Chapter 1 Introduction

1.1 Motivation

Due to the complexity and the number of sources, control of noise inside car cabins presents a challenging problem. The complexity continues to increase with the development of lighter more fuel-efficient automobiles, which are more susceptible to low frequency vibration and noise. The improvement of noise reduction technologies in the automobile industry will likely remain an important area of practical research, since passenger comfort is key to competition between manufacturers.

Low frequency noise in automobiles is primarily caused by a combination of the disturbances due to the firing of the engine and the interaction between the tires and the road surface. Conventional passive noise absorption techniques work well at higher frequencies but are not effective at the low frequency range because of the long wavelengths associated with these frequencies. Fortunately, this is exactly the range in which active control technology demonstrates its best results [1-4]. Also, due to variations that may occur in the manufacturing process from vehicle to vehicle, there is no guarantee that a conventional passive solution designed for one vehicle will be as effective for another sample of the same vehicle type. This problem of variation is overcome with active techniques, since an appealing characteristic of the technology is that it is adaptive. Therefore, a good approach to controlling low frequency interior automotive noise would be to add the benefits of an active noise control (ANC) system to a preexisting, more classical, passive noise reduction technology in order to lower sound levels with little penalty in terms of space requirements.

The implementation of such an active system is far from straightforward, since the structural-acoustic system is very complex. No clear description of the acoustic and vibration disturbance path in automobiles can be found in the literature today. Nevertheless, as it will be described in section 1.3, practical application of active control to power train noise has shown very good results, and has been applied in the industry. Control of noise induced by the interaction of the tires and the road surface also shows promise for practical research in active techniques in the future.

The present research investigates the potential of a broad band multi-reference control system, which implements state-of-the-art lightweight, compact piezo-electric acoustic sources for the reduction of interior noise in a sport utility vehicle. Details concerning the mechanics of the source will be given in chapter 5. As it will be explained in section 1.3, most of the previous research in this area has been performed on light Japanese or European cars. The research discussed in this thesis was applied to a Ford Explorer, which already utilizes extensive passive noise control techniques unlike light Japanese and European cars. Thus, the initial acoustic field inside the cabin is heavily damped, even at low frequencies.

1.2 Active Noise Control

Before starting any discussion on active noise control strategy, it is important to define the application of the technology in terms of sound frequency. In general, noise problems exist in the entire human audible range of frequencies (20 Hz to 20 kHz). For disturbances above 1000 Hz, the noise control strategy is typically to use passive techniques. Passive control techniques are based on the relationship between the wavelength of the noise and the thickness of the passive material. This means that control of low frequency noise, having longer wavelengths, would require very thick absorption materials. This is not feasible in an automobile due to the dimensions of the cabin. Active noise control, on the other hand, uses acoustic sources to generate a secondary sound field that destructively interferes with the original undesired sound field. The performance of this technique is also wavelength dependant. The lower the frequency, the larger the area throughout which noise cancellation occurs. It can be concluded that Active Noise Control is complementary to passive control.

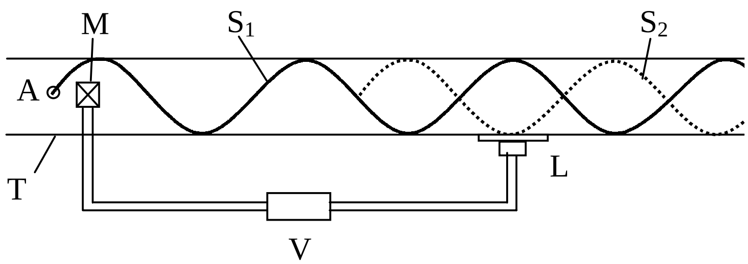

Figure 1.1 Diagram 1 from the illustration's page of Lueg's 1936 patent

The concept of active noise control has been understood since the early part of this century, but it is only with the recent development of fast digital signal processing that the development of practical systems has been feasible. Overviews of the history of active control have been presented by Warnaka [1] and Nelson and Elliot [2]. In addition, recent textbooks by Nelson and Elliot [3] and Fuller, et al. [4] present the actual state-of-the-art active control techniques. The German physicist, Lueg, was the first to provide a description of active noise control in a 1936 U.S. patent [5], which gives the example of a one-dimensional duct problem (Figure 1.1) with a wave propagating only in one direction. The technique, as described in the patent, relies on the principle of superposition. In Figure 1.1 the duct, T, contains a disturbance (S_1) which propagates along the tube through point A. The microphone, M, detects the noise and transforms it into an electrical signal. The signal is fed through amplifier V to the loudspeaker L. The electronic system is seen as a delay line, whose length has to be adjusted such that the loudspeaker can produce the 180° phase reversal of the disturbance wave (S_2). If the loudspeaker is perfectly synchronized to the passage of the noise, the acoustic pressure due to the noise and the loudspeaker add to zero (i.e. static pressure) and the noise downstream of the speaker is cancelled.

In 1953 Olson and May [6] developed a different system, in which the disturbance noise is detected by a microphone and fed back through a controller to a loudspeaker located close to the microphone. This approach is different because it does not require prior knowledge of the disturbance. The technique demonstrated good local reduction in the 20-300Hz band. The classic work of Lueg and Olson demonstrated two different control approaches. Lueg's approach can be seen as a feedforward strategy since prior knowledge of the disturbance is obtained with an upstream microphone. Olson and May's approach is a feedback arrangement where the error sensor is close to the loudspeaker.

In the mid-1950s, Conover [7] investigated the problem of active control of electrical transformer noise, which has since become a classic problem in the field of active noise control. His scheme was to place a loudspeaker near the surface of the transformer and to cancel the pressure in the near field. His arrangement showed reduction in the direction perpendicular to the loudspeaker but showed an increase in the sound pressure level in other directions. In fact 'spillover' or localized increases in sound pressure level due to constructive interference between the disturbance and control fields, is a very common problem in active noise control [3]. Conover's feedforward controller required manual adjustment of the phase and amplitude of the sinusoidal reference signal. Therefore, he abandoned the technology for a more classical passive technique since automatic systems were not feasible.

Renewed interest in active noise control has been sparked with the recent advances in digital electronic technology research, which has allowed ANC application to more complicated systems. Initially, the effectiveness of the technology was limited to spectrally simple noise or noise that contains only a few frequencies that are closely spaced such as transformer noise or the spatially simple noise we hear in ducts. Nowadays, researchers have been exploring a wider range of problems with more complicated systems such as passenger vehicles. As described by von Flotow [8], three main parameters can be seen to define the application of active noise control:

- The Spatial Extent: The spatial distribution of the disturbance field is complex, and its application is limited by the accuracy with which the control actuators can reproduce this complex field.
- The Spectral Extent: if the disturbance spectrum is broadband, the control design must exploit much more detailed knowledge of the problem's dynamics in order to achieve its inversion over a large band.

- Passive Damping: The difficulty increases as the level of resonance of the system increases. In [9], a relationship for the minimum damping required to obtain some control is derived.

Recent applications of active noise control technology have been successful and commercially available systems have been developed. Saab [10] has implemented a filtered-X feedforward controller for interior cabin noise (discussion of the algorithm is presented in chapter 2.3). This system, using 24 control sources and 48 error microphones, resulted in the Sound Pressure Level (SPL) decreasing by 10 dB at the propeller tones. Digisonix Inc. has developed a system for HVAC (Heating Ventilating and air Conditioning) duct noise, noise that is caused by upstream fans. This system has been installed in a number of buildings and has achieved a broadband sound cancellation of 10 to 13 dB. The technology used by Digisonix is slightly different since the algorithm used to update the controller is the filtered-U feedforward algorithm, which is a modification of the filtered X approach [11]. Feedback-silenced headsets demonstrate application of a third technique [11]. This technique can be seen as the development of the classic work of Olson and May, where a control source and an error sensor are co-located within the earcup. Maximum reduction (20 to 30 dB) is typically obtained around 200 Hz [11].

1.3 Automobile Interior Active Noise Control

The problem of automobile interior cabin noise has been addressed since the 1950s [12]. Most of the principles on how the noise was generated were not understood. The only experimental knowledge available was obtained with microphones measuring the cabin noise. In the early 1960s, one-dimensional tube analysis was used to explain early experimental work [13]. It was in the early 1970s that both finite difference and finite element analyses were applied. In 1975 structural-acoustic analysis using the NASTRAN computer program was implemented [14-16]. Then analyses focused on very low frequencies, typically between 20 and 80 Hz, and gave details of modal data and panel-excited interior noise. The model used was a two-dimensional one, where the pressure field is assumed to be uniform in the cross-body direction. These works have led to passive modifications and redesign of the automobile structure, which have resulted in significant noise reduction. At this time, problems regarding the use of passive materials to control low frequency cavity resonance were pointed out, in particular those concerning the prohibitively large amount of passive materials required.

Application of active noise control technology to interior cabin noise first appears in the 1980s. At this time, research was focused on the control of engine noise. The first system was presented by Oswald [17], who was able to obtain significant reduction below 200 Hz. The system was a SISO (Single Input, Single Output) arrangement of one speaker, one microphone, and one reference signal. A waveform generator was used to convert the pulse train from the engine tachometer into a set of sine waves having the frequency of the engine rotation rate. The amplitude and the phase of these sine waves were adjusted and the waves were summed such that the control speaker would broadcast a mirror image of the engine noise at the control location. Attenuation of up to 30 dB was achieved at a single frequency, depending on the harmonic of the engine and on the engine speed. In fact, Oswald found that the ANC system would reduce the engine noise to about 5 to 7 dB above the 'background noise' at all engine orders and speeds.

In 1986, Nelson, et al. [18-29] provided a detailed analysis of active suppression of enclosed sound fields. This analysis was based on the assumption that the sound field in an enclosure can be expressed as a sum of modal contributions. It was shown that global reduction (i.e. reduction in the acoustic potential energy) was possible at resonant frequencies and also at frequencies of high modal density, providing that the secondary source is located within half the wavelength of a compact primary source. This work was later applied to the control of engine noise [21,22]. In [21] the system used is somewhat more complicated than the one used by Oswald, since four error microphones (located at passenger and driver ear position) are used in conjunction with two control sources (loudspeakers below the front seats). The aim of the controller proposed by Nelson [21] is global attenuation for a single frequency (engine firing frequency), which is accomplished by minimizing the sum of the pressure squared measured at the four error sensors. Attenuation of 10 to 15 dB was obtained for a range of engine speeds (3500 to 6000 rpm) at the driver's head position. It is interesting to see that the system used is very compact (27 cm by 26 cm by 12 cm) and can be operated from a 12 volt, 1 amp power supply. Similar results have been obtained on a wide range of vehicles from small sport cars to large vans using four control sources and eight error microphones [23]. These tests also exhibited significant reduction in the rear of the vehicle for engine speeds varying from 2000 rpm to 3000 rpm.

In 1992, the first commercial power train noise control system was introduced into production by a car manufacturer [24]. The next step has been the development of systems to control interior road noise. This problem is more challenging because the road-tire interaction is random in nature and because the disturbances are distributed between the four wheels. A comprehensive review of tire noise generation has been given by Heckl [25], while measurements of wavenumber spectra for typical roads have been reported by Dodds and Robson [26]. In 1994, an experimentally based model of tire noise was proposed by Roggenkamp, et al [27]. This model takes into account both structure-borne and airborne paths. It uses measured transfer functions between force on the body of the car and pressure inside the cabin (for structure-borne path) as well as frequency response functions (FRF) between pressure around the tire and inside the cabin (for air borne paths). The authors use NIS analysis to determine the optimum exterior microphone number and location required to obtain an accurate model of the air borne paths. Once these two factors have been determined, the noise source can be characterized. The model has been verified with a car, whose front right wheel was mounted on a roller to simulate a speed of 50 mph. It was concluded that the structure-borne model was efficient in the prediction of tire noise only at low frequencies, typically below 150 Hz. It is an important limitation, since road noise is characterized by frequencies from 50 to 300 Hz.

Since most of the noise is structure-borne at low frequency, two different approaches have been considered: Active Structural Acoustic Control (ASAC) and Active Noise Control (ANC). In ASAC, the control is achieved by modifying the vibrating behavior of the disturbance structure. In ANC the control is achieved by direct modification of the pressure field via loudspeakers. In reference [28], both approaches are applied to a scaled model of an automobile in order to control harmonic disturbances. The model is a PVC cavity (one-fifth scale) which has roughly the same shape of the test structure presented later in chapter 3. The bottom plate of the structure is a 2-mm steel plate, which causes the disturbance when excited by a shaker. ANC was performed by using one accelerometer that was mounted on the plate as a reference signal. In addition, one speaker served as the control source and one microphone as the error signal. In the case of ASAC, the same set up was used except that the control source was replaced by a force actuator acting directly on the plate. Results showed that higher reductions were obtained using ASAC, due to the complexity of reproducing the sound field inside the cavity with only one speaker.

Numerous studies have been performed with ANC systems [23, 29-31], both in simulations and on real roads. In [23], computer simulations are used to predict reduction at the error sensors, for different sets of reference signals. As a first step, the maximum possible attenuation is estimated using the coherence function. This method does not include causality constraint on the filters and therefore only computes the maximum reduction achievable. Attenuation of 7 dB is predicted over the 40-150 Hz band at one error microphone, using a combination of six reference signals. As a second step, optimum digital filters are computed. These filters are causal and of finite length (128 coefficients). The attenuation predicted at 150 Hz and below varies between 5 and 20 dB. Later in [29], the same system was applied using a multiple-channel filtered-X LMS algorithm designed for six reference signals, two loudspeakers as secondary sources, and two error microphones. The authors used multiple coherence between different possible accelerometer positions and the signal from an interior cabin microphone to determine the best combination of the six reference sensors. SPL reductions of 7dB at the peaks were measured when the car was driven at 60 km/h on a rough road. More simulations have been performed [30, 31], where the disturbance is caused by the excitation of a single wheel of the car. The goal was to minimize the sound pressure level at one error microphone using one secondary source for the control. In [30], only one reference signal was used and poor reduction is obtained, averaging 1 dB between 10 and 100 Hz. Reference [31] uses a MISO system, including 3 reference signals, to obtain attenuation of 9 dB between 50 and 170 Hz.

The same type of procedures have been used for ASAC (force actuators acting directly on the body of the car that are used as secondary sources instead of loudspeakers) [32-36]. Up to 10 dB reduction in a laboratory environment can be achieved with the procedure (i.e. one wheel is excited). Also, attenuation on the road has been achieved in a very narrow band, typically 6 to 7 dB at the error sensor, between 75 and 105 Hz [36]. These results were obtained using 6 reference signals, 150 control filter coefficients, and 128 secondary path filter coefficients.

Key to controlling road noise is the proper selection of a location for the reference signals, as well as the number of reference signals. Six reference signals were used in [29, 36], which requires very high computational power. The impact of the number of references used in the controller has been studied in [37]. Results showed that in the case where one wheel is excited, four reference signals are required. A method to reduce the number of signals has been introduced by Akiho [38]. In his experimental verification it is shown that two virtual reference signals can be computed from three accelerometers. The multiple coherence between the three accelerometers and a microphone located inside the car was compared with the multiple coherence between the virtual references and the same microphone. The two coherence curves showed little difference, demonstrating that the virtual reference technique can be applied to the active control of road noise inside automobiles to reduce the number of reference signals.

1.4 Thesis Objectives and Organization

Due to the complexity of the numerous acoustic transmission paths in automobiles, the implementation of active noise control technology is not straightforward. In the research presented here, active noise control was applied to the control of power train noise as well as to the noise caused by the interaction of the road and the tires. First, a test structure was built to understand the mechanism of the control and to develop a strategy for the more complex case of the actual car. This strategy involved the optimization of the actuators and sensors locations for global reduction of sound pressure level in the cavity. In order to achieve this goal, a finite element model of the test structure was also built, and is the basis for optimization of the locations of the error sensors and actuators using a genetic algorithm. Second, the technology was applied to a Ford Explorer with standard passive treatment for power train noise control, between 40 and 500 Hz. Third, tests were conducted to investigate the potential of the active noise control technology to control real life noise generated by the interaction of road and tires.

The primary goals of the project can be summarized as follows:

- To investigate the possibility and performance of active control of cabin noise in a Sport Utility Vehicle, in which typical passive noise and vibration treatments are present.
- To further develop and improve the active control of power train and road noise over a wider frequency range than in previous studies.

- To investigate the potential of piezoelectric based advanced acoustic sources for the control of power train and road noise.

The secondary goals of the project can be summarized as follows:

- To develop, validate and "fine tune" a finite element model of the test structure.
- To develop an optimization approach for the locations of the error sensors and control sources.
- To experimentally implement active control in the test cavity. These experiments will be used to investigate the potential of harmonic and broadband control in a simulated vehicle under laboratory conditions.
- To develop a multiple coherence technique for the optimization of the configurations of the reference sensors for the control of power train and road noise in the Ford Explorer.

A Ford Explorer donated by Ford Motor Co. and resident at Virginia Tech was used to perform active noise control in a realistic environment. As it will be outlined in chapter 3, experiments were conducted both in a laboratory and on the road with all the equipment on board. Chapter 2 addresses some theoretical background needed for the work. A numerical method for simulation of active control is presented; also provided is a discussion of techniques for optimization of the locations of the system components. To conclude chapter 2, the algorithm used to experimentally perform the control is presented. Chapter 3 details the development of the finite element model created to perform the optimization of the locations of the actuators and sensors. The optimization by use of genetic algorithms is presented in chapter 4, together with experimental data obtained on the test cavity. Tests in the cavity were performed for various harmonic disturbances and also in the case of broad band disturbances (i.e. random noise between 40 and 500 Hz). Chapter 5 deals with the application of the control developed in the test cavity to the Ford Explorer. It includes reference signal selection, active noise control of power train noise and road noise for various configurations of the actuators and the error sensors. The control was achieved using commercially available speakers, but also with advanced piezoelectric sources. Finally, chapter 6 presents the overall conclusions drawn from this project and suggestions for future directions of this research.