

NVH Integration of Twin Charger Direct Injected Gasoline Engine

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ABSTRACT

The increased focus and demands on the reduction of fuel consumption and CO₂ requires the automotive industry to develop and introduce new and more energy efficient powertrain concepts. The extensive utilisation of downsizing concepts, such as boosting, leads to significant challenges in noise, vibration and harshness (NVH) integration. This is in conflict with the market expectation on the vehicle's acoustic refinement, which plays an increasingly important role in terms of product perception, especially in the premium or luxury segment. The introduction of the twin charger boosting system, i.e. combining super and turbo charging devices, enables downsizing/speeding in order to achieve improved fuel economy as well as short time-to-torque, while maintaining high driving dynamics. This concept requires also extensive consideration to NVH integration. The NVH challenges when integrating a roots type supercharger are very extensive. The high frequency source characteristics of the supercharger result in complex wave propagation inside the intake duct system since exciting pulsation orders are well above duct cut-on frequencies. The source strength in relation to audible interior tonal noise threshold is also very high. In addition the background masking levels in terms of mainly combustion related powertrain and road noise are low with the consequence that the orders (tonal noise components) can be prominent and annoying even with a high degree of acoustic source treatment (remedies). The scope of this paper is to describe quantification of the charging system noise radiation and propagation including subsystem target cascading synthesis.

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INTRODUCTION

The refinement of powertrain induced noise and vibration is one of the most important tasks during the vehicle development process. It strongly influences the driving perception of the vehicle together with attributes like drivability and vehicle dynamics. The expectation on NVH refinement from the customer point of view is also continuously increasing, especially in the premium and luxury segments. The task of NVH refinement gets even more challenging when new powertrain technologies like downsizing and down-speeding are introduced to meet both market expectations and legal regulation on CO₂ emission reduction. In combination with shorter product development time, due to that the automotive OEMs have to put products faster to the market and which is driven by competition, the reduction of both hardware stages and number of builds in a given hardware stage leads to that NVH refinement becomes even more complex. To minimize the risk of failure, advanced or research projects need to be executed up front of the ordinary product development projects in order to address all new concepts and technologies. The created new knowledge due to these "front-loading" activities may result in new specifications and requirements with associated assessment methods. All of this needs to be used in the later product development projects in terms of

benchmarking, specification of technical requirements including system and component requirement down-cascading, concept evaluation, selection and optimization using virtual and hardware methods and finally validation. To be able to close the loop, the feedback from the learnings during the development projects is transferred back into the specifications, standards and methods.

In the newly developed VEA engine family at Volvo Car Group a twin charger boosting system, i.e. a combination of a belt driven roots type supercharger and a turbo charger, is used for the top performance petrol engine to meet requirements on power and torque as well as fuel efficiency [1,2]. The supercharger provides boosting in the low engine speed range while a single large turbocharger, which is needed to meet peak power and torque requirements, operates from mid to top end engine speeds. The concept of a belt driven supercharger has significant advantages in response time at lower engine speeds compared to a single large turbocharger as the transient response time, or turbo lag, depends on the exhaust gas projection into the turbine geometry and polar inertia of the turbocharger [3]. The supercharger is controlled by an electromagnetic clutch, which enables disengagement and hence improves the fuel economy at steady state driving conditions as well as at the mid to top speed ranges where the

turbocharger provides the boost. The outline of the twin charging system with its operating modes is visualized in Figure 1.

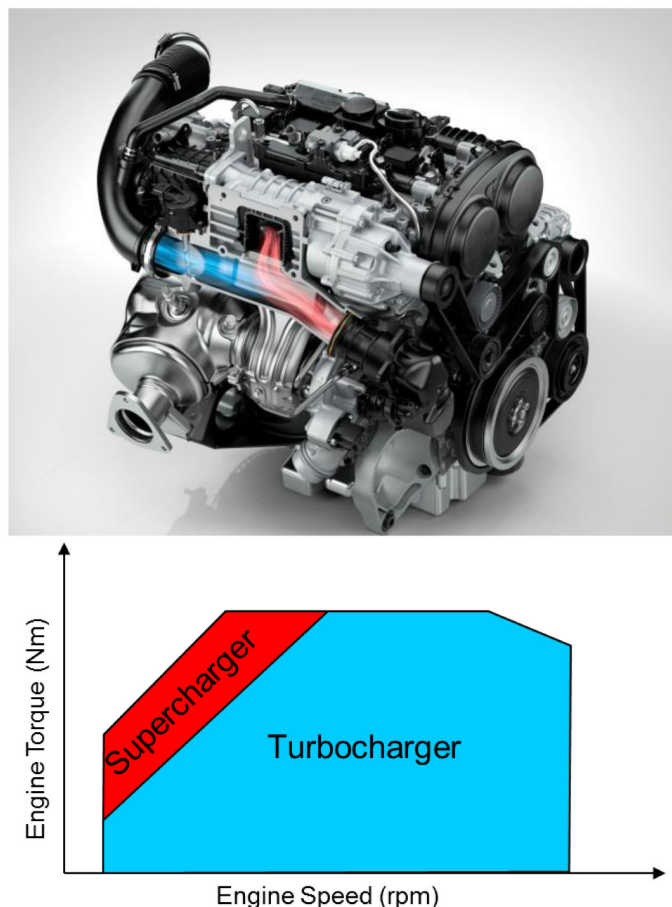


Figure 1. The new VEA twin charger petrol engine with air intake flow through supercharger (red) and bypass (blue).

The inherent design of a roots type supercharger features however significant disadvantages in terms of NVH [3,4]. At both intake and discharge ports pressure pulsations and turbulence are generated which results in very high amplitudes of the acoustic wave propagation in the air intake duct system. The frequency range of the supercharger noise in this application is from 800 to 5000 Hz, determined by operating speed range of the rotors and the number of rotor lobes. This results in complex high frequency wave propagation inside the duct system as the supercharger orders are well above the duct cut-on frequencies. The airborne noise abatement for a roots type of supercharger system may include multiple resonators of both Helmholtz and absorptive type in order to maximize duct insertion loss (IL), acoustic treatment of the supercharger housing and the duct system in order to minimize sound radiation or maximize duct wall transmission loss (TL), and finally considerations to engine bay to passenger compartment airborne noise reduction, i.e. engine compartment absorption, front of dash sound transmission loss (STL) and passenger compartment absorption, see Figure 2. The background masking noise level given by road and wind

noise as well as the remaining powertrain induced noise, i.e. direct and indirect combustion noise and mechanical noise, needs to be incorporated in the refinement as the tonal noise content may be prominent. Structure borne noise abatement is also important, but is not discussed in this paper.

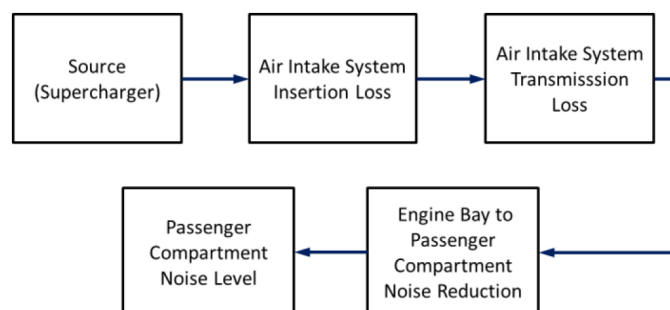


Figure 2. Source-path-receiver model of supercharger noise into passenger compartment.

The methods presented in this paper describes application of high frequency transfer path analysis techniques focusing on complete vehicle to component target cascading for the charging system, where consideration to spatial scatter in measurements are done in combination with objective evaluation parameters as a method to describe the phenomena of tonal noise propagation from the duct system into the passenger compartment. The study is limited to frequency domain airborne noise transfer only due to complexity. The effectiveness of the method, i.e. reaching the targeted noise performance through analysis of component source quantification and noise reduction, is demonstrated together with the effect of remedy integration.

MATERIALS AND METHODS

Transfer Path Analysis

In the automotive industry transfer path analysis is widely used and can hence be considered to be a common tool. The method is in general most often applied to low frequency range NVH issues like powertrain/driveline booming or road noise [5, 6, 7, 8, 9, 10]. Application of transfer path analysis in mid to high frequency NVH can be found in [7, 8, 9, 10]. The method is used to perform comprehensive assessments and create understanding of a noise and vibration phenomenon and then effectively identify and verify remedies for vibration, structure borne and/or airborne noise issues. The method relies on system identification theory of multiple input/output systems. More comprehensive reading can be found in [11,12].

In transfer path analysis the sound pressure at any occupant position is in general the property to be analysed. The output in terms of sound pressure at occupant position i as the objective response, p_i , can be synthesized in the frequency domain as a sum of all structure borne and airborne path contributions (and the residual) as:

$$p_i(f) = \sum_{j=1}^N H_{ij}^{SB}(f) F_j(f) + \sum_{k=1}^M H_{ik}^{AB}(f) \dot{Q}_k(f) + p_{res} \quad (1)$$

where H^{SB} and H^{AB} are the structure borne and airborne acoustic transfer functions, respectively, and F the structure borne operating interface force, $\dot{Q}$ the acoustic volume velocity source and p_{res} the residual (model error). As only airborne noise is of interest in this paper, Eq. (1) can be simplified into:

$$p_i(f) = \sum_{k=1}^M H_{ik}^{AB}(f) \dot{Q}_k(f) + p_{res} \quad (2)$$

In mid to high frequency noise problems, both the magnitude and the phase of the frequency response function varies considerably and rapidly with frequency [7, 8, 9]. In order to have useful estimations of the sound pressure path contributions, a power-based approach for the contribution analysis may be chosen. Eq. (2) will then be redefined into:

$$\tilde{p}_i^2(f) = \sum_{k=1}^M \left| H_{ik}^{AB}(f) \right|^2 \dot{Q}_k^2(f) + p_{res}^2 \quad (3)$$

Instead of using volume velocity as source quantity, the surface radiated noise quantified in terms of sound power may be used. The sound power is proportional to the square of the sound pressure. The following expression was hence used for characterizing the acoustic transfer function (ATF):

$$ATF_{k,i}(f) = \left\langle L'_{p_{k,i}} \right\rangle(f) - L'_w(f) \quad (4)$$

where L'_p is the sound pressure level (in dB) and L'_w the sound power, respectively. The prime dots in Eq. (4) indicate that the quantities are from a substitution source. Sound power implies spatial averaging, hence appropriate for symmetrically radiating sources. Most of the subcomponents may assume to fulfill this criterion since they are homogenous and have symmetric geometries (e.g. tubes). The characterization of the path or ATF between each subcomponent and occupant position may be then derived by the quotient of interior sound pressure squared and sound power at each subcomponent or by the reciprocal procedure [13].

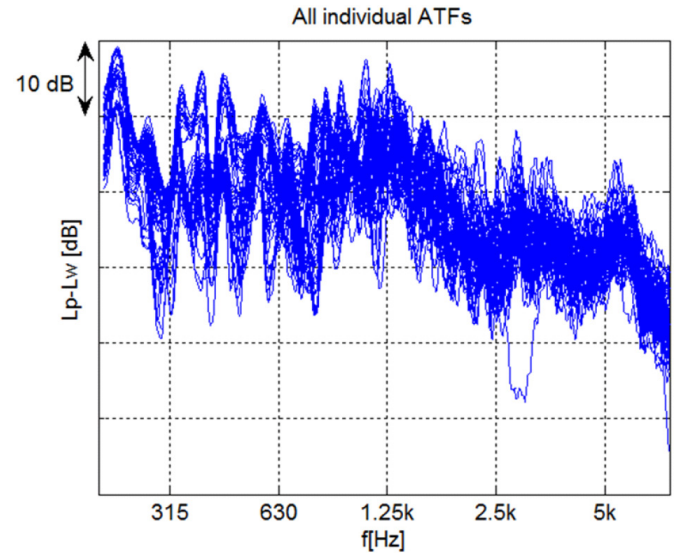


Figure 3. Example of the individual ATFs, measured from one interior position to 54 points located at the surface of a cylindrical electric traction motor [15].

In the frequency range of interest, the ATF magnitude may vary significantly depending on geometric location [14,15], see Figure 3. The ATF used for each subcomponent may also utilize a spatial average of individual ATFs. The number of source microphones may then increase with the size of the subcomponent. Using multiple numbers of equally distributed points is necessary in order to yield a reliable average ATF over the entire subcomponent's surface.

For the measurements, a high frequency omnidirectional source with white noise excitation was adopted. The sound power of the white noise source was previously measured according to the ISO-3745 [16] standard. Assumptions were made that the acoustic loading of the source was not altered significantly between the ATF measurements and the white noise source sound power measurements. During the ATF measurements, the orifice of the high frequency source was roved around at the same receiver positions.

The ATFs were defined in narrow bands ($\Delta f=1\text{Hz}$) with a 12th octave smoothing filter applied. Applying a smoothing filter to narrow band data is a way of reducing the temporary fluctuations in amplitude and frequency to obtain a rather invariable system description of the vehicle's ATF. This approach was successfully used for characterizing the ATFs in a previous study, also concerning contribution analysis for high frequency tonal components [14,15].

The principles for the ATF and operational measurements including the gathered quantities are depicted in Figure 4.