optimal length for acoustic tuning is validated. Pressure measurements were recovered from the engine measurements and fed to a GT-Power^{TM} model of the engine. An engine simulation was produced, it comprises of a step from 1000 RPM to 2000RPM, which is the critical low speed operating region in question. It is shown that with the optimal acoustic length it is possible to achieve better torque response and lower fuel consumption

Nomenclature

ω	angular velocity (rad/s)
ò	damping coefficient
f	resonant frequency (Hz)
FFT	Fast Fourier Transform
i	resonant mode number
j	complex constant ($j^2 = -1$)
L	Laplace Transform
n	number of modeled resonant modes
p	dynamic pressure response (mbar)
pabsolute	absolute pressure with pressure drop, dynamic and initial components (mbar)
p0	initial pressure (mbar)
pdrop	steady state pressure drop (mbar)
P	frequency domain pressure data (mbar)

Qm	frequency domain mass flow data (kg/h)
PFmodel	modeled transfer function
PFmeasured	measured transfer function
qm0	excitation mass flow level (kg/h)
s	Laplace variable ($s = j\omega$)
t	time (sec)
X	inertial ram effect (mbar/kg.h)

2. Dynamic impulse bench

2.1. Experimental setup

The dynamic impulse bench consists on establishing a steady flow inside a given geometry, then once the flow is stabilized, a hydraulic actuator extinguishes the flow very abruptly in approximately 0.5ms. A pressure response is created inside the geometry following the mass flow excitation of the system. The dynamic bench was first introduced by Fontana *et al.* [14] and an electro-mechanical analogy was presented between the pressure response and the mass flow excitation. The hydraulic actuator and operating mechanism is explained by Chalet *et al.* [10]. The uniqueness of the bench is that permits the creation of high amplitudes dynamic pressure responses similar to the ones observed at the intake of an actual engine. The goal of this paper is to make use of this bench to find the optimal length of piping between the water cooled charge air cooler and the intake plenum of the engine in order to benefit the most from acoustic tuning effects at low engine speeds, mainly at 1250RPM. The length between the WCAC and the dynamic bench excitation was varied in order to study the effect of length according to three configurations: 238mm, 720mm length, 1250mm length. The internal diameter of all the tubes is fixed at 30mm. Figure 1(a) shows the water cooled charge air setup on the dynamic bench, the WCAC is represented as a schematic for it is a prototype version at this time. Pressure signals are measured using a Kistler 4045A2 piezo-resistive transducer with a sampling frequency of 20 KHz. Figure 1(b) shows comparison between the pressure responses for the three setups in relative measurement to the atmospheric pressure and in mbar. The initial steady pressure drop corresponds to the initial stabilized mass flow qm_0 inside the geometry. This mass flow is extinguished at 1.1 sec in from qm_0 to zero. Given the extremely rapid shut off time of 0.5ms a linear decline of the aspirated mass flow can be supposed. Figure 1(b) shows a comparison between the pressure responses for the three setups. For the case of the CAC with 238mm the initial mass flow is qm_0=190 kg/, with 500mm the initial mass flow is qm_0=178 kg/h whereas it is qm_0=167 kg/h for the third case with 1250mm between the bench's excitation and the CAC. The pressure response that is of most interest for this work is the dynamic response, after the mass flow excitation occurs, i.e. after the step in mass flow from the initial nominal value to zero.

2.2. Gas dynamic modeling

The pressure can be linearly decomposed [15] to three components: initial pressure inside the geometry, the pressure drop and the dynamic pressure component that exists after 1.1 sec. The absolute pressure is given by equation (1).

$$p_{absolute}(t) = p_0 + p_{drop} + p \tag{1}$$

The pressure oscillations that take place inside the geometry in response of the mass excitation can be modeled with a second order differential equation written in terms of the dynamic pressure and a source term [12][14].

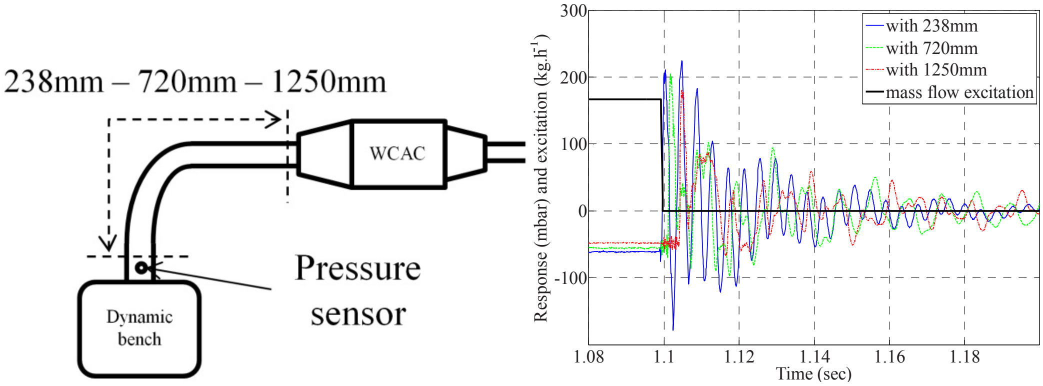

Figure 1. (a) WCAC with different lengths on bench; (b) Pressure responses for the three setups and mass excitation

Desmet [16] theorized that the pressure distribution upstream of the intake valve during the intake stroke follows a second order differential equation dependant on piston speed and valve diameter. If this equation is developed the gradient of the mass flow is found on the right hand side of the equation as depicted in equation (2).

$$\sum_{i=1}^{n}\left[\frac{1}{X_i}\left(\frac{1}{\omega_i^2}\frac{d^2p_i}{dt^2}+2\frac{\grave{o}_i}{\omega_i}\frac{dp_i}{dt}+p_i\right)\right]=\frac{dq_m}{dt} \qquad (2)$$

The pressure p_i of each resonance mode i is modeled with a second order equation then the dynamic response would be given by the sum of each individual mode as highlighted by equation (3). n being the number of modeled modes. In theory, there exists an infinite number of resonating modes, however only a certain number is considered; the energy associated to high frequency modes is much smaller when compared to the fundamental frequency.

$$\sum_{i=1}^{n}p_i=p \qquad (3)$$

Three parameters characterize the pressure response for each mode. The first is the angular frequency $\omega_i=2\pi f_i$, f_i being the resonance frequency. The second parameter is the damping coefficient $\grave{o}_i$ that model losses and viscous dissipation. The last parameter is the inertial ram parameter. These set of parameters will be used to judge which one of the three setups would be best for low frequency acoustic filling. Equation (2) can be transcribed in the frequency domain by applying a Laplace transform L . The transfer function given by equation (4) is obtained.

$$PF_{model}=\frac{L(p)}{L(qm)}=\frac{P}{Qm}=\sum_{i=1}^{n}\frac{X_i\,s}{\left(\frac{s}{\omega_i}\right)^2+\frac{2\grave{o}_i\,s}{\omega_i}+1} \qquad (4)$$

In order to identify the parameters of the transfer function, the latter is first measured from the experimental data on the bench. The transfer function is measured by accessing the frequency spectrum of the pressure response and mass flow excitation after appropriate windowing and high frequency filtering. Equation (5) gives the measured transfer function, in the frequency domain, using Fast Fourier Transforms. Figure 2(a) shows the modulus of the measured transfer function for the three test cases. The next step is to identify the parameters of the transfer functions given by equation (4). The resonant frequencies can be easily identified. They correspond to the location of the modulus peaks.

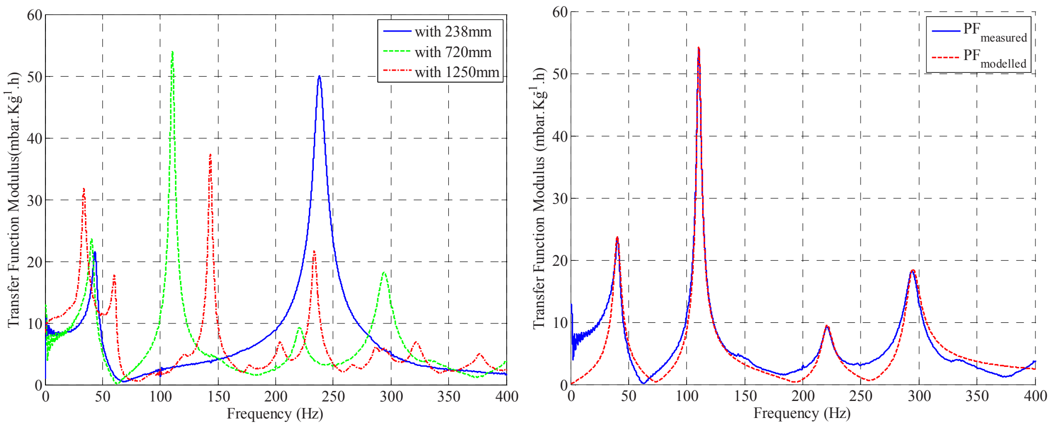

Figure 2. (a) Modulus of the measured transfer functions for all the test cases; (b) Modulus of the measured and modeled transfer function for the case with 720mm

$$PF_{measured} = \frac{FFT\left[p(t)\right]}{FFT\left[qm\right]} = \frac{P(j\omega)}{Qm} \tag{5}$$

The damping coefficient is given by equation (6)

$$\grave{o}_i = \frac{(\Delta f_i)}{f_i} \tag{6}$$

Where Δf_i characterizes a resonance mode's bandwidth relative to its center frequency It is defined as the frequency range for which the signal loses 3 dB from its maximum amplitude [17]. Once the frequency and damping coefficient are identified for a given resonance mode, it is possible to calculate the inertial ram parameter by replacing s in equation (4) with $j\omega$. Equation (7) is obtained. $\left|PF_{measured}\right|_i$ is the modulus of the transfer function at the ith resonant mode. Figure 2(b) shows the good correlation between the modeled and the measured transfer function for the case of the WCAC setup with 720mm. This means that the transfer function methodology is correctly capable of reproducing the dynamic effects.

$$X_i = \frac{2\left|PF_{measured}\right|_i \grave{o}_i}{\omega_i} \tag{7}$$

2.3. Results

As explained before, the engine speed to be optimized is around 1250RPM, which gives a target frequency of 1250/30=41.66Hz. The objective is to have maximum pressure amplitude at and around this frequency. The solution of the differential equation (2) in the time domain can be approximated by equation (8). On a real engine, the mass flow is given by the load and operating speed. By maximizing the inertial ram parameter, it is possible to maximize the pressure amplitudes and thus improve acoustic tuning.

$$p(t) = qm_0 \sum_{i=1}^{n} X_i \frac{\omega_i}{\sqrt{1-\epsilon_i^2}} \exp\left(-\epsilon_i \omega_i t\right) \sin\left(\omega_i \sqrt{1-\epsilon_i^2}\, t\right) \tag{8}$$

Figure 3 gives the calculated inertial ram parameter using equation (7) as a function of frequency for the different cases. The case where the WCAC is mounted with 1250mm gives the greatest values of this parameter at the frequency range of interest. As the frequencies get higher, the length of piping between the bench (or the engine) and the WCAC must be smaller if acoustic tuning effects are solely wanted.

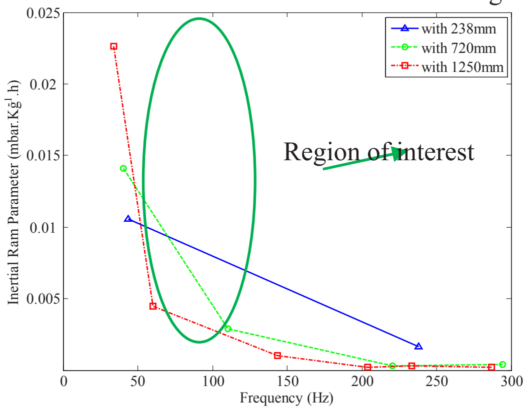

Figure 3. Inertial ram parameter for the test cases

3. Engine application

3.1. Experimental pulse generator

The same test cases were repeated on an experimental engine test bench. Figure 4(a) gives a schematic representation of the bench in question. The 1.5L Diesel engine is driven by an electric motor without combustion, so the 8 valves arrangement acts as a pulse generator. Because the engine is run without combustion, the turbocharger is controlled independently. This was done by regulating pressure and temperature on the turbine side using a gas stand in a way to simulate actual exhaust conditions. A full load situation was considered, thus the pressure and temperature at the inlet of the turbine were closely controlled in a way to satisfy these demands. The original intake line from the CAC outlet to the plenum inlet was replaced with a series of tubes and bends. These tubes are interchangeable and thus permit to change the length. This will highlight the effect of these lengths on engine performance. The engine was run at operating speeds ranging from 1000RPM to 2000RPM, and the pressure at the entrance of the intake plenum was recovered. The length was changed in the same fashion as the previous experiments on the bench, however due to the intake plenum's geometry and the restricted place on the engine, it was not possible to test the exact same series of lengths. A modular design of piping is chosen where small bits of tubes are interchangeable. The three test cases are 245mm, 900mm and 1430mm, the distance is measured from the intake plenum to the WCAC boundary. Figure 4(b) shows the registered pressure at 1250RPM, the operational speed for which the system is to be optimized in order to enhance low end torque and fuel consumption. The relative pressure from Figure 4(b) shows that the 1430mm length gives the greatest pressure amplitudes upstream of the intake valve at 1250RPM. This was already verified from measurements of the inertial ram parameter on the dynamic bench.

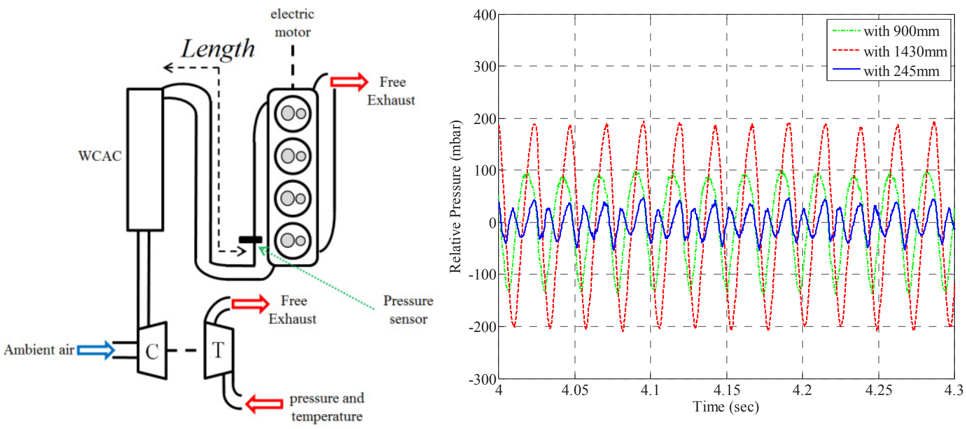

Figure 4. (a) Schematic representation of the engine bench; (b) Relative pressure at the intake at 1250RPM for different lengths

3.2. Engine simulation

The recorded pressure on the engine test bench was used to perform an engine simulation and study the effect of wave action on the performance of the turbocharged engine. First a 1D model of the engine was constructed in GT-Power^{TM} and calibrated against real engine data. Next the engine block (pistons, valves...), exhaust line and plenum chamber were kept and the rest of the intake line was replaced by a virtual boundary. At this boundary, the experimental relative pressure was fed into the model. The mean pressure remains unchanged, it is provided by the turbocharger and is given by Table 1. The boost pressure is controlled by the variable geometry on the turbine. The boost pressure at low engine speeds is virtually inexistent and a downsized turbocharged engine suffers from poor efficiency at these speeds. The goal of the consequent engine simulation is to address this issue using the wave action effects brought on by the correct length. Indeed the added length of piping will introduce an additional pressure drop at the intake; however this would be compensated by variable geometry of the turbine in order to remain consistent with the pressure mapping given in Table 1. It is during low speed transients, mainly in urban conditions that the engine has to be optimized. For this reason a simulation case was considered where the driver requests full load at third gear, the car being at an initial rolling speed of 17 km/h at 1000 RPM.

Table 1. Boost pressure map

Engine Speed (RPM)	1000	1250	1500	1750	2000	2250
Boost pressure (absolute bar)	1.07	1.504	1.957	2.207	2.48	2.567

The simulation is interrupted for a terminal speed of 34 km/h or 2000 RPM, this is done to study the impact on low end torque and response time for the turbocharged Diesel engine. Real car data was entered as input parameters to the software, such as curb weight, tire diameter and drag coefficient. Figure 5 gives the torque response and the specific fuel consumption as a function of engine speed during the engine transient. By looking at the torque response from Figure 5(a) it can be seen that for the 1430mm length, the low end torque has been enhanced by as much as 7% compared to the short length. At 1500RPM the torque curves for the long and intermediate lengths meet, whereas for speeds greater than

1500RPM, the intermediate length of 900mm offers an improvement in torque. This is because acoustic tuning for higher engine speed occurs for shorter lengths.

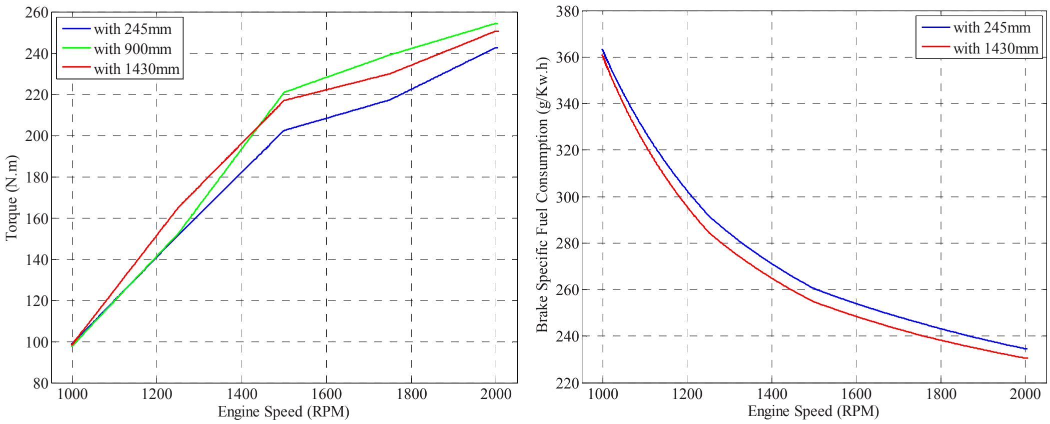

Figure 5. (a) Transient torque response with the different lengths; (b) Brake specific fuel consumption BSFC

This can be explained and predicted using the dynamic bench experiments. In fact the frequency of the intake pressure pulsations for 1500RPM is 50Hz, and by looking at Figure 3, it can be seen that the inertial ram parameter values coincide around 50Hz for the long and intermediate lengths, and that for frequencies above 50Hz, the ram parameter has a larger value for the 720mm length than it for the 1250mm length. The desired length can be chosen a priori on the dynamic bench following simple rapid tests depending on the engine speed range that is to be optimized. As explained before, the engine speed to be optimized here is around the 1250RPM mark. For this case, the optimal length would be a runner of 1430mm. Figure 5(b) gives the specific fuel consumption during the transient simulation, a gain of 2.2% in fuel consumption is observed when changing the length. That being said, an engine simulation on a complete cycle such as the NEDC would be more precise that a transient low speed step. However this would require the measurement of pressure on the engine bench for different engine loads versus the full load case presented here. This work is the subject of current research and would give a better overall idea on the proposed gain by this method. One way to benefit from the higher torque potential presented in Figure 5 is by using downspeeding techniques. The idea is to benefit from the gained torque by changing the gearbox ratios to longer gears leading to lower engine operation speeds and improved efficiency. It was recently presently that the application of downspeeding in a Diesel vehicle, in conjunction with optimizing transmission gear ratios delivers improvement in fuel economy that can reach 6.6% over the engine baseline [18]. Thus extending the gear ratios for the optimized intake with acoustic tuning would enable lower CO2 emissions and fuel consumption.

4. Conclusion

The effect of wave action on the performance of a turbocharged engine was presented. This was done by studying the effect of length between a water cooled charge air cooler prototype and the intake valves. A dynamic bench with a mass flow excitation was used to highlight the acoustic wave effects using a simple transfer function methodology. The method was validated against experimental engine measurements and it was found that is capable of producing reliable results. Correctly choosing the length of piping between the charge air cooler and the intake valves along with appropriate turbocharger control enhances the low end torque and efficiency of the engine without compromising high speed operation. This was verified using a combination of experimental pressure data and engine simulation software by setting up a theoretical low speed transient operation, aiming at reproducing urban driving conditions.

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