PREDICTION AND CONTROL FOR WHOOSH NOISE OF A TURBOCHARGED ENGINE

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Turbocharger noise, especially broadband whoosh noise, is a difficult problem in vehicle noise vibration and hardness (NVH) development. To solve this issue, a prediction and control method is proposed in this paper by researching generating mechanism of the whoosh noise based on an intake system of a four-cylinder turbocharged engine. First, a computational fluid dynamics (CFD) model is built with polyhedral meshing and K-Omega turbulent model. By calculating the model, the pressure and flow rate distributions of a bypass pipe on the intake system is obtained under blowoff conditions. In order to predict whoosh noise, the Proudman aeroacoustic model is simulated based on a broadband noise source. Furthermore, an optimized design of the bypass pipe is achieved. Second, a high-frequency Helmholtz resonator installed on the intake pipe is designed to attenuate the whoosh noise. Last but not least, an electronic fuel injection (EFI) strategy is modified to open the blowoff valve in advance to achieve the designed vehicle NVH performance. The vehicle experiment result shows that optimization of the bypass pipe, the design of the high-frequency Helmholtz resonator and the EFI strategy can provide an effective method for controlling the whoosh noise, which also provides a reliable reference for solving the turbocharged noise issues of turbocharged engines.

Keywords: Turbocharger; Whoosh noise; NVH; Prediction; Control

1. Introduction

As more and more vehicle power and fuel economy are demanded in the market, the turbocharged automobiles gradually become mainstream products. However, after the turbochargers are used in the passenger cars, many supercharger NVH issues will arise, especially the whoosh noise. The whoosh noise is mainly generated in transient driving conditions such as climbing, accelerating and so on. The air intake system experiences a rapid airflow pressure and velocity changes at this moment. The pressure difference across the air intake system during the transient events changes the flow direction in the system. At the moment of release of the gas pedal, the throttle is closed, so the gas pressure inside the intercooler pipe increases rapidly. In order to avoid surge and damage of the supercharger, the blowoff valve (Fig.1 D location) is opened, the compressed air is released to the outlet pipe of the air clean box quickly through the bypass pipe and forms a return flow (shown as Fig.1). The high-speed airflow mixed rapidly with the airflow from the intake outlet pipe (shown as Fig.1 D location), which produces strong pulsating turbulence. The noise produced by this turbulent is the whoosh noise.

Evans, D et al. [1] investigated the whoosh noise at acceleration conditions on some European Diesel engines in 2005, and the research results showed that the whoosh noise mainly transmitted to the vehicle interior through the intake pipe could be reduced with a broadband resonator and i-
creasing thickness of the intake pipe. Charlie, T. and Steve H. [2] studied the whoosh noise and the results showed that altering the compressor trim vale, improving flow conditions at the compressor inlet and engine system calibration, adding air inlet resonators on both the low and high pressure sides of the compressor effectively improved the whoosh noise of a v6 engine in 2009. Vivek Kolhe et al [3] designed the multi-chamber silencers on the outlet pipe of the air clean box and the inlet of the intercooler to reduce multiple whoosh noises in 2013. Kuang, X. et al. [4] explored how to identify the surge noise through intercooler inlet temperature, pressure and other parameters, and proposed to use a parallel 1/4 wavelength group-tube to reduce the surge noise. The research results of the above scholars provide a good reference for the whoosh noise control in this article, however, none of their research results suggest a method for predicting the whoosh noise.

![Figure 1: Mechanism of whoosh noise generation](image1)

The whoosh noise is a kind of fluid dynamic noise generated by the intake system. According to Lighthil acoustic analogy theory, sound generating mechanism of the fluid-dynamic noise source and its corresponding physical model (shown as Fig.2), the aerodynamic sources can be divided into the following three categories. The first category is a monopole source, the second category is a dipole source, and the third category is a quadrupole source.

In 1955, Curle [5] applied the Kirchhoff integral method to extend the Lighthil [6,7] equation to an equation containing solid boundaries in a fluid. His research showed that the action of solid boundary can be equivalent to the monopole and dipole sources distributed on its boundary. The magnitudes of the monopole and dipole sources are related to the expansion and compression action and exerted force between the fixed surface of the source point and the adjacent fluids. In 2012, Zhang Qiang [8] derived the general non-homogeneous wave equation for sound pressure of aero-dynamic noise, as shown in Eq. (1).

\[
\frac{\partial^2 p}{\epsilon_0 \partial t^2} - \nabla^2 p = \frac{\partial Q(y,t)}{\partial t} - \frac{\partial F_i(y,t)}{\partial x} + \frac{\partial^2 T_{ij}}{\partial x_i \partial x_j}. \tag{1}
\]

In Eq. (1), the right items are monopole source, dipole source and quadrupole source respectively, the monopole and dipole sources are surface noise sources, which only appear on the surface of the moving object. The monopole and dipole sources are zero at the areas far away from the surface. The quadrupole source is a volume source that only appears in a turbulent spatial area, and the quadrupole item’s \( T_{ij} \) is the Lighthill turbulent stress tensor.

\[
T_{ij} = \rho u_i u_j - \tau_{ij} + \delta_{ij}(p - c_0^2 \rho'). \tag{2}
\]

where, \( \rho u_i u_j \) is the turbulent stress associated with turbulent pulsation velocity, and it often referred to as Reynolds stress. \( \delta_{ij}(p - c_0^2 \rho') \) is the effect of heat conduction. \( \delta_{ij} \) is Kronecker symbol, and its impact can be ignored in the absence of heat conduction effect. \( \tau_{ij} \) is a viscous tangential stress with thermal dissipation properties, and it is too small to be considered compared with the Reynolds stress. Therefore, \( \tau_{ij} \) can be considered as only related to Reynolds stress, that is, \( \tau_{ij} = \rho u_i u_j \). The magnitude of the Reynolds stress depends on the local turbulent conditions. The quadrupole source is located in the turbulent flow of the fluid.
According to the above study, the whoosh noise is mainly the turbulent stress source noise generated by the intake airflow and the deflating airflow at the mixed location, and the noise generated by the interaction between the airflow and the wall of the outlet pipe of the air clean box.

A computational fluid dynamics (CFD) model of an intake system is built with polyhedral meshing and K-Omega turbulent model in this paper. By calculating the model, the pressure and flow rate distributions of a bypass pipe on the intake system are obtained. In order to predict the whoosh noise, the aerodynamic sound source is simulated based on the Proudman acoustic model. A method to control the whoosh noise is proposed by optimizing the vehicle EFI and the sound transmission loss of the air intake system.

2. Prediction of whoosh noise based on Proudman acoustic model

2.1 Whoosh noise issues of a passenger car

For most turbocharged automobiles, the whoosh noise is easily perceived under transient driving conditions such as climbing and accelerating. In order to reduce structural modifications of the intake system in late period, it is necessary to predict the whoosh noise in the early stage of vehicle development. This article takes the intake system (shown as Fig.3) of a 7-seats Multi-Purpose Vehicle (MPV) installed a 4-cylinder turbocharged engine as the research object. The blowoff valve is arranged apart from the turbocharger. The whoosh noise generated at the interaction between the bypass pipe and the outlet pipe transmits to the passenger compartment mainly through the airborne path. Fig.4(a) and (b) are the colormaps of the whoosh noise at the inlet orifice and inside the compartment, respectively, and its corresponding frequencies are between 1000 Hz and 2500Hz. This broadband noise (as shown in Fig.4) is very easily identified and perceived, and it is difficult to completely eliminate the noise by the vehicle sound insulation and absorption, but the issue needs to be solved immediately.

This paper predicts the magnitude of the whoosh noise using 3D-CFD simulation based on the Proudman acoustic model. The main characteristic parameters of the whoosh noise are deflation pressure gradient, deflation velocity, spatial Proudman sound source distribution. According to engineering experience, when the deflation flow rate is less than 110m/s and the Proudman sound pressure level is less than 115dB, the whoosh noise will be not generated.

2.2 Proudman acoustic model

In 1952, Proudman, I. [9] researched the noise generation by isotropic turbulence using statistical models of various two-point moments and Lighthill analogy theory. In 1993, based on Proudman's high Reynolds number model, Lilley [10] added the effects of retarded time, and obtained turbulent sound power of per unit volume, expressed as follows.

$$ W = \alpha_c \rho_0 \frac{U^3}{L} \frac{U^3}{\alpha_l^3}. \quad (3) $$

where, $ U = L/T$, $ \alpha_c = 0.629$, $ \rho_0$ is the air density, $ U$ is turbulence speed, $ L$ is turbulence length, $ T$
is turbulence duration, $a_0$ is the sound speed.

Convert the sound power of the per unit volume to dB level,

$$W(\text{dB})=10\log\left(\frac{W}{W_{\text{ref}}}\right).$$

where, $W_{\text{ref}}$ is reference sound power.

### 2.3 CFD model of air intake system

Before CFD simulation modelling, the assumptions and simplifications need to be made according to actual needs as follows:

1) In order to reduce the amount of calculation, the intercooler, intercooler inlet pipe, intercooler outlet pipe with less influence on the model calculation are removed. 2) The turbulence is isotropic, and the intake airflow is ideal gas and incompressible. 3) The filter is simplified as a regular cube that matches the filter element size and position. 4) The outlet is extended to a certain length that is suggested to be about five times length of the outlet pipe’s diameter.

The 3D model of the intake system with polyhedron elements is meshed after the simplifications. The meshing quality is checked. The detailed settings of meshing parameters are listed in Table 1 and the intake system fluid meshing is shown in Fig. 5.

Table 1: Meshing parameters of whoosh noise analysis

<table>
<thead>
<tr>
<th>Meshing type</th>
<th>polyhedral</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base surface mesh size</td>
<td>2mm</td>
</tr>
<tr>
<td>Number of boundary layers</td>
<td>3</td>
</tr>
<tr>
<td>Boundary layers total thickness</td>
<td>0.3mm</td>
</tr>
<tr>
<td>Boundary Layers Stretching</td>
<td>1.5</td>
</tr>
</tbody>
</table>

The boundary conditions of the air intake system CFD model are listed in Table 2. The physical parameters of the air intake system CFD model are listed in Table 3. The CFD model of the intake system is basically completed after finishing the intake system meshing processing, and applying boundary conditions, and loading physical model.

Table 2: Boundary conditions of whoosh noise analysis

<table>
<thead>
<tr>
<th>Mass flow rate</th>
<th>183.9kg/h</th>
</tr>
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<tbody>
<tr>
<td>Air inlet temperature</td>
<td>25°C</td>
</tr>
<tr>
<td>Air outlet pressure</td>
<td>-4.8kPa</td>
</tr>
<tr>
<td>Gas compressor outlet temperature</td>
<td>125.1°C</td>
</tr>
<tr>
<td>Gas compressor outlet pressure</td>
<td>0.8773bar</td>
</tr>
</tbody>
</table>

Table 3: Physical parameters of whoosh noise analysis

<table>
<thead>
<tr>
<th>Air</th>
<th>Ideal gas</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbulent model</td>
<td>SST K-Omega</td>
</tr>
<tr>
<td>Wall function</td>
<td>Two-layer all y+ wall treatment</td>
</tr>
<tr>
<td>Pressure-velocity couple model</td>
<td>Segregated flow &amp; segregated fluid temperature</td>
</tr>
<tr>
<td>Wall temperature</td>
<td>Adiabatic wall boundary condition</td>
</tr>
<tr>
<td>Aeroacoustics model</td>
<td>Broadband noise sources/Proudman</td>
</tr>
</tbody>
</table>

### 2.4 Whoosh noise prediction based on CFD model

Figures 6-9 show the pressure distribution, velocity distribution, streamline distribution, and Proudman sound source distribution, respectively, which characterize the whoosh noise of the air intake system.

The analysis shows that the high-pressure area is mainly concentrated before the blowoff valve, and there is no high pressure area after the blowoff valve from the pressure distribution. From the pressure slice distribution at the junction of the bypass pipe and the outlet pipe of the air clean box,
the pressure gradient is small (shown in Fig.6). The maximum flow rate at the blowoff valve is 294.3 m/s, and the maximum flow rate at the interaction of bypass pipe and the outlet pipe is 126 m/s (shown in Fig.7) that exceeds the acceptable value of 110 m/s by engineering experience. The vortex under the air clean box is serious, but after the filter, the vortex does not exist at the upper shell. There are some local vortices at the interaction of the bypass pipe and the outlet pipe (shown in Fig.8). From the Proudman source distribution (shown in Fig.9), the maximum sound pressure level at the noise source is 146.7 dB at the outlet of the blowoff valve, the sound pressure level at the interaction of the bypass pipe and the outlet pipe is 125 dB that exceeds the acceptable value of 115 dB by engineering experience. Based on the pressure gradients, flow rates, and Proudman sound source values, the whoosh noise could bring a serious noise problem.

3. **Source and path control of whoosh noise**

The control of whoosh noise mainly includes source noise control and path control. As shown in Fig.10, the source noise control of the whoosh noise is to control turbocharger’s operation, including EFI design, using large pressure impeller and vortex intake structure, optimizing leaf structure, and so on. The path control consists of two parts, one is noise elimination control by using high-frequency Helmholtz resonator, combined wavelength tubes, braided tubes and so on; the other one is noise insulation and absorption control by choosing material of intake pipe, increasing thickness of intake pipe, and vehicle sound insulation.

The turbochargers are tested on the test bench by the suppliers and then its design is finalized before installed in the vehicle. It is too late to modify the turbocharger structure if there exists the whoosh noise. The effective way to reduce the whoosh noise is to optimize EFI data in acceptable tuning range, and to use the high-frequency Helmholtz resonator and braided tube in the path control.
3.1 Deflation flow control based on vehicle EFI

According to the Proudman acoustic model, the deflation sound power is proportional to the \(8^{th}\) power of the turbulence velocity. Therefore, the flow rate is the major contributor to the whoosh noise. The purpose of the EFI control is to open the blowoff valve as early as possible without affecting the dynamic performance of the vehicle, to reduce deflation pressure and deflation flow rate to reduce whoosh noise.

The opening and closing of the blowoff valve depend on whether it will generate the surge problem. In order to complete vehicle calibration, the surge line tested on the bench is converted to the vehicle calibration value and then input into the vehicle ECU. First, the mass flow is converted to volume flow through density conversion. Second, the converted value is input into the vehicle ECU and the vehicle surge line is optimized according to the vehicle dynamics and driving calibration.

As shown in Fig.11, the practical surge line provided by the turbocharger manufacturer is mainly determined by the turbocharger structure, but the virtual surge line is determined by vehicle calibration. Generally speaking, the adjustable range of the surge line is about 10%. The EFI control of the whoosh noise is mainly to optimize the intake flow and pressure ratio, to modify the virtual surge line to achieve opening of the blowoff valve in advance, to reduce the pressure, and furthermore to reduce the whoosh noise from the source. The conditions of opening the blowoff valve are to force the pressure after the turbocharger or flow rate to close to the surge line.

Figure 11: Turbocharger characteristics diagram

As shown in Fig.12, generally speaking, the opening of the blowoff valve and the closing of throttle is not synchronous, and the delay time of the opening is about 0.5 to 0.71 seconds. The pressure after the turbocharger will increase instantly because of the opening delay. The airflow rate and the magnitude of the whoosh noise will rapidly increase. Therefore, the best way to control the whoosh noise is to open the blowoff valve in advance. At the same time, the vehicle testing shows that the whoosh noise can be significantly reduced by controlling the blowoff valve according to the predicted flow rate. According to the turbocharger characteristics (Figure 11) and the control model of the blowoff valve (Figure 12), a schematic to open the blowoff valve in advance by flow rate control is shown in Fig.13. It can be seen from the figure that compared with the actual flow rate line, the predicted one declines in advance and touches the surge line earlier, so the blowoff valve is opened, the deflation pressure is released, the deflation velocity is reduced and the whoosh noise is attenuated.

![Figure 12: Control model of blowoff valve](image)

![Figure 13: A schematic to open blowoff valve in advance by flow rate control](image)

Fig.14 Flow rate data comparison before and after EFI optimization comparison

Fig.15 Whoosh noise sound pressure level before and after EFI optimization

Fig.14 shows the results by using the predicted flow rate to control whoosh noise, \(t_1\) is the time to open the blowoff valve in the initial design, \(t_2\) is the time to open the blowoff valve after optimizing the EFI, \(t_1-t_2=0.1s\), that is, the blowoff valve is opened 0.1s in advance with using the pre-
dicted flow rate. Fig.15 shows the tested noise at the deflation vent before and after optimizing the EFI. It can be seen that compared with the original design, the whoosh noise is reduced about 5dB(A) by the optimized design. In the subjective evaluation, the whoosh noise is attenuated obviously, while there are no impacts on power performance and driving performance.

3.2 Whoosh noise control based on transmission loss optimization

The path control of the whoosh noise mainly contains noise elimination control and noise insulation and absorption control. Usually, the most direct and effective way is to optimize the sound transmission loss and increase the noise reduction of the air intake system. So a high-frequency Helmholtz resonator is installed on the air intake system to reduce the whoosh noise.

The test results shows that the closer the high-frequency Helmholtz resonator to the noise source, the better the insulation effect. It is better to install the high-frequency Helmholtz resonator on the outlet pipe of the air clean box and close to the bypass pipe. However, the high-frequency Helmholtz resonator is forced to be installed inside the air clean box because of the limited engine compartment space of the MPV shown as Fig.16(b).

According to the frequency range of the whoosh noise, 1000 to 2500Hz, the high-frequency Helmholtz resonator with 0.5L volume is designed. Fig.16 shows the air intake system before and after installing the resonator.

The 3D finite element method is used to calculate the transmission loss of the high-frequency Helmholtz resonator, to optimize the volume of each chamber, and to obtain the number and diameter of the orifices on the pipe. The boundary conditions for the model are that an anechoic end is at inlet pipe and 10W sound power energy is applied at the outlet pipe [11].

The sound transmission loss (noise reduction) is predicted using following formula [12].

$$TL = 20 \log_{10} \left( \frac{P_{in}}{P_{out}} \right)$$

where, $P_{in}$ is the pressure at the outlet pipe, $P_{out}$ is the pressure at the inlet pipe.

As shown in Fig.17, the sound transmission loss increases about 20dB from 1250 to 2500Hz after the resonator is installed, which achieves a good noise attenuation performance.

3.3 Test result of whoosh noise by combination control

A combined control including EFI control and the resonator is applied on the vehicle. Fig. 18 (a) and (b) shows the noise at the outlet pipe and the vehicle interior, respectively for the original system. Fig. 19 (a) and (b) shows the noise at the outlet pipe and the vehicle interior, respectively for the combined controlled system.
FIG. 19 Noise test of optimized air intake system and EFI on vehicle

Both the noises at the outlet and the interior are significantly attenuated by the EFI control and the resonator combined. Subjectively, the whoosh noise is not perceived by the evaluation team.

4. Conclusion

The conclusions are obtained by the study of generation mechanism, prediction and control methods of the whoosh noise in this paper.

1) The CFD analysis method can be used to obtain the pressure, velocity, streamline and Proudman noise source distributions of the air intake system, to predict the whoosh noise effectively and to propose reliable optimization measures.

2) An EFI control strategy to control the blowoff valve by the predicted flow rate the high-frequency Helmholtz resonator installed on the intake pipe are used to attenuate the whoosh noise from the source.

3) The test results show that the whoosh noises from the bypass pipe and the vehicle interior noise are greatly improved by optimizing the air intake system and the EFI data. The optimized whoosh noise cannot be perceived by the most drivers and passengers.

REFERENCES