Yansong Wang
Hui Guo
Chao Yang

Vehicle Interior Sound Quality

Analysis, Evaluation and Control

Springer

Vehicle Interior Sound Quality

Yansong Wang · Hui Guo · Chao Yang

Vehicle Interior Sound Quality

Analysis, Evaluation and Control

Springer

Yansong Wang
School of Mechanical and Automotive Engineering
Shanghai University of Engineering Science
Shanghai, China

Hui Guo
School of Mechanical and Automotive Engineering
Shanghai University of Engineering Science
Shanghai, China

Chao Yang
School of Mechanical and Automotive Engineering
Shanghai University of Engineering Science
Shanghai, China

ISBN 978-981-19-5578-5 ISBN 978-981-19-5579-2 (eBook)
https://doi.org/10.1007/978-981-19-5579-2

This Springer imprint is published by the registered company Springer Nature Singapore Pte Ltd.
The registered company address is: 152 Beach Road, #21-01/04 Gateway East, Singapore 189721, Singapore

Preface

The new generation of road vehicles is developing toward electrification, intelligence, interconnection, and sharing, and besides being a means of transportation, vehicles are expected to have more functions, such as work and entertainment. In line with these trends, the vehicle interior sound quality control calls for more attention than just controlling physical acoustic quantities. The last decade has witnessed revolutionary progress in the materials, structures, control methods, and technologies that create a quieter and more comfortable interior sound environment for vehicles. The existent challenge is that the interior noise in the running vehicle, especially under high-speed conditions, features strong time-varying and nonlinear characteristics. New challenges are also emerging for interior noise control, even sound quality, when different functional zones are expected in the future cabin, like the working or entertainment zone.

The book aims to provide readers with theoretical and engineering advances in vehicle interior sound quality control. Postgraduate students majoring in automotive engineering may use it as an independent study text, and engineers, designers, researchers, and educators may use it as a reference book.

Researchers and engineers have investigated vehicle sound quality in the past few decades. However, there has been a lack of a book to integrate the analysis and evaluation approaches and control algorithms that provides a means for approaching the concepts of vehicle interior sound quality and give readers a general picture and framework on the subject. Existent publications are only papers, books, or chapters containing relevant exploration of particular systems.

Absorbing the authors' various achievements in their relevant work as a teacher, researcher, designer, and engineer, the book covers wide vehicle sound quality issues systematically: the mechanism of vehicle noise generation, propagation of elastic waves from vibration and noise source, subjective and objective evaluation methods on stationary and non-stationary noise, passive and active sound quality control technologies, and hybrid vibro-acoustic control methods. These issues are addressed by providing depth and breadth to the mechanism study, characteristics analysis, novel control algorithms, and evaluation system for vehicle interior sound quality.

The following projects strongly support the relevant research in this book: the Project of the National Natural Science Foundation of China, the Program for Professor of Special Appointment (Eastern Scholar) at Shanghai Institutions of Higher Learning, Program of Shanghai Academic/Technology Research Leader, and Technical Service Platform for Noise and Vibration Evaluation and Control of New Energy Vehicles at Science and Technology Commission of Shanghai Municipality, China.

Vehicle interior sound quality is an interdisciplinary research field across physical acoustics, physiological acoustics, and psychoacoustics. The authors are motivated to promote the new technology of the topic, novel mechanisms and new materials for vibration and noise reduction, and original mathematical models and control strategies for a better sound quality environment. Errors are inevitable in this book due to the continuous development of vehicle interior sound quality and the authors' limitations. Any comments from readers are appreciated.

Shanghai, China
July 2022

Yansong Wang
Hui Guo
Chao Yang

Acknowledgments

This book is honorably funded by the Project of the National Natural Science Foundation of China (No. 52172371, 51675324 & 51175320) and partly supported by the Program for Professor of Special Appointment (Eastern Scholar) at Shanghai Institutions of Higher Learning, Program of Shanghai Academic/Technology Research Leader (No. 21XD1401100), and Technical Service Platform for Noise and Vibration Evaluation and Control of New Energy Vehicles (No. 18DZ2295900) at Science and Technology Commission of Shanghai Municipality, China.

Special thanks go to Dr. Tao Yuan, Ningning Liu, Xiaolan Wang, Lifeng Ma, and others for their significant contribution to the book's content with their academic work on vehicle interior sound quality. Also, thanks to Xuechun Wu, Shuai Zhang, Dongpo Yang, Tongzhou Wang, Yarong Li, Wenwu Li, Qian Cheng, Fengxiang Guo, Wenchao Jin, Tianpei Feng and other students who gave unstinted effort in this area.

In addition, Springer Nature Press has done much productive work editing the manuscript, and we sincerely thank all the people and units who contributed to the book's publication.

Contents

Chapter 1
Introduction to Vehicle Interior Sound Quality

In a broad sense, a vehicle is a machine that carries and transports people or cargo, including on-road and off-road vehicles. Automobile, the most popular road vehicle, is dedicated in this book.

The drivers or passengers can sense the vehicle interior noise and the vibration radiated by the vehicle body or components of a running vehicle. With the development of damping technology and the wide application of sound-absorbing and insulating materials, the sound pressure level (SPL) of vehicle interior noise continues to decrease, and the vibration is well controlled.

Noise elimination, sound absorption, and vibration isolation (also named "passive noise control (PNC)") are common methods to reduce the vehicle interior vibration and noise that have achieved excellent performance. Recently, the emerging acoustic metamaterials and smart materials offer a novel method to reduce vehicle interior noise, especially the low-frequency noise, and promote the lightweight of the vehicles. With the help of advanced control theory and method, active noise control (ANC) and active vibration control (AVC) technologies have effectively optimized vehicles' vibration and sound environment. However, the laws and regulations on vehicle interior noise control are getting stricter, and people's requirements for excellent interior sound quality are also increasing.

Among the performances of the vehicles, noise, vibration, and harshness (NVH) are increasingly important for the stakeholder, including the vehicle manufacturers, component suppliers and customers. Engineers and researchers have been devoted to improving vehicle interior sound quality through new techniques in the last few decades. However, the occupants still feel uncomfortable or annoyed even if the sound pressure of the vehicle has been reduced greatly. Therefore, many researches are more concerned with sound quality (SQ) in terms of the perceptional characteristics of the hearing sensation of the sound, where the psychological factors, physiological factors, and social practices of a listener are considered. The major sources are equalized actively by active noise equalization (ANE) to realize desirable sound quality. If the active control objectives are sound quality metrics, i.e., active sound

Y. Wang et al., *Vehicle Interior Sound Quality*,
https://doi.org/10.1007/978-981-19-5579-2_1

quality control (ASQC), the results matching desired brand characteristics of vehicles can be obtained.

The SQ refers to sound suitability in a specific technical target or task. The "Sound" no longer refers to the objective physical quantity of sound waves but the auditory perception of the human ear. The "Quality" refers to the auditory perception process of the human ear to sound events and the subsequent subjective judgment. Because of the vehicle interior sound quality, the driver or the passengers receive and process the sound with a comprehensive evaluation from psychological and physiological aspects.

The passenger cars are similar in composition, including several systems such as body, powertrain, chassis, ventilation, and air conditioning. Most individual systems, even the subsystems or components of a vehicle, are noise sources. The powertrain system is the major source of vehicle interior noise. Gasoline or diesel engines of traditional automobiles emit mechanical noise, combustion noise, and aerodynamic noise. Electric motors of emerging electric vehicles generate electromagnetic noise. In addition to the powertrain system, tire/road noise and wind noise are the other two major noise sources, especially when vehicles run at high speeds. Of course, other noises/sounds exist, such as door closing, entertainment, and interactive systems. The noise in different frequency bands is closely related to the sources: the low-frequency noise component is related to the engine and transmission mechanism, the intermediate frequency noise is generated from the tire/road, and the high-frequency noise is related to wind noise and external environmental noise.

There are two sound quality evaluation (SQE) methods: subjective and objective. The former is a process in which people, as the evaluation subject, participate in sound evaluation events and make judgments. It is the basis of sound quality research. The subjective evaluation methods include rank order, rating scale, paired comparison, semantic differential, magnitude estimation methods, etc. However, subjective evaluation features high-level requirements for evaluators, complicated processes, and poor repeatability.

Therefore, the objective parameters are used instead of human auditory perception to evaluate SQ. The earliest and most mature one is A-weighted SPL, which cannot fully reflect the actual sensation of human hearing. For example, some sounds with low SPLs sound very uncomfortable. Several psychoacoustic parameters and models, quantitative and relevant sound stimuli to human auditory perception have been proposed, such as loudness, sharpness, roughness, fluctuation strength, and annoyance. Most prominently, Zwicker's loudness model has been standardized as ISO-532B and adopted by some non-commercial programs. The auditory-sensation-based sharpness model has not yet been internationally standardized but has been included in various national standards, such as the German standard DIN 45,692.

Chapter 2 introduces the vehicle interior noise mechanism, such as air-borne and structure-borne noise. As a complex vibration system with multiple excitation sources, the automobiles deliver energy to multiple target points through different transfer paths. Transfer path analysis (TPA) methods are regularly used to identify the transmission and contribution of each vibration and noise source, which are based on signal prediction methods. The interior noise signals of high-speed vehicles are non-linear and non-stationary. Thus, based on machine learning and compressed sensing, two methods are proposed to process and reconstruct the noise signals, collecting high-precision reference signals for vehicle active noise control (ANC).

Chapter 3 details the subjective evaluation of vehicle's interior sound quality, including the global and instantaneous evaluation. Global evaluation has a wide range of evaluation objects, covering the whole vehicle and components' noise and stationary and nonstationary noise. Instantaneous evaluation mainly aims at the non-stationary sound. The subjective evaluation methods include the rank order, rating scale, paired comparison, and semantic differential method. The establishment of evaluation methods and the processing of evaluation analysis are introduced.

The objective evaluation of sound quality is based on the sound's physical and objective psychological parameters. To establish the relationship between subjective perception attributes of sound quality, physical acoustics, and psychoacoustics, three aspects are analyzed in Chap. 4, including the mapping method, acoustic features extraction of noise samples, and subjective evaluation of results.

As mentioned above, PNC technologies are mainly addressed for vibration damping and noise elimination, which can prevent noise from propagating into vehicles. Chapter 5 then address the specific PNC method: how are the sound insulation and absorption materials developed and designed for the acoustic packages, and how is the multilayer sound-absorbing structure improved to get excellent sound absorption properties? Particularly, acoustic metamaterials (AMs) are a class of artificial materials with distinctive characteristics that are prospective for vehicle interior noise control, especially for low-frequency noise.

Chapter 6 focuses on active noise control. As the vehicle interior noise is mainly concentrated in low- and middle-frequency bands and wavelengths beyond the scope of the PNC method, the ANC method is introduced. How to design and develop the promising ANC algorithm with higher convergence speed, lower steady-state error, and high robustness? And how to improve the algorithm to process the unsteady vehicle interior noise effectively?

Even when the automotive interior acoustic environment is quiet enough, some low-frequency noise disturbances, or maybe slightly softer, could make people feel uncomfortable. The ASQC methods are provided to consider the psychoacoustical parameters as the control objectives. Meanwhile, vibration control is taken into account to reduce the noise level. Several novel hybrid active vibro-acoustic control methods are designed and verified for ASQC in Chap. 7.

Experiments are made to verify the effectiveness of ANC and ASQC methods in improving the vehicle's interior sound quality. Chapter 8 introduces the self-developed hardware and software of the test system, high-speed vehicle experiments, and real-time evaluation results.

Chapter 2
Vehicle Interior Noise Mechanism and Prediction

The studies for vehicle noise, vibration and harshness (NVH) are related to the modification and optimization of noise and vibration characteristics of vehicles, particularly cars or trucks. This chapter introduces vehicle interior noise generation mechanisms, including air-borne and structure-borne noises. An automobile is a complex vibration system with multiple excitation sources (engine, intake/exhaust system, transmission system, tire/road excitation, vehicle body, and wind noise) which deliver vibrational energy to multiple target points through different transfer paths. Several transfer path analysis methods are introduced to identify the transmission and contribution of each vibration/ noise source. Furthermore, the vibration/noise prediction methods based on the model and data are discussed. Under varying high-speed conditions, the vehicle interior noises are nonlinear and nonstationary. Based on machine learning and compressed sensing approaches, the so-called signal decomposition optimization-based back propagation neural network for ear-side noise reconstruction (DBENR) and the multi-variable based time-domain signal reconstruction (MTSR) are introduced for vehicle interior noise. The research results suggest that the two methods can effectively reconstruct the nonlinear and nonstationary vehicle interior noise signals and may provide high-precision reference signals for vehicle active noise control.

2.1 Mechanism and Influence Factors of Vehicle Interior Noise

2.1.1 Mechanism of Vehicle Interior Noise

According to the different generation mechanisms, the vehicle interior noise can be divided into two types: (a) air-borne noise, such as wind noise; (b) structure-borne noise, such as engine noise [1, 2], as is shown in Fig. 2.1. It is necessary to consider the

Y. Wang et al., *Vehicle Interior Sound Quality*,
https://doi.org/10.1007/978-981-19-5579-2_2

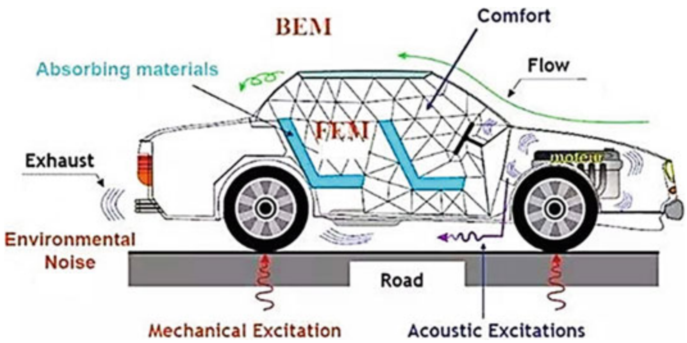


Fig. 2.1 The mechanism of the vehicle interior noise

two types of sources comprehensively, find the sources with the most contributions to the interior noise, and then implement noise reduction purposefully in the vehicle.

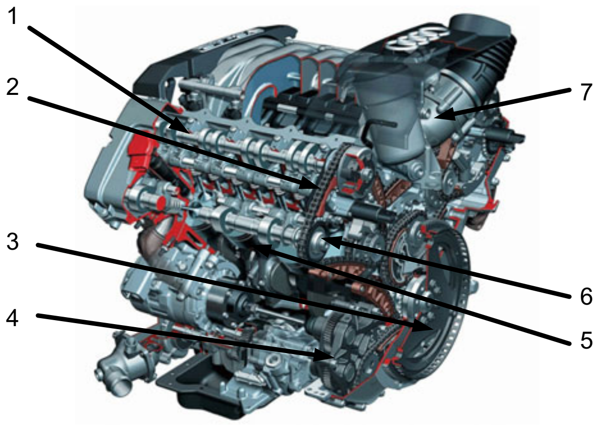
The sound sources in Fig. 2.1 include engine noise, chassis noise, wind noise, etc. The noise radiated from these sources forms an uneven sound field around the vehicle. There are two main paths through which noise propagates into the vehicle:

(a) the noise is directly transmitted into the car through the holes and seams on the vehicle body (joysticks and various instruments in the passenger compartment, the gaps and seams on the vehicle body are almost inevitable);
(b) the sound wave of the external noise sources excites the body panel to vibrate and then radiate the noise into the vehicle.

The vibration sources in Fig. 2.1 mainly include the vibration generated by the powertrain and the ground excitation. These vibrations are transmitted to the passenger compartment through the vehicle chassis, which excites the passenger compartment panel and radiates noise into the vehicle. It should be noted that while the vibration and the sound waves are coupled and are difficult to distinguish, their propagation paths, frequency characteristics, and noise reduction methods are all different. As the walls of the passenger compartment are mainly composed of sheet metal and glass with strong sound reflection properties, the closing of doors and windows of the passenger compartment forces reflections of the air-borne and structure-borne noises in the closed space and superimposed on each other, thus forming a reverberation space.

2.1.2 Main Sources of Vibration and Noise

As is shown in Fig. 2.2, the vehicle vibration and interior noise are mainly excited by the engine, intake and exhaust system, transmission system, tire and road, vehicle body, and wind. Therefore, for modelling, it is necessary to study the primary



1-valvetrain; 2-timing chain noise; 3, 4- noise from accessories such as belt, oil pump, and fan; 5-piston noise; 6-bearing noise; 7-intake and exhaust noise;

Fig. 2.2 Typical noise sources of the engine

sources of vehicle vibration and interior noise, the generation mechanism, and the characteristics in different working conditions [3].

Engine Vibration and Noise

The periodic movement of the crank generates the engine's vibration. The vibration energy is transmitted to other systems through the crank linkage and the body.

The engine noise mainly includes mechanical noise, combustion noise, and aerodynamic noise. Mechanical noise is caused by the mutual collision and vibration between engine parts, which is a rotating mechanical noise affected by the engine speed. The combustion noise is caused by a large amount of heat emitted when the fuel is burned in the cylinder block, forcing the high-pressure gas in the cylinder to impact the engine's combustion chamber. Finally, an engine's aerodynamic noise is mainly caused by the noise radiated directly to the atmosphere in the valve opening and closing process. When the intake and exhaust systems work together with the engine, the aerodynamic noise appears as intake and exhaust noise. The frequency ranges of the engine noise are listed in Table 2.1.

Powertrain Vibration and Noise

Figure 2.3 shows the subsystems of vehicle power, mainly including the vibration isolation system, power transmission system, and intake/exhaust system. The primary purpose of the vibration isolation system is to absorb the energy transmitted to the vehicle body, such as road excitation and engine excitation. The whole powertrain system, including transmission shafts, half shafts, drive axles, etc., generates vibration and noise during driving. During the rotation of each shaft, radial excitation will occur, resulting in vibration and noise. In addition, the meshing errors among the transmission gears generate vibration and noise, which are transmitted to the vehicle

Table 2.1 Frequency ranges of engine noise

Noise sources	Frequency range (Hz)	Remark
Combustion noise	500–7000	Gasoline engine: 500–4000 Hz
Piston noise	2000–8500	Dependent on speed and the cylinders' number
Intake/exhaust noise	50–5400	Intake turbulence < 1 kHz, Exhaust turbulence > 1 kHz
Gear	< 4300	Dependent on speed and number of tooth
Accessory belt	> 3200	Dependent on speed, misalignment, and friction coefficient
Timing belt	> 2500	Dependent on the speed and number of tooth

body through the bearings. The intake noise is mainly caused by the opening and closing of the engine valve. And the exhaust system, connected to the engine on one side and the vehicle body on the other to transmit the engine's vibration to the vehicle body, also generates vibration and noise through the high-temperature airflow in the exhaust pipe.

Vibration and Noise of Body and Chassis Systems

The vehicle body is the main transmission path for vibration and noise. The research on vibration and noise generally involves the modal analysis of the car body (see Fig. 2.4), the transfer characteristics of the structure, etc. The vehicle body is directly connected to most parts, such as the engine, intake, exhaust, transmission, and suspension systems. The vibration and noise of these subsystems are transmitted to the body, and resonances occur when the natural modal frequencies of the body are close to the frequencies of the vibration sources (or sound sources). The body must be sufficiently rigid to avoid low-frequency resonance in the acoustic cavity. The transfer characteristics of the vehicle body change according to the connection modes, so the system connected to the body should be installed where the body's sensitivity is low. Vibration isolation and sound absorption methods are generally used to control the noise transmission of the vehicle body, such as installing damping sheets on the front panel and installing sound-absorb materials on the door panels and ceilings.

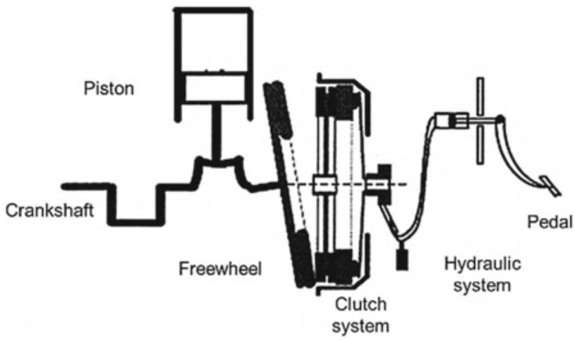


Fig. 2.3 Vibration excitation sources of a clutch pedal [1]

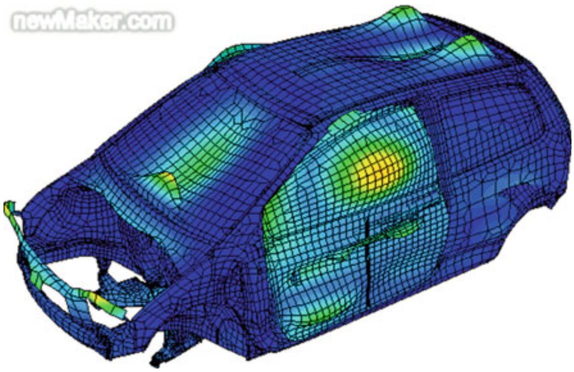


Fig. 2.4 Modal analysis of a car body

Wind Noise

Wind noise, a kind of aerodynamic noise, is the primary noise source of high-speed vehicles. During driving, the car's collision with the airflow cause airflow excitation, turbulence and turbulent noise all over the body. On the other hand, studies have shown that if the car travels at low-speed, the interior noise mainly comes from the engine and tire-road noise. But wind noise gradually surpasses other noises if the car speed exceeds 80 km/h.

According to the aerodynamic acoustics theory, aerodynamic noise can be approximated by the superposition of three linear sound sources:

(a) Monopole sound source. Its sound intensity is greatly affected by airflow velocity, which is proportional to the 4th power of the velocity. There are two prominent cases of using monopole sound sources: one is the noise leaking transmitted through the door and window seals, and the other is called aspiration noise generated by the dynamic air pressure outside the vehicle, forcing a considerable local negative pressure at the window sealing strip and the subsequent deformation of the sealing strip and radiation of noise into the vehicle.

(b) Dipole sound source. Its sound intensity is proportional to the 6th power of the flow velocity. An example of a dipole sound source is the unsteady air pulsation on the car's surface. This dynamic pulsation sound is uncertain and hard to measure in the time and space domain. It can be understood as hundreds of related micro-speaker and exciter arrays acting on the surface of the vehicle structure by a particular statistical law, resulting in the transmission of the interior noise. However, the simulation research of high-speed vehicle wind noise mainly adopts the dipole sound source.

(c) Quadrupole sound source. Its sound intensity is proportional to the 8th power of the flow velocity. Usually, the air velocity outside the vehicle is much less than the speed of sound (Mach number is much less than 1), and the sound generation efficiency of the quadrupole sound source is lower than that of the dipole sound source.

The generation of unsteady aerodynamic pulsations on the car surface is closely related to the shape of the car body surface. Figure 2.5 shows several types of unsteady aerodynamic pulsations caused by gas flowing through different shapes of car bodies. Figure 2.5a shows the flow separation by the frontal right-angle shape of the structure; Fig. 2.5b shows that the flow separation is reduced by the frontal right-angle passivation of the structure, but an adhesion zone is generated; Fig. 2.5c indicates that the flow separation is avoided by the streamlined design of the structure and results in an entire adhesion zone. Aerodynamic pulsations in different regions affect interior noise, and it is generally desirable to reduce aerodynamic separation.

The unsteady pulsating noise on the body surface is divided into two types in the frequency-domain. One is the effect of the window and the cabin cavity, which produces low-frequency "wind vibration" (flutter or buffering); the other is the effect of the roof, which produces a "wind rush" whose frequency is generally above 500 Hz.

Tire/Road Noise

As is shown in Fig. 2.6, engine noise aside, the tire/road noise represents the most significant percentage of noise in accelerating conditions.

The tire/road noise mainly consists of two parts: (a) the noise caused by the vibration of different parts of the tire; (b) the noise caused by the interaction between the tire and the road surface. According to the generation mechanism, it is mainly divided into three categories: tire vibration noise, air pump noise and aerodynamic noise.

(a) Noise caused by tire vibration

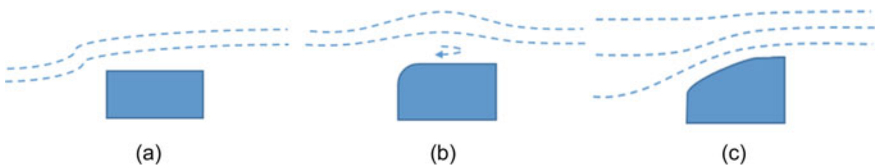
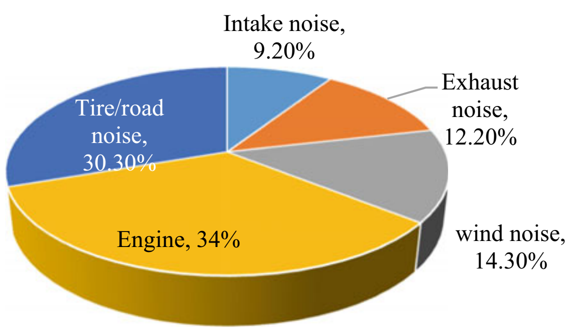


Fig. 2.5 The influence of geometry on airflow field: **a** pneumatic separation; **b** pneumatic separation and reattachment flow; **c** fully attached flow

Fig. 2.6 Noise sources of a vehicle



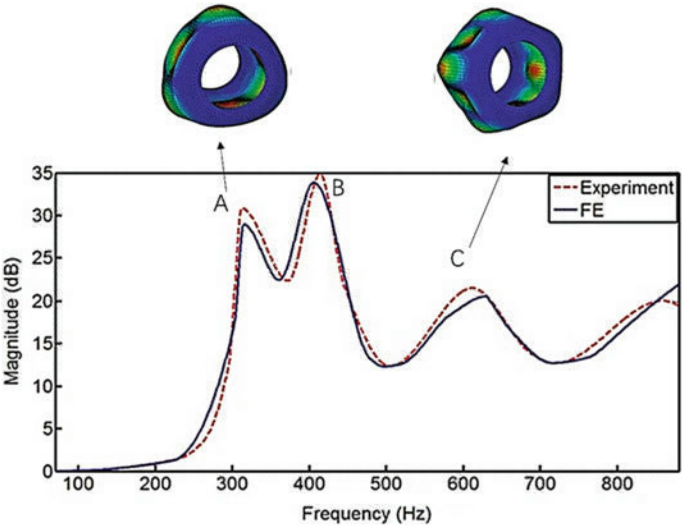


Fig. 2.7 Experiment and FEM analysis of a tire

The noise radiated by tire vibration is the main component of tire noise, caused by the internal excitation of the tire and the road when the wheel is rolling. The noise generated by road excitation mainly includes impact noise, tread radiation noise and friction noise. Due to the tire's unevenness, the tire's internal vibration is produced, which in turn resonates with the tire cavity and radiates noise. Figure 2.7 illustrates the spectrum of a loaded tire, in which Peak A corresponds to the first-order mode of the tire, Peak C corresponds to the higher-order mode, and Peak B is caused by the resonance of the coupling between the tire structure and the acoustic mode of the air chamber.

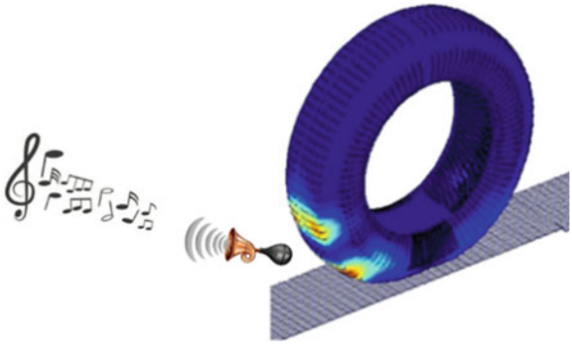
(b) Air pumping noise
The air pumping noise refers to the noise generated by the continuous compression and expansion of the air in the tire. When the tire is rolling, the volume of the tire groove is reduced. Like air jets, the air in the groove is quickly squeezed out and produces noise. When the tire leaves the road, the tire pattern returns to the original, so the grooved pressure is reduced, and the air-intake generates noise. The noise generated by the continuous discharge and intake of air from the grooves is also called "groove noise". In addition, the unevenness of the road surface creates a gap between the tire and the road surface, so the air in the gap is squeezed/released, producing noise similar to the groove noise (Fig. 2.8).

(c) Tire aerodynamic noise

When the tire rotates at high speed, the surrounding air flow is disturbed, so aerodynamic noise is generated with the air pressure change caused by this unstable air flow. This type of tire noise is the most apparent vehicle noise at high-speed. When

Fig. 2.8 Noise measurement of a rolling tire

Fig. 2.9 Horn effect



the airflow near the wheel passes through the tire groove, if the vibration frequency of the cavity in the groove is consistent with the vibration frequency of the airflow, the resonance phenomenon occurs. Figure 2.9 shows that a horn-shaped semi-closed space between the tire and the road amplifies the radiated SPL. This phenomenon is called the "horn effect".

The tire noise generation mechanism is complex, and the abovementioned noises may coexist. The tire noise has a wide frequency band, and its noise energy is mainly concentrated around 1000 Hz. The noise generated by tire vibration and internal excitation is in the low-frequency range below 500 Hz. Such low-frequency noise is easily transmitted to the vehicle's interior through the body structure and dramatically impacts passengers' comfort.

2.2 Transfer Path Analysis of Vibration and Noise

The propagation of vibration and noise in the vehicle also needs to be studied. The transfer path analysis (TPA) is a test-based method known as the vector superposition method. It determines, on the premise of the linear time-invariant characteristic of the automotive, the influence of each excitation source on the response position by analysing the transmission path of vibration and noise energy in the mechanical system. The predicted noise signal on the occupant's ear is the vector superposition of each excitation source via the corresponding path, as is shown in Fig. 2.10.

Therefore, the response on the occupant's ear side can be expressed as:

$$P = P_i + P_k = \sum_i H_i(\omega)F_i(\omega) + \sum_k H_k(\omega)Q_k(\omega) \tag{2.1}$$

where P is the total response of the passenger's ear side, P_i is the structure sound response, P_k is the air sound response, $H_i(\omega)$ and $H_k(\omega)$ are the frequency response functions (FRF) of the structure-borne noise and air-borne noise, and $F_i(\omega)$ and $Q_k(\omega)$ are the structural and acoustic loads. It can be seen from Eq. 2.1 that the key of TPA is the calculation of the FRF and the acquisition of the input load. The FRF between the excitation and output signals can be measured by the standard sound source or force hammer method. The relationship between the three functions is shown in Eq. 2.2:

$$\begin{bmatrix} a_1 \\ \vdots \\ a_i \end{bmatrix} = \begin{bmatrix} H_{11} & \cdots & H_{1j} \\ \vdots & \ddots & \vdots \\ H_{i1} & \cdots & H_{ij} \end{bmatrix} \begin{bmatrix} F_1 \\ \vdots \\ F_i \end{bmatrix} \tag{2.2}$$

where H is the FRF between the excitation signal and the output signal, F is the excitation force, and a is the acceleration output.

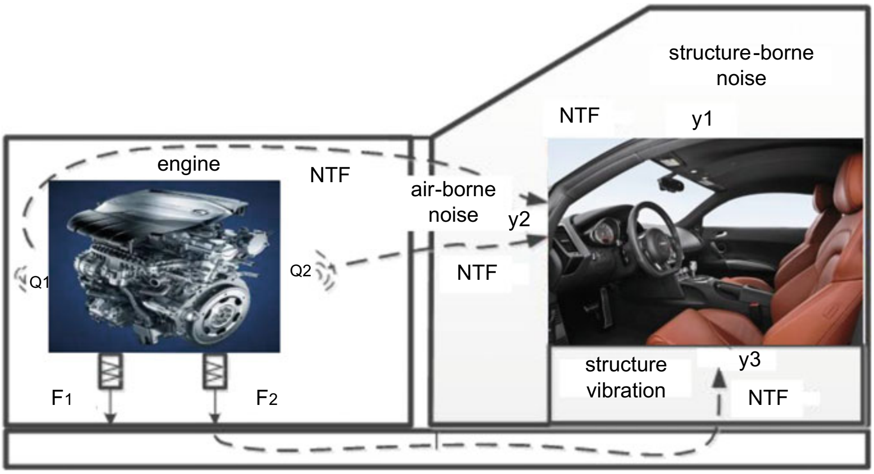


Fig. 2.10 Transfer path model diagram

The excitation load can be obtained by multiplying the inverse matrix of the FRF to the left of Eq. 2.2.

$$\begin{bmatrix} F_1 \\ \vdots \\ F_i \end{bmatrix} = \begin{bmatrix} H_{11} & \cdots & H_{1j} \\ \vdots & \ddots & \vdots \\ H_{i1} & \cdots & H_{ij} \end{bmatrix}^{-1} \begin{bmatrix} a_1 \\ \vdots \\ a_i \end{bmatrix} \tag{2.3}$$

The different TPAs are derived from different calculation methods of the FRF and the load signal. There are three TPAs [4–13]: conventional transfer path analysis (CTPA), operational transfer path analysis (OTPA), and operational-X transfer path analysis (OPAX). The basic principles of these methods are as follows.

2.2.1 Conventional Transfer Path Analysis

Each excitation could be transmitted to multiple response points in the vehicle interior through varied transfer paths with either an amplification or attenuation effect. TPA can quantify various vibration and noise sources and their paths and determine which is dominant. The contribution calculation accuracy of CTPA is high and often used to compare with other modified TPA methods. Next, the fundamentals of TPA are described.

$$P(\omega) = \sum_{i}^{m} H_i(\omega) F_i(\omega) + \sum_{j}^{n} H_j(\omega) Q_j(\omega) \tag{2.4}$$

where P is the total response of the target point, $i = 1, 2, \ldots, m$ and $j = 1, 2, \ldots, n$ are the ith and jth transfer paths, H_i and H_j are the FRF of the ith and jth transfer paths, F_i and Q_j are the structural loads and the acoustic load.

Either calculation or testing can obtain the transfer functions of vibration and noise paths. The experimental transfer function is usually obtained by applying the excitation at the load position and measuring the interior sound pressure. Take the exhaust/intake noise for example, the vibration-noise transfer function from the hanger of the exhaust system to the passenger can be obtained by hitting the hanger of the exhaust system with hammer and measuring the interior sound pressure near the ear, while the noise-noise transfer function from the intake system to the passenger can be obtained by measuring the interior sound pressure near the ear and the sound pressure radiated from a standard loudspeaker at the inlet position.

The core of CTPA is transfer function measurement and load identification. Although CTPA has high calculation accuracy, it needs to disconnect the active and passive ends of the system while measuring its FRF, which is very time-consuming and labor-intensive, therefore, inefficient in modelling practice.

2.2.2 Operational Transfer Path Analysis (OTPA)

To overcome the shortcomings of the CTPA, such as low modelling efficiency and time-consuming, researchers proposed a method to improve the efficiency of traditional TPA, named operational transfer path analysis (OTPA). OTPA is a test analysis method similar to operational modal analysis (OMA). The method calculates the transfer function of each path and the input excitation signal. It uses the operating condition data to calculate the system's transfer function and replaces the traditional FRF without adding additional tests. Based on the ranking of the contribution of each path by OTPA, the main factors affecting the occupant's comfort can be determined. Finally, each subpath's noise/vibration contribution can be synthesized to predict the noise/vibration level at the target position.

The OTPA significantly saves the test time at the cost of reducing the energy transfer path in the modelling. Moreover, this method cannot be used in combination with CAE technology.

$$P(\omega) = \sum_{j=1}^{m} T_j(\omega) P_j(\omega) + \sum_{i=1}^{n} T_i(\omega) \ddot{X}_{pi}(\omega) \tag{2.5}$$

where $T_j, j = 1, 2, \ldots, m$ is the transfer function of the jth air–borne noise transfer path, P_j is the sound pressure signal, T_i, $i = 1, 2, \ldots, n$ is the transfer function of the ith structure-borne noise transfer path, and $\ddot{X}_{pi}$ is the acceleration signal.

It can be seen that the OTPA is similar to the CTPA, but it uses the transfer function calculated from the working condition data instead of the FRF. If the critical energy transfer path is omitted in the modelling process, the accuracy of the calculation results will be affected significantly.

2.2.3 Operational-X Transfer Path Analysis (OPAX)

Although the OTPA does not need a complicated test, the accuracy of the calculated contribution is low. Therefore, researchers proposed the operational-X PA (OPAX) method to compensate for those shortcomings. The basic theory of this method can be denoted by CTPA formulas, except that the OPAX method introduces the function of target point y and reference point u_q .

$$y_k(\omega) = \sum_{i=1}^{n} H_{ki}(\omega) F_i(\omega) + \sum_{j=1}^{p} T_{kj}(\omega) Q_j(\omega) \tag{2.6}$$

$$u_q(\omega) = \sum_{i=1}^{n} H_{qi}(\omega) F_i(\omega) + \sum_{j=1}^{p} T_{qj}(\omega) Q_j(\omega) \tag{2.7}$$

Though a load recognition method based on dynamic stiffness, like CTPA, the OPAX method uses the parameterization model that is constructed while recognizing the load. As the number of reference points affects the accuracy of the dynamic stiffness, this method features adjustability. It also needs to calculate the FRF of the system, but compared to the CTPA method, the number of reference points is significantly reduced. Therefore, the workload of the OPAX method is much less than the CTPA method.

However, in the OPAX model, the stability of the super-ordinary equation composed of the FRF is poor. To solve this problem, one needs to use working data to supplement the parameterization model. Still, the OPAX method makes a good balance in calculating precision and time consumption, although the CTPA method's accuracy is higher, and TPA methods are more time-consuming.

2.2.4 Transfer Function and the Estimation Method

The modal superposition method can be used to calculate the transfer function of the automotive system.

$$\mathbf{M}\{\ddot{x}\} + \mathbf{C}\{\dot{x}\} + \mathbf{K}\{x\} = \{f(t)\} \tag{2.8}$$

$$\mathbf{C} = \alpha_{para}\mathbf{M} + \beta_{para}\mathbf{K} \tag{2.9}$$

where α_{para} and β_{para} are proportional constants, $\mathbf{M}$ is the mass matrix, $\mathbf{C}$ is the proportional damping matrix, K is the system stiffness matrix, and $f(t)$ denotes the force.

By mode transformation,

$$\{x\} = \sum_{r=1}^{N_{mode}} q_r\{\Psi_r\} \tag{2.10}$$

where $\{\Psi_r\}$ is the rth mode of a non-damping system. Substituting Eq. 2.10 into the Eq. 2.8:

$$\mathbf{M}\left(\sum_{r=1}^{N_{mode}} \ddot{q}\{\Psi_r\}\right) + \mathbf{C}\left(\sum_{r=1}^{N_{mode}} \dot{q}\{\Psi_r\}\right) + \mathbf{K}\left(\sum_{r=1}^{N_{mode}} q\{\Psi_r\}\right) = \{f(t)\} \tag{2.11}$$

Left multiply the equation by $\{\Psi_s\}^{\mathrm{T}}$

$$\{\Psi_s\}^{\mathrm{T}}\mathbf{C}\{\Psi_r\} = \begin{cases} 0, & r \neq \mathrm{s} \\ \alpha_{para}m_s + \beta_{para}k_s = c_s, & r = s \end{cases} \tag{2.12}$$

where c_s is damping coefficient, so

$$m_s \ddot{q}_s + c_s \dot{q} + k_s q_s = \{\Psi_s\}^{\mathrm{T}}\{f(t)\} \tag{2.13}$$

Make $\{f(t)\} = [F]e^{j\omega t}$, $q_s = Q_s e^{jwt}$

$$\left(-\omega^2 m_s + j\omega c_s + k_s\right) Q_s e^{j\omega t} = \{\Psi_s\}^{\mathrm{T}}[F]e^{j\omega t}$$

$$Q_s = \frac{\{\Psi_S\}^T[F]}{-\omega^2 m_S + j\omega c_S + k_S} \tag{2.14}$$

By Eq. 2.14, the response of the vehicle system can be calculated. Based on Eq. 2.10, the displacement response can be calculated:

$$\{X_{disp}\} = \begin{bmatrix} X_1 \\ X_2 \\ \vdots \\ X_N \end{bmatrix} e^{j\omega t} = \sum_{r=1}^{N_{mode}} q_r \{\Psi_r\} = \sum_{r=1}^{N_{mode}} Q_r \{\Psi_r\} e^{j\omega t} \tag{2.15}$$

$$\begin{aligned} \{X_{disp}\} = \begin{bmatrix} X_1 \\ X_2 \\ \vdots \\ X_N \end{bmatrix} &= \sum_{r=1}^{N_{mode}} Q_r \{\Psi_r\} = \sum_{r=1}^{N_{mode}} \frac{\{\Psi_r\}^{\mathrm{T}}[F]\{\Psi_r\}}{-\omega^2 m_r + j\omega c_r + k_r} \\ &= \sum_{r=1}^{N_{mode}} \frac{\{\Psi_r\}^{\mathrm{T}}\{\Psi_r\}}{-\omega^2 m_r + j\omega c_r + k_r}[F] \end{aligned} \tag{2.16}$$

Equation 2.15 can be expressed as

$$\{X_{disp}\} = \sum_{r=1}^{N_{mode}} \frac{\{\Psi_r\}^{\mathrm{T}}\{\Psi_r\}[F]}{k_r\left[1 - \left(\frac{m_r\omega^2}{k_r}\right) + 2j\xi_r\left(\frac{m_r\omega\sqrt{k_r/m_r}}{k_r}\right)\right]} \tag{2.17}$$

Two calculation methods, the multi-point and the single-point excitation, are used to determine the system model and the modal response parameters.

Assume that the j point in the structure is excited by F_j, so

$$\mathbf{F} = \begin{bmatrix} 0 \ldots 0 & F_j & 0 \ldots 0 \end{bmatrix}^{\mathrm{T}}$$

$$\{X_{disp}\} = \sum_{r=1}^{N_{mode}} \frac{\{\Psi_r\}\Psi_{jr}F_j}{k_r\left(1 - \left(\frac{m_r\omega^2}{k_r}\right) + 2j\xi_r\left(\frac{m_r\omega\sqrt{k_r/m_r}}{k_r}\right)\right)} \tag{2.18}$$

Then the response X_i at any point is

$$X_i = \sum_{r=1}^{N_{\mathrm{mod}\,e}} \frac{\Psi_{ir}\Psi_{jr}F_j}{k_r\left(1-\left(\frac{m_r\omega^2}{k_r}\right)+2j\xi_r\left(\frac{m_r\omega\sqrt{k_r/m_r}}{k_r}\right)\right)} \tag{2.19}$$

and

$$\frac{X_i}{F_j} = \sum_{r=1}^{N_{mode}} \frac{\Psi_{ir}\Psi_{jr}}{k_r\left(1-\left(\frac{m_r\omega^2}{k_r}\right)+2j\xi_r\left(\frac{m_r\omega\sqrt{k_r/m_r}}{k_r}\right)\right)} \tag{2.20}$$

Equation 2.20 is defined as the transfer function between the system excitation point *j* and the response point *i*. Therefore, the system response can be calculated at the frequency domain by this formula.

The transfer function, force, and response are denoted by H_{ij}, F_j, and X_i, so

$$H_{ij}(\omega) = \frac{X_i}{F_j} \tag{2.21}$$

According to the mutuality of linear systems, $H_{ij} = H_{ji}$. Based on the definition of the transfer function, $X_i = H_{ij}F_j$.

If $\mathbf{F} = [F_1, F_2, \ldots, F_N]^{\mathrm{T}}$, according to the principle of linear superposition

$$X_i = H_{i1}F_1 + H_{i2}F_2 + \cdots + H_{iN}F_N = \left[\, H_{i1}\; H_{i2}\; \ldots\; H_{iN} \,\right] \begin{bmatrix} F_1 \\ F_2 \\ \vdots \\ F_N \end{bmatrix} \tag{2.22}$$

$$\{X_{disp}\} = \begin{bmatrix} X_1 \\ X_2 \\ \vdots \\ X_N \end{bmatrix} = \begin{bmatrix} H_{11} & H_{12} & \cdots & H_{1N} \\ H_{21} & H_{22} & \cdots & H_{2N} \\ \vdots & \vdots & \ddots & \vdots \\ H_{N1} & H_{N2} & \ldots & H_{NN} \end{bmatrix} \begin{bmatrix} F_1 \\ F_2 \\ \vdots \\ F_N \end{bmatrix} = \mathbf{HF} \tag{2.23}$$

The above transfer function is based on the ideal hypothesis: the output is entirely caused by input, without any noise. However, this hypothesis is difficult to satisfy because the input and output will inevitably contain noise, as is shown in Fig. 2.11. Therefore, various estimation methods are proposed to reduce the noise impact by estimating the transfer function with the actual measurement input and output data. The transfer function estimation method of the actual system mainly includes three estimation methods named H_1, H_2 and H_v.

H_1 estimation: As the most commonly used estimation method, it assumes that the input is noise-free and the output is noisy, so:

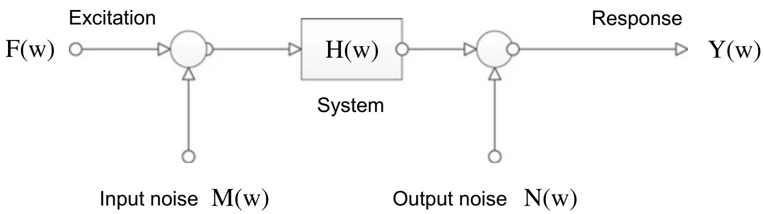


Fig. 2.11 Actual state system flow chart

$$Y(\omega) = H(\omega) \cdot F(\omega) + NS(\omega) \tag{2.24}$$

where $H(\omega)$ is the transfer function of the system. To suppress the influence of the output noise on the system response, the least-squares method is used to calculate the n test results. It is shown in Eq. 2.25.

$$J = \sum_{i=1}^{n} |NS_i(\omega)|^2 = \sum_{i=1}^{n} [Y_i(\omega) - H_i(\omega)F_i(\omega)][Y_i(\omega) - H_i(\omega)F_i(\omega)]^* \tag{2.25}$$

where "*" is complex conjugation. Make

$$\frac{\partial J}{\partial H^*} = 0 \tag{2.26}$$

The estimated equation of the transfer function H_1 can be obtained as:

$$H_1 = \frac{\left[\sum_{i=1}^{n} Y(\omega)F^*(\omega)\right]}{\left[\sum_{i=1}^{n} F(\omega)F^*(\omega)\right]} = \frac{G_{fy}}{G_{ff}} \tag{2.27}$$

where G_{fy} denotes the cross power spectrum of the input signal and the output signal, and G_{ff} denotes the auto power spectrum of the input signal.

H_2 estimation: As the opposite of the H_1 estimation, it assumes that the output response is noise-free and the input excitation is noisy.

Therefore, all response measurements are accurate. But one must perform a least-squares estimate of n input measurements to get the least noisy input. Since the output is noise-free, the transfer function is calculated by dividing the self-spectrum of the output by the cross-spectrum of the input and output signal.

$$Y(\omega) = H(\omega)[F(\omega) - MS(\omega)] \tag{2.28}$$

Similar to the H_1 estimation method, the H_2 estimation formula is derived as

$$H_2 = \frac{G_{yy}}{G_{yf}} \tag{2.29}$$