Turbo charger acoustics shifts more into the focus of development due to the increasing application of turbocharged engines. Here the noise behavior of turbochargers leads to a conflict between costs and acoustics. A research program was initiated in order to describe the noise behavior of turbochargers. The research project was assigned by Forschungsvereinigung Verbrennungskraftmaschinen e.V. (FVV, Frankfurt). Systematic experimental investigations were performed and a hybrid simulation methodology was developed. Applying this simulation process, a characterization of the acoustical behavior and the implementation of acoustical measures in the early development phases of the turbo charger can be ensured.

Key Words: Turbo Charger, Acoustics, Simulation / Constant Tone, Whining Noise, Whistle Noise, Unbalance, Measurement, CFD Simulation, MBS Simulation (12)

1. Introduction

The trend to supercharged direct injection gasoline engines and the rising performance requirements of modern supercharged diesel engines implies a set of acoustic risks for the developers. Today the acoustic optimization is based exclusively on empirical models to describe the noise behavior of turbochargers. An efficient simulation-based calculation tool, which can be used in an early stage of the development process, is necessary to describe the extensive mechanism of noise generation and their dependency.

In most cases the disturbing turbocharger noise is not relevant for the overall level of the engine. Although its tonal character is commonly pronounced in a frequency range higher than 500 Hz and thus outside the level-determining frequency range for vehicle interior noise, this leads to an intensified perception by humans. Especially characteristic noise must be regarded in the context of the expectations of the driver.

Under low-speed (<2500rpm) and part load operation such noises could be annoying or even be characterized as malfunction [1]. On the other hand, however, a slight whistling could be perceived quite pleasantly by the driver when the engine is operated at high speed and full load since it is a direct acoustic feedback of the power output of the engine.

This generates a lot of requirements on a development tool whereby the different gas and rotor dynamic excitation sources of the turbocharger need to be represented properly in the complete speed range. In addition, the structure-borne noise propagation over the involved engine components has to be considered.

In order to develop a tool to predict the acoustic behavior of a turbo charger this paper describes a two-stage approach: first, measurements were conducted while doing extensive parameter variations on the turbochargers (TC) of a passenger car and a heavy duty engine in order to characterize the different noise phenomena caused by the particular applications and to quantify the individual dominating factors of the acoustics of the turbocharger. Secondly, a hybrid computation method has been developed based on flow, multi-body and dynamic finite-element simulation. The simulation results obtained with the chosen approach correlates very well with the experimental data. Thus the individual results will contribute significantly to

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further understanding of the complex acoustic mechanisms of a turbocharger and allows a virtual layout of the turbocharger system.

2 Experimental Investigation

In a first step experimental investigations were performed in order to identify the driving parameter for the noise phenomena of turbocharger (especially constant tone, unbalancing whistle, pulsation and rotating noise) and to acquire verification data for the simulation models.

2.1 Test engines

The experimental investigations were performed using either a passenger car engine and a truck engine. Basically, acceleration measurements at the turbocharger, measurement of the pressure pulsations in the exhaust and intake system, and airborne noise measurements under anechoic conditions were performed.

2.2 Noise phenomena

The analyzed noise phenomena of the investigated turbochargers can be divided in four acoustical phenomena caused by different physical effects. These four phenomena can be perceived as noises of a tonal character.

Unbalancing Whistle

Because of high rotor speeds, large unbalancing forces can occur in turbocharger systems. These rotating forces are linearly proportional to the frequency of the turbocharger-speed and cause a tonal noise, commonly known as unbalancing whistle [2].

Pulsation Noise

The pulsation noise occurs with the rotation frequency of the turbocharger rotor. Physical causes of this phenomenon are rotor asymmetries, which lead to alternating forces when the irregularities are rotating relative to the non-moving housing. This noise effect is mainly radiated by the air ducts which are excited by pressure fluctuations.

Rotating Noise

The rotating noise is caused by the pressure difference between pressure and suction side of the particular rotor blades and by the blades passing the housing tongue. The frequency of the excitation is determined by the product of the TC-speed and the number of blades of the turbine respectively compressor wheel [3]. Because of the high turbocharger-speed of the passenger car TC, its rotating noise lies mainly above the audible range. Therefore this phenomenon plays a subordinate role in this investigation. In contrast to that, the essential larger TC of the truck engine generates rotating noise in the frequency range up to 13 kHz due to its lower rotation speed. Here, the rotating noise can be clearly perceived under full load condition.

Constant-Tone

The constant-tone is different from the other three noise phenomena, because its related frequency is not proportional to the turbocharger-speed. After a slight increasing slope at low TC-speed the phenomenon is associated with a constant frequency versus rotorspeed. This frequency is defined by the lateral bending vibration of the TC-shaft in the hydrodynamical oil film of the journal-bearings of the rotor. For the passenger car TC typical values are in the range of 1 kHz, and for the truck TC around 500Hz, as shown in Figure -.
2.3 Influencing factors

During the experimental investigations several parameter were varied in order to determine the most influencing factors for the individual noise phenomena. Figure -1 and show the dependency of the constant tone and the unbalancing whistle on the unbalancing of the moving parts and on the oil temperature. The displayed acceleration level were measured at the compressor side of the TC at 200,000 rpm.

Influence of the unbalance

With grub screws on the compressor side an unbalance was added to the remaining unbalance caused by already existing manufacturing tolerance ranges. Two turbochargers with an additional mass of 20mg and 40mg were tested. As expected, Figure -1 shows a clear dependency of the unbalancing whistle level to the applied mass. Here, the level of the first order follows the unbalance in a linear rising shape. In contrast to that the acceleration level of the constant-tone decreases significantly with a unbalance mass higher than 20mg.

![Figure -1: Influence of unbalance weight on 1st order and constant tone acceleration level, passenger car turbo charger at 200 000 rpm](image)

Influence of the oil temperature

To investigate the influence of the oil temperature and therefore the oil viscosity on turbocharger noise, the oil temperature was changed in the range from 30 °C to 90 °C (Figure -2). The level of the first turbocharger order and the constant-tone show a graduate reduction with increasing oil temperature. Besides the displayed influence on the first order level, the higher oil temperature also leads to a small shift of the constant-tone to lower frequencies. This effect can be explained by the altering damping effect of the hydrodynamic bearing on the natural vibration behavior of the TC-rotor, due to the oil viscosity variation.

![Figure -2: Oil temperature influence on 1st order and constant tone acceleration level, passenger car turbo charger at 200 000rpm](image)

3 Hybrid simulation method

In order to analyze the mechanisms of noise generation and to investigate their influence by performing detailed parametric studies, a multidisciplinary CAE model was developed. The structure and the interfaces between the separate calculation methods are depicted in Figure -3.

![Figure -3: Principal workflow](image)
3.1 CFD Simulation

Based on a three-dimensional gas dynamic simulation (CFD), the gas-flow-dynamics in turbine and compressor is calculated using the software ‘Star-CD’. Target is to investigate the influence of a deviation of the turbine and compressor wheel on the pressure excitation of the gas flow, and on the resulting gas forces acting on the wheels. Furthermore, the gas forces are used as an input for the multi body simulation in order to calculate the structure excitation caused by the gas flow. The wheel deviation can be separated into a purely parallel offset in the bearing clearance, caused by unbalance, and into an angular deviation, caused by bending of the rotor. Both deviations are rotating with the rotor speed, as is depicted schematically in Figure 5. The flow region covered by the CFD calculation contains scroll, wheel, inlet and outlet of compressor and turbine respectively. The computational grids consist of about 1 million cells for the compressor and 3 million cells for the turbine (Figure -4).

Steady state simulations of particular points of operation within the performance map show a good correlation to the measured maps of turbine and compressor (Figure -4). The flow field at the impeller outlet of the compressor exhibits a clear dependency on the back pressure. With increasing back pressure the mean outlet mass flow is shifted towards the start of the scroll, which leads to an uneven load and thus evokes a radial force on the wheel.

Transient flow simulations were performed to resolve the pressure and force oscillation for the acoustic assessment. The motion of the wheels in terms of rotation was realized by a moving mesh. At the borders of the area considered for the calculation, pressure and mass flow rate traces as well as temperatures are applied as boundary conditions. These boundaries were taken from one dimensional gas exchange calculations.

The evaluation of the simulation results for the compressor, comparing a centered arrangement with two different rotor deviations is shown in Figure 5. The force and pressure for the centered rotor show a clear coherence between radial force on the impeller and pressure after compressor p2, which can be explained by the shift of the mean outlet flow as described before. An excitation of the 1st turbocharger order is shown with an amplitude of about 1 N, gained from the radial forces gained from the simulation of the variants with eccentric rotor. Here, the mean trend follows exactly the one for a centered rotor. From this, a complete separation of the force excitation caused by boundary conditions and wheel deviation can be inferred (Figure 5 bottom). The excitation amplitude due to eccentricity shows only a marginal effect of the boundary condition, but depends linearly on the amount of deviation. However, the force excitation caused by the gas flow is negligible low in comparison to the dynamic bearing forces calculated by the multi body simulation.

The post-compressor pressure characteristics, p2, are acoustically more remarkable than the radial force. Similar to the description of the radial force, a wheel deviation causes an excitation of the first turbocharger order which is superimposed on the pressure trace...
induced by the engine operation. Also here, the deviation length has a significant impact, as is shown in Figure 5 on the right side. A deviation of 100 % corresponds to the full bearing clearance. The variant with a bended rotor (red bar) shows a slightly different behavior. The eccentricity “\(e\)” of the impeller, which is even larger than with purely parallel deviation, also causes a pressure excitation. This is, however, clearly reduced by the additional angle “\(\alpha\)”. Reason for that is the tip clearance between rotor blades and housing, which becomes larger within the range of the rotor outlet due to this angle. At the compressor especially this region plays an important role for the load transfer onto the gas flow, so that the increased gap leads to a decrease of the pressure excitation.

A similar correlation is found on the turbine side, between impeller eccentricity and the excitation of pressure and force as at the compressor. Differently than with the compressor the bending angle “\(\alpha\)” has almost no effect on the turbine side, so that an additional eccentricity due to rotor bending directly increases the amplitude of the pressure excitation. In total the amplitudes of force and pressure pulsation on the turbine side are significantly lower than for the compressor.

It can be summarized that an impeller eccentricity primarily leads to airborne noise dominated by the 1st turbocharger order, whereas the level of the excitation by the forces acting on the rotor are negligibly small. On the compressor side, which is clearly more susceptible to this phenomenon, the noise excitation is again reduced by a bending of the rotor.

3.2 MBS Simulation

The intention of an MBS-simulation is to simulate the system behavior of coupled bodies. As a result, forces, moments, displacements, velocities and accelerations can be evaluated to analyze the system behavior. The rotor is a crucial element of the investigated system. As already mentioned, the rotor rotates at very high speeds, which leads to high mass forces. In the used commercial software ADAMS/Engine, bearing elements exist with hydro-dynamical properties, which solve the Reynolds equation considering bearing-geometry, bearing-clearance, temperature and oil viscosity for each simulation time step [4]. A specialty of the considered turbocharger is the floating bush bearing of the rotor. This bearing features two hydro-dynamical bearings, which are coupled by a floating bush. The rotor as well as the housing is integrated in the model as a flexible body. In Figure 6 the principle set-up of the MBS model is depicted.

A run-up simulation is performed from 0 to 240,000 rpm with the baseline passenger car turbo charger, including a remaining unbalance at the rotor. The system behavior will be analyzed based on the displacement of the rotor relative to the housing. For identifying the phenomena, a conversion of the time signals into the frequency domain depicted in a Campbell diagram has proved to be the best way to evaluate the simulation. When considering the rotor displacement two significant phenomena can be observed:

1. A dominant first rotor order, which is caused by the unbalance of the rotor.
2. A half order, which changes into a constant frequency, also known as constant-tone. The root cause mechanism of the constant-tone is based on two different effects: in the half order-dominating speed range up to 60,000 rpm the rotor rolls within the clearance of the hydrodynamic bearing as a rigid body (swirls). This roll speed is affected by the characteristics of the oil and the axial clearance. This roll speed is clearly lower than the rotational speed of the rotor. The transition to the resonant frequency is initiated when the roll speed coincide with the bending-critical over-speed of the rotor. In Figure -7 the displacement behavior of the shaft is represented, where the time slice shows a different behavior of the two shaft ends. A bending of the shaft on the turbine side dominates the behavior concerning the constant-tone, while on the compressor side the unbalance causes unbalancing whistle.

It can be summarized that an impeller eccentricity primarily leads to airborne noise dominated by the 1st turbocharger order, whereas the level of the excitation by the forces acting on the rotor are negligibly small. On the compressor side, which is clearly more susceptible to this phenomenon, the noise excitation is again reduced by a bending of the rotor.
To verify the MBS model calculations of different variants were performed, and compared with the corresponding measurement results. One of the variants was a reduction of the oil viscosity which was examined by increasing the oil temperature. Both, measurement and calculation show a reduction of the constant-tone frequency. Furthermore, the influence of a larger unbalance of 40 mgmm on the compressor side was investigated. The result is a significant increase of the first TC-order. Simultaneously a reduction of the constant-tone level can be observed. Both noise phenomena are well pronounced, as the comparison with the measurements displays in Figure -5. This proves the high quality of the simulation model.

![Figure -5: Verification due to unbalancing whistle](image)

Further calculation steps were performed in order to reach a minimization of the individual noise phenomena without a degradation of the responsiveness of the rotor itself. Table 3-1 shows an overview of performed calculations and the corresponding results. However, the feasibility of some variants must be investigated regarding further criteria.

<table>
<thead>
<tr>
<th>Constant tone</th>
<th>increased oil viscosity</th>
<th>decreased oil viscosity</th>
<th>decreased bearing distance</th>
<th>increased shaft diameter (ring bearings)</th>
<th>decreased shaft diameter (hub bearings)</th>
<th>decreased weight of blade wheels</th>
<th>40 mgmm unbalance @ comp. side</th>
<th>15 mgmm unbalance @ turb. side</th>
<th>15 mgmm unbalance @ turb. and comp. side</th>
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<td>Unbalance 15 mgmm</td>
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Table 1: Overview of performed modifications

3.3 FE calculation

The last step in the interdisciplinary concept is the finite element simulation with the commercial software Ansys. The gas forces extracted from the CFD analysis and the bearing reaction forces from the MBS simulation are loaded on the housing structure and a forced-response calculation is performed. As a result the structural answer and the associated surface mobility are obtained.

For evaluating the phase relationships of the individual force components, the calculation takes place in the time domain. A first step in this calculation methodology is the modal analysis, which determines the natural frequencies of the structure. The natural frequencies for this model were calculated up to 4.5 kHz. In the second step the forces and moments extracted from the MBS simulation including the gas forces of the CFD analysis are loaded in the corresponding areas of the turbocharger housing, and finally the structure answer was calculated.
Apart from the occurrence of the phenomena the influence of bearing specification can be described with such an approach. Finally, the presented simulation method enables an acoustically refined design of the turbocharger in the early development process.

REFERENCES


Summary

A fundamental understanding of turbocharger noise phenomena was obtained by the described analyses of experimental testing and calculations. An interdisciplinary concept was developed, in order to represent the complex excitations and transfer paths sufficiently in detail. This concept is based on a combination of the simulations of flow dynamics (CFD), rotor dynamics (MBS) and dynamic structure transfer (FEM). The quality of the calculation model was verified with measurements obtained from the complete engine of a small-passenger vehicle and a commercial vehicle.