TURBOCHARGER SURGE NOISE MEASUREMENT AND SOLUTION USING EXPERIMENTAL TECHNIQUES

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Turbocharging is used as a means to downsize petrol engines, thereby, producing more power for a lower engine size, when compared with a naturally aspirated engine. The introduction of a turbocharger means that a throttle body is included in the petrol engine downstream of the turbocharger compressor. Due to the presence of this throttle valve, flow is restricted through the outlet pipe of the turbocharger during low load engine operation. For example, during transient tip out – tip in maneuvers. Hence, there is a chance of the turbocharger operating in near surge or surge conditions. Surge noise generated during this stage is clearly audible and can also affect the durable life of the compressor and other rotor systems of the turbocharger. This paper describes an experimental method to predict and measure the turbocharger surge noise. An experimental turbocharger noise rig, designed and built for this purpose, is explained. Using a time and frequency analysis of the measured data the fundamental mechanism of noise generation is identified. Finally, a passive solution to reduce the surge noise is proposed.

1. Introduction

There is a continuous effort to improve efficiency, reduce fuel consumption and reduce the emissions of the internal combustion engine. Downsizing, down speeding, cylinder deactivation, charge dilution and adapting the Miller cycle are some of the major changes made to engine designs to achieve this. The power to weight ratio or the power to volume ratio of the engine is improved by downsizing. Introduction of a turbocharger in the gasoline engine is an important method used to downsize the engine. The weight to power ratio has approximately halved over the last 25 years for both petrol and diesel engines [1]. It has been stated in the literature [2] that a turbocharged engine with the same torque output as that of a non-turbocharged engine produces higher sound pressure level. The increase in the noise level of a turbocharged engine in the speed range of 1500 to 2800 rpm is attributed to the boost pressure build-up and, hence, due to the turbocharging. The noise observed in the turbocharged engine are classified into: (i) constant tone noise; (ii) unbalanced whistle and pulsation noise; (iii) rotating noise; and (iv) blow noise or whoosh noise.

The frequency of noise does not change with time in the case of a constant tone noise (frequency range 500-1000Hz). Unbalanced whistle or pulsation noise is caused generally by the rotor eccentricities and blade geometry and is highly dependent on the rotating speed of the turbocharger blades (frequency range 1.2 to 4.5kHz)[3, 7]. Rotating noise is a product of pressure difference between the suction and compression side of the blade and also by the blades passing the tongue of the housing. (frequency up to 13 kHz). Whoosh noise is a broad band noise. This is typically caused due to the high torque demand at lower engine speed which translates to the turbocharger surge operation, i.e. in the event of low air flow across the rotor and high pressure ratio. This condition is generated during the throttle tip-in condition, but also found during steady state driving condition, in petrol engines. In diesel engines this is caused by higher loads at low engine speeds. This can also occur during the transient tip-in tip-out manoeuvre of automobiles during which, high pressure is imposed on the compressor outlet followed by lower air flow. This causes the turbocharger to operate at a near surge or surge condition, thereby, leading to the generation of whoosh noise.

This paper focusses on the surge noise caused by the transient manoeuvre of the throttle valve in a petrol engine. Simulation methods used to predict the dynamic pressure fluctuations in the intake system of a petrol engine are presented. The design of a compressor rig used to re-create the field problem under laboratory conditions is explained. Experiments conducted on the supercharger rig to create the surge noise and subsequent analysis of the data are discussed in detail. Finally, the mechanism of noise generation is identified and a passive solution to the problem is proposed.

2. Methodology

Whoosh or surge noise is observed in a turbocharged petrol engine during the throttle tip-in tip-out manoeuvre. The phenomena is assumed to be due to the operation of the turbocharger near to the surge zone and also due to the timing of the compressor recirculation valve (CRV) used in the system.

The compressor map of a turbocharger is divided into three zones: a) stable operating zone (centrally placed); b) surge zone; and c) choking zone [4]. When the mass flow through the compressor is reduced at a constant pressure ratio, a point arises when local flow reversal happens at the boundary layers. This will result in a low efficiency of the compressor. If the flow rate is further reduced, complete reversal of flow occurs and this will relieve the adverse pressure gradient until a new flow regime and pressure ratio is established. The compressor is then said to be in the surge operation.

This paper focusses on the surge zone and near surge zone operation of the compressor. A typical turbocharger map as described above is based upon measurements on a static test bench. However, when the turbocharger is integrated to the intake system of a vehicle, the map could shift and the surge margin can be reduced. This is due to the dynamic operating condition of the engine. One of the aims of this project is to predict the occurrence of surge in a given engine design and to find the precise mechanism of noise generation. The two routes followed in order to achieve the aim are explained: (i) prediction of the surge occurrence using one dimensional engine simulation software; and (ii) experiments on a specially designed compressor rig.

3. Engine simulation

A commercial one-dimensional engine code is used to simulate the turbocharged petrol engine performance. Navier-Stokes equation is solved in the flow model and applied to the laminar and turbulent flow [5, 6]. The objective of the simulation is to predict the occurrence of higher pressure fluctuations during the defined manoeuvre and to compare this with the normal working of the engine.

Other properties such as the mass flow rate and the temperature are also monitored. A spark-ignited, four-cylinder, turbocharged, gasoline direct-injection engine model is used for the simulation. A turbocharger with an intercooler to maintain the temperature of intake air is integrated in the model. A CRV with a changeable opening area is modeled and included. Sensor connections are included to monitor the static pressure at intake pipe before the air filter, before the compressor, at the compressor outlet and throttle inlet. Sensors to monitor the mass flow rate, the speed of the engine, the speed of the turbocharger shaft, the opening area of the CRV and the opening area of throttle are also incorporated in the model.

A short duration time step of the order of 0.4 micro second is used to obtain good results at higher frequencies. The throttle position, CRV position and the engine speed data are obtained from experimental measurements on a turbocharged petrol engine car undergoing the tip-in and tip-out manoeuvre. The CRV is opened and closed in a profile different to that of the throttle profile. In an engine, the CRV is typically opened using mechanical means by using differential pressure acting on a diaphragm. Hence, the timing of the CRV is highly dependent on the responsiveness of the mechanical valve to the pressure fluctuations in the intake system. In the simulated engine model, the CRV profile is provided as an input. The opening time, closing time and, hence, the duration of opening are variables in the model. For the current simulation, the CRV is set to open when the throttle valve is beginning to shut, i.e. when the engine speed begins to drop.

As the present study involves analysis of transient signals, the FFT may not be applied directly. Hence, the short time Fourier transform (STFT) was used to analyse the pressure signals in the intake system. The analysis results are shown in Fig. 1 and the CRV is kept closed. A pressure rise develops as there is no recirculation of air. The results show that the noise level in the circled region (Fig. 1) is higher than when the CRV is open [10].

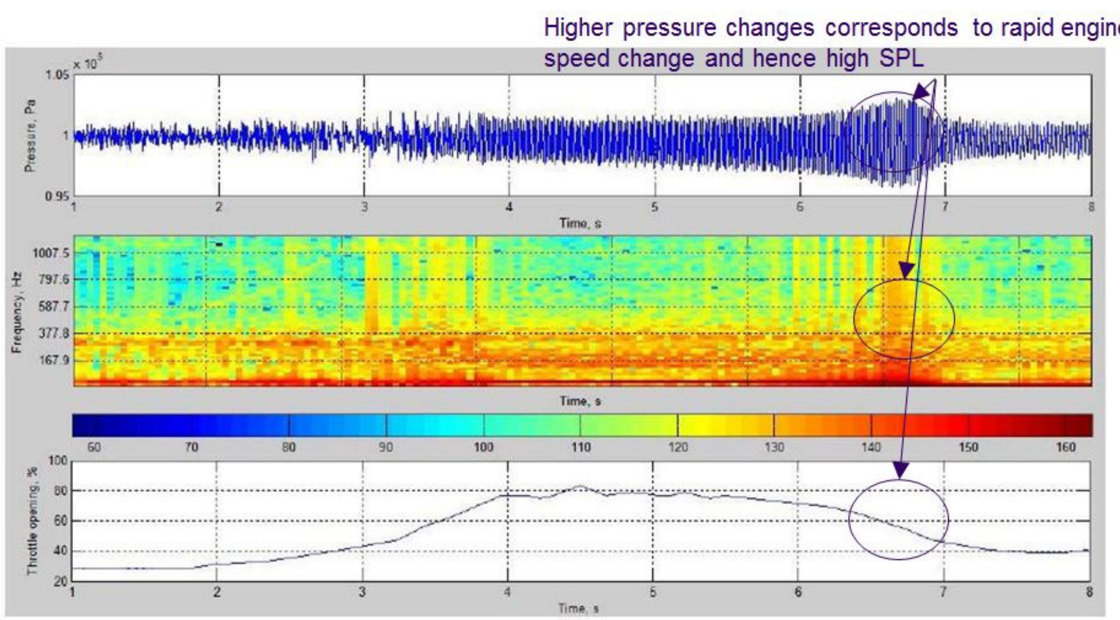

Figure 1. - STFT of the static pressure at the compressor inlet location with the CRV permanently closed (colour code shows the sound pressure level in dB)

4. Experimental rig

The aim of the test rig is to recreate the pressure dynamics within the intake system of an automotive engine under laboratory conditions. The test rig is designed to be of modular type and the intake system can be changed to conduct experiments on different types of induction systems and turbochargers (Fig. 2). Further details of the design are given in reference [10].

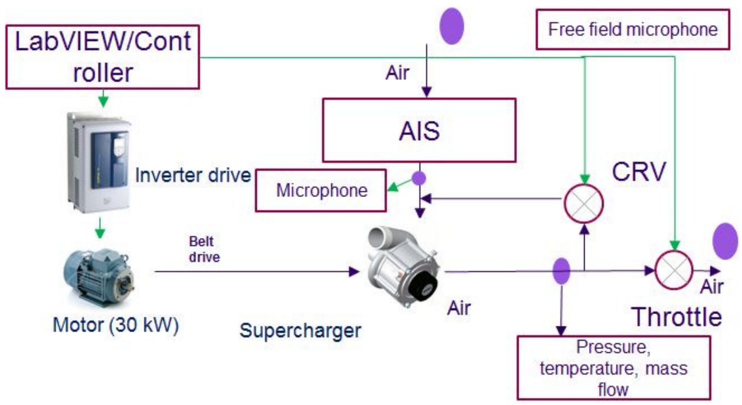

Figure 2. Schematic representation of test rig

Design of Experiments (DoE) was used to study the transient operating conditions. In the DoE technique, factors are systematically varied during an experiment in order to determine their effect on the response variable. The factors chosen in the experiment can have only a few values which are known as factor levels in the design. Factors can assume continous values or specific values in the boundary condition definition. In the case of continuous factors, specific levels are chosen to perform the experiment. The first step is to identify the factors and also the levels to be considered. The previous experiments are used to select the factors. The objective of this study is to understand the causes of the surge noise generation and then direct the analysis to establish the mechanism.

A half-factorial experiment comprising 3 factors and two levels was designed using Minitab software. The factors considered were the compressor shaft speed, throttle open position and the CRV open position. The response noted is the maximum sound pressure level measured during the 0% throttle open condition.

The interaction of factors that generate the maximum sound pressure level is difficult to determine by just observing the results in a tabular form. Hence, a plot is generated using the ANOVA section of the Minitab software. An interaction plot is used to represent a matrix of plots for the present set of experiments (Fig. 3). These plots are useful for judging any interaction as they represent the plots of the mean noise level for each factor with the level of a second factor held constant. The interaction between the levels of two or more factors can influence the response levels in experiments. An interaction is absent when the lines in the interaction plot are parallel. A departure from the parallel state indicates a higher interaction between levels.

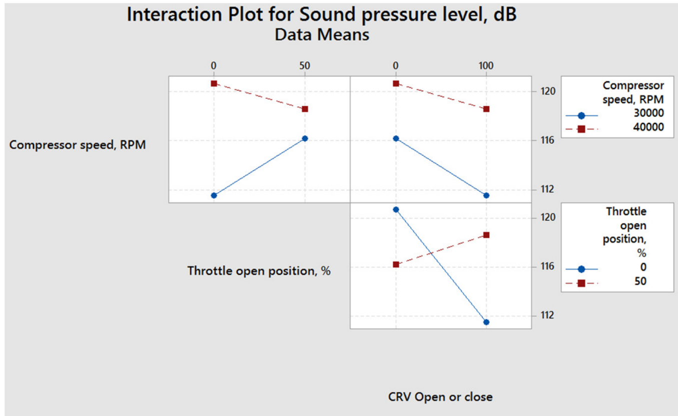

Figure 3. Interaction plot showing the influence of factors on the mean noise level data

Observing the lines in Fig. 3, it can be concluded that the surge noise levels are not dependent on the compressor speed variation for the same CRV open position. Hence, the surge noise is likely to occur during different compressor speeds. It is noted that the surge noise amplitude is higher in the closed CRV condition irrespective of the compressor speeds. The lines at different levels of compressor speeds are not parallel and are highly deviating from the parallel condition. This indicates a strong interaction between the throttle open position and the response. The lines show that at 0% throttle condition, 40,000 rpm shaft speeds demonstrates a higher noise level than the 30,000 rpm. The higher noise level at 40,000 rpm is due to the higher flow rate and associated dynamics. However, the noise level at 30,000 rpm is lower at 0% throttle than at 50% throttle open condition. This is due to the fact that the CRV is opened at 0% throttle and this reduces the sound pressure level as indicated in Fig. 3. At 50% throttle condition, the noise level at 40,000 rpm is reduced as the CRV is opened in this case also there is no surge occurrence noted. Thus, it is derived that the throttle opening position has a strong contribution to the surge noise occurrence with lower throttle open position generating a higher surge noise. However, in actual automotive engine operations, the throttle positions cannot be restricted from being fully closed.The interaction plot for this section is provided in the bottom right of the interaction plot matrix. The blue coloured line, which is for the throttle fully closed condition, shows that the CRV opening condition has a strong influence on the response. The CRV fully closed position creates a higher sound pressure level at a fully closed throttle condition. The sound pressure level is reduced considerably when the CRV is opened at the same throttle condition.

5. Design solution to the problem

5.1 Acoustic radial resonances

The effect of the CRV and the throttle opening on the noise intensity during the surge region is noted from the experimental and analytical work performed. The surge noise frequency is found to be generally independent of the speed of the supercharger. This was evident from the experiments at different compressor shaft speeds of 30,000 rpm and 40,000 rpm in that the surge noise frequency remains within fairly narrow bands. Also the variation in the frequency was not found to be proportional to the change in the speed of the supercharger. However, it was found that the intensity of the noise was much higher in the 40,000 rpm shaft speed than with the 30,000 rpm shaft speed. This phenomenon is possibly due to the increase in the flow velocity at higher supercharger shaft speeds [9] and, hence, higher pressure dynamics. Hence, it may be concluded that the surge occurrence is a flow induced phenomenon.

One type resonance explained in the literatures is the radial acoustic resonance or cross-mode resonances [8, 9]. The intake duct represented in the design of the turbocharger acts as a waveguide for the acoustic waves in the intake air media. Fahy and Gardonio [9] showed that the pressure distribution of modes of a rigid – walled, uniform, cylindrical waveguide of infinite length take the form

(1) $$p_{np}(r,\phi,z) = \tilde{p}_{np}\,{}^{\cos}_{\sin}(n\phi)J_n(k_r\, r)\exp(-jk_z z)$$

where, $\tilde{p}_{np}$ is the pressure amplitude, J_n is a Bessel function, k_r is the radial wave number and is determined from the equation

(2) $$[J'_n(k_r r)]_{r=a} = 0$$

where, n is the number of diametral pressure nodes and p is the number of concentric circular pressure nodes. The cavity resonance frequency is expressed as

(3) $$f_{np} = J_{np}(a/d)$$

where, a is the speed of sound in the contained fluid (m/s) and d is the basic hydraulic (geometric) diameter of the pipe (m). With reference to a table of characteristic values for the cylindrical pipe solutions, the natural frequency for a given mode can be calculated [8, 9]. Representations of nodal patterns as examples are given in Fig. 4.

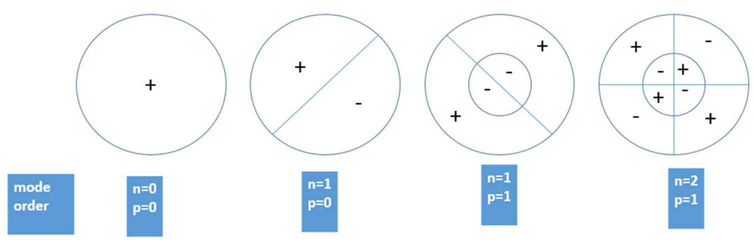

Figure 4. Nodal pattern for Acoustic Radial Modes

Referring to Fig. 4 containing nodal pattern, the following combinations of n, i.e. number of diametral pressure nodes and p, i.e. number of concentric pressure nodes, are selected for the analysis: (n=1, p=0); (n=0, p=1); (n=2, p=0); (n=0, p=2). The resonant frequencies calculated corresponding to a temperatures of 45°C are given in Table 1.

Table 1. Resonant frequencies corresponding to 45 deg C

Location	Supercharger inlet pipe (blue)		Supercharger inlet (aluminium)	Supercharger outlet (aluminium)	Supercharger outlet pipe (blue)		Throttle inlet	CRV hose
Diameters (mm)	50	76	70	53	63	57.5	52	12
Resonant frequency (1,0)	4190	2757	2993	3953	3325	3644	4029	17458
Resonant frequency (0,1)	8720	5737	6228	8226	6920	7582	8384	36332
Resonant frequency (2,0)	6950	4573	4964	6557	5516	6044	6683	28959
Resonant frequency (0,2)	15964	10503	11403	15061	12670	13882	15350	66518

This resonant frequency zone is marked in the Table 1 as yellow coloured boxes. It may be concluded that the radial acoustic resonant frequencies are closer to the surge frequency. Hence, it may be concluded that the mechanism of surge noise in the intake system is due to a radial acoustic resonance at the supercharger inlet or outlet locations.

5.2 Design of a solution

This section discusses the methodology adopted to solve the problem of surge noise. The aim is to reduce the intensity of the noise. The focus of the solution is given to a passive method. One of the major difficulties in the implementation of an active solution is that the additional electronics and hardware could cause a cost increase in the turbocharger part. However, compressor map based corrections are possible with the active solution. In the case of a passive solution, the design should lead to a more cost effective solution.

Referring to the calculation performed for radial acoustic resonance, it has been found that the compressor inlet and compressor outlet are the regions for which the resonance frequency occurrences match with the measured surge noise frequency. Hence, modifications or addition of parts for passive solution was planned for the compressor inlet and outlet locations. Modification of existing parts has been given a low priority as the shape of the parts affects the packaging of the intake system on the powertrain system.

Referring to the Table 1, n=1, i.e. number of diametral pressures nodes and p=0, i.e. number of concentric pressure nodes corresponds to the case where the calculated radial acoustic frequency matches with the measured surge noise frequency. Fig. 4 illustrates this nodal pattern. Thus, additional parts to be inserted into the cavity has been designed and is given in Fig. 5. The parts are designed to enable them to be bolted into the inside of the compressor inlet and outlet pipes. The ribs and the wall of the parts are designed with adequate thickness to withstand the pressure due to high mass flow rate and temperature. Minimum thickness is very important in order to maintain the required pressure drop. The prototype parts were manufactured using a rapid prototyping machine.

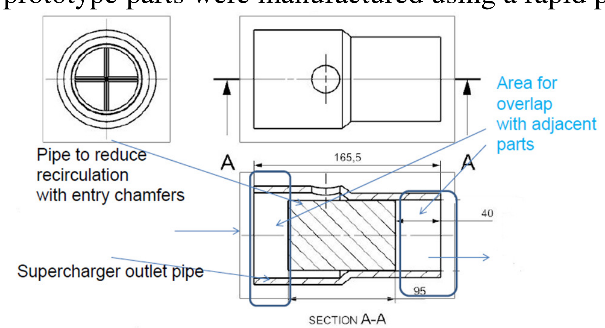

Figure 5. Drawing of the pipe used for the passive solution

5.3 Results

Four experiments were planned to quantify and understand the effect of the introduction of the passive solution parts inside the compressor inlet and outlet pipes. The experiments conducted are: (1) without using passive solution; (2) passive solution installed at the outlet of the compressor; (3) passive solutions installed both at the inlet and outlet of the compressor; and (4) with passive solution only at the inlet of the compressor. The supercharger shaft speed is maintained at 30,000 rpm for all of the experiments. The results displayed in Fig. 6, for the part fitted at the inlet, shows a maximum noise level of 106 dB. This is significantly reduced (10 dB) from the original condition. This reduction is comparable to the use of the CRV on the turbocharger. The passive solution installed displayed a noise reduction of 10 dB in spite of the fact that the CRV is not opened.

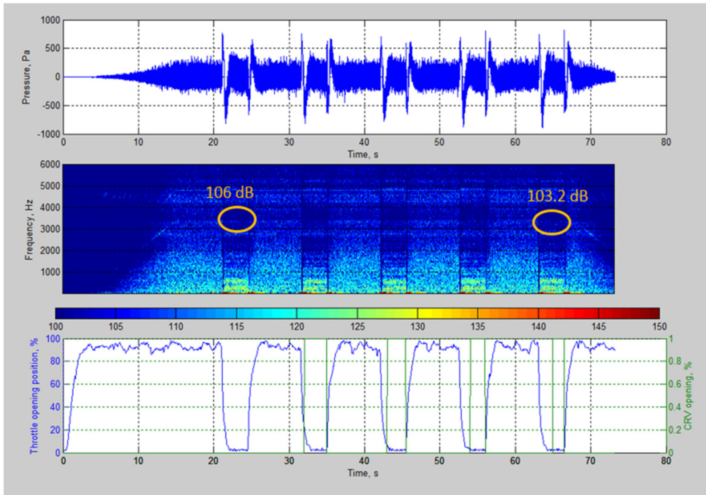

Figure 6. STFT of surface microphone reading at compressor inlet location (colour code shows sound pressure level in dB)

6. Conclusions

One of the challenges with the introduction of a turbocharger in an engine is the increased noise in the intake system. In particular, whoosh noise at the intake manifold during surge operation in a gasoline engine. In this paper, simulation and experimental methods were used to investigate the surge operation and to predict the resulting pressure pulsations and noise. The influence of the CRV opening on the noise characteristics in the intake system are illustrated by performing a STFT of the predicted static pressure signals. An interaction plot of various factors and their effects on the sound pressure level has been discussed. The mechanism of the noise generation has been identified using a radial acoustic frequency calculation. A passive solution has been designed and the reduction of surge noise is demonstrated using experiments conducted on a specially designed test rig.

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