AERODYNAMIC AND JET NOISE
This contribution reviews progress in the technology for assessing the silencing performance of piston engine intake and exhaust systems on a running engine, with particular reference to flow noise. The new developments include the application of selective averaging, order tracking and optimised sampling rate methods to clearly identify and quantify the small fraction of the total fluctuating wave energy that is being propagated across each discontinuity along the flow path through a highly reactive flow duct system. Such measurements then quantify the interaction between the aeroacoustic sources of excitation and the associated local acoustic characteristics that govern the generation, transfer, propagation and emission of wave energy to the environment.

**INTRODUCTION**

Intake and exhaust systems are designed to reduce engine breathing noise emissions to levels that comply with legislation, provide a sound quality and internal vehicle climate that meets customer expectations, while maintaining optimum fuel efficiency and vehicle performance. This diversity of function, coupled with operational, system layout and space allocation constraints, together with the benefits of rapid prototyping, underline the practical advantages of a design methodology that is firmly based on realistic predictive numerical modelling [1] of acoustic and operational performance. Appropriate existing software [2] can adequately describe the passive acoustic and resonant behaviour of geometrically complex flow duct systems, but does not yet include any influence of flow noise sources on predicted acoustic performance. This deficiency is of practical importance, since flow generated noise is commonly the major contributor [1-3] at higher engine speeds to IC engine breathing noise spectra above 200 Hz.

**FLOW NOISE AND ITS MEASUREMENT**

The sustained excitation of a tuned resonator by shed vorticity in a separating shear layer [4] has been exploited empirically for making musical and other sounds from time immemorial. In the normally highly acoustically reactive intake and exhaust systems such mechanisms [1-4] are seen to generate both coherent and broadband sound, or noise, by a flow driven generator that is highly nonlinear, exciting a resonant system with effectively linear acoustic behaviour. Similar mechanisms [1, 2] can also cause selective amplification of incident sound. The dominant aeroacoustic mechanism of concern here [4] is the action of a fluctuating Coriolis force working on the fluctuating flow. However, with some specific exceptions [1, 4], the relevant controlling features of the flow cannot be specified significantly explicitly, due to current ignorance of many essential details of the associated vortical or turbulent fluid motion. Thus one must normally rely on measurement to establish quantitative descriptions of aeroacoustic sources in relation to the associated boundary geometry, acoustic climate and fluid motion.

Intake and exhaust system geometry consists essentially of silencing and other component elements connected in sequence by lengths of uniform pipe. Wave reflections at the junctions, combined with any local sound generation, produces an acoustic climate comprised of both standing and progressive waves. Superimposed on the time averaged flow, the unsteady fluid motion includes both the acoustically related potential fluctuations combined with vortical disturbances cast off with the separating shear layers [1, 4] or generated at the boundaries, that travel with the mean flow. New robust procedures [2-4] using cross-power spectral analysis of the signals from pairs of flush wall mounted pressure transducers were used to quantify the incident and reflected wave amplitude spectra at appropriate positions along the flow path. The application of selective averaging, order tracking
and optimised sampling rate methods [2-4] provided sufficient precision to identify clearly the small fraction of the total fluctuating wave energy that was being propagated through the system. The location, strength and spectral structure of any flow noise sources/sinks was then derived from the small gains/losses in power flux at the relevant sites. Pilot experiments [2, 4] revealed that the flow generated sound spectra were always closely related to the system acoustic resonances.

Earlier experimental studies of flow generated sound [1, 5] have largely been concerned with measurements of the sound power radiated from the open termination into anechoic, semianechoic or reverberant enclosures, rather than at the sites where it is generated. However, the observed harmonic structure of the emitted sound was also closely associated with the system resonances, while the overall intensity was proportional to the mean flow Mach number $M$ raised to a power lying between four and six for $M < 0.4$. Experiments with flow excited expansion chambers [1, 4, 5] showed that the sound power was also a rather complex nonlinear function of $x/d$, $2 < x/d < 15$, where $x$ is the chamber length or that of the free shear layer between inlet and exit pipes with diameter $d$. Similarly complex nonlinear behaviour was observed with other typical exhaust system component geometries.

Measurements were also made with strongly excited expansion chambers [2, 3], either on the test bench, or in the exhaust line of a running engine accelerating to full speed on a test bed with wide open throttle. The observations provided clear evidence of reverberant amplification or attenuation of the incident sound at specific frequencies, although the relationship with the associated boundary geometry, acoustic climate and fluid motion has not been established yet. Otherwise, the relative observed spectral behaviour corresponded to that predicted by one dimensional linear acoustic models. This was in spite of the fact that the observed amplitude of some spectral components of the cyclic excitation exceeded 3.5 kPa, corresponding to 165 dB. With purely flow excited systems on the test bench [2, 4] the maximum sound pressure levels recorded were some 20 dB lower than this, although the corresponding overall fluctuating pressure levels were close to 160 dB. Otherwise, acoustic characteristics calculated with linear acoustic models were found to be in good agreement with measurements, indicating that wave steepening was not yet a significant factor at the conditions of these experiments.

**DISCUSSION**

Pilot studies [2-4] have established and validated robust and sufficiently precise experimental technology for establishing the position strength and spectral characteristics of flow noise sources in relation to the boundary geometry, the mass flow and the associated acoustic climate. This includes the influence of regenerative amplification by acoustic feedback [2, 3] on source characteristics, with some basic examples [4] of source modelling. The record shows that problems arising from poor signal to noise ratio associated with the presence of standing waves and signal contamination by turbulent pressure fluctuations can be overcome [2, 3] by adopting swept sine or swept periodic excitation combined with appropriate selective averaging. With uncontrolled excitation by flow, coherent power flux measurements [2, 4] were closely identified with the acoustic power flux.

Further progress in predictive modelling of aeroacoustic sources depends on establishing appropriate details [4] of the associated fluid motion. This includes the mapping of the measured or calculated distribution of the fluctuating potential and vortical velocity components with the corresponding Coriolis accelerations. To accomplish such measurements without introducing further sources represents a challenge that is a subject of current studies at the ISVR.

**REFERENCES**

Flow Induced Noise from a Simple Expansion Cavity Type Muffler with Flow Pulsation

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The high flow rate in a muffler induces flow noise. This noise sacrifices the noise reduction effect of the muffler. The flow induced noise from a simple cavity type muffler with pulsating flow and steady flow was comparatively studied. The predominant components of flow induced noise were related to the resonance of the cavity and the tail pipe of the muffler. The flow induced noise was extracted from the total noise radiating from the open end of the muffler by introducing a noise generation model. The sound power level of the flow induced noise in the case with pulsating flow was different from the case with steady flow, especially for the cavity resonance components. Since the pressure fluctuation on the solid boundary is the main source of flow induced noise for a low Mach number flow, the pressure fluctuation spectrum was compared between the cases with pulsating flow and steady flow.

INTRODUCTION

According to the improvement of automobile engine to provide high speed and high power, the amount of working gas increases. When a high flow rate of gas exists, a large flow induced noise radiates from the open end of the muffler. This flow induced noise sometimes sacrifices the noise reduction effect of muffler predicted through the acoustic theory.

In the previous papers\cite{1,2}, the flow induced noise generated from the simple cavity type and the inserted type mufflers with steady flow was experimentally studied. In this paper, the characteristics of flow noise generated from simple expansion cavity type muffler with pulsating flow were studied. The noise sources inside the muffler were also discussed in comparison of the case with steady flow.

EXPERIMENT AND DISCUSSION

The experimental set up is illustrated in Fig.1. The air flow from a blower was supplied to the pre-muffler which had the sound absorbing structure inside its walls. In the experiments for pulsating flow, the pulsation generator was inserted at the position of 3070mm up-stream side of the entrance of the muffler. The probe microphone was employed to detect the pressure fluctuation on the inner wall of the muffler cavity. The mean velocity of the air flow supplied to the muffler was 33m/s both for steady and pulsating flow experiments.

Extraction of Flow Induced Noise from Total Radiated Noise of Muffler

In the experiments for the pulsating flow induced noise, the radiated noise contains not only the flow noise but also the mechanical noise from pulsation generator. It is necessary to extract the noise generated by pulsating flow from the total noise of muffler. Then, the noise generation model of muffler was introduced.

The noise power radiated from the open end of the muffler $W_{pm}$ is recognized as a sum of two kinds of noise power. The other one is the noise power induced by the air flow $W_f$ inside the muffler, then

$$ W_f = W_{pm} - W_{tg}F $$

Figure 2 shows the flow induced noise for pulsating flow calculated by this method in comparison of the case with steady flow. In the case with pulsating flow, the noise power of tail pipe resonance is larger 10-12dB than in the case with steady flow. For the components of cavity resonance in axial direction, this difference becomes larger as 15-25dB. These results strongly suggest that the generation mechanism of flow induced noise in the pulsating flow is not the same as that in the steady flow.

Effects of Cavity Walls on Flow Induced Noise

Two types of noise source can be considered for the flow noise generated inside the muffler. One is the turbulence of the flow and the other is the pressure fluctuation at the solid boundaries. Curle\cite{3} proposed the relationship between these two types of noise source and generated sound pressure. Introducing the adiabatic process, the sound pressure formed at a point
apart from these noise sources is derived as;

\[ p = \frac{1}{4\pi^{2}} \frac{D_{2}^{2}}{r^{2}} \left( \frac{\partial}{\partial t} \right) \int_{V} \rho \, J_{1} \, dV + \frac{1}{4\pi^{2}} \frac{D_{4}^{2}}{r^{2}} \left( \frac{\partial}{\partial t} \right) \int_{S} P \, dA \, (2) \]

The first term of this formula expresses the sound pressure component generated by the disturbance of the flow, and the second term expresses the component generated by the fluctuation of the pressure at the solid boundaries such as inner walls of muffler. From the results of order estimation of these two terms, the first term was two orders smaller than the second term in as low Mach number flow as employed in this experiment. Then, the flow induced noise generated inside the muffler has to be considered in the relationship with the second term of Eq(2).

Figure 3 shows the typical example of pressure fluctuation on the inner walls of the muffler cavity in comparison between the case with steady flow and pulsating flow. As shown in Figs.3(a) and (b), the pressure fluctuations on the inlet wall and the side wall of the cavity are larger in the case with pulsating flow than in the case with steady flow as much as 2.0-3.0 in log(\(\frac{\partial P}{\partial t}\)). This difference will correspond to 20-30dB in the sound power level of noise sources. One of the reasons for this increase in pressure fluctuation is possible that the disturbance of flow in radial direction becomes strong by introducing the pulsating jet flow into the cavity.

When we introduce the steady flow to the muffler, the pressure fluctuation on the side wall becomes small in high frequency range as shown in Fig.3(b). While on the exit wall of the cavity, the pressure fluctuation is larger than on the side wall as much as 1.5-2.0 in log(\(\frac{\partial p}{\partial t}\)) as shown in Fig.3(c). This must come from the situation that the disturbance in jetting flow is increased by introducing the flow and is transported mostly in axial direction, then impinges on the exit wall of the cavity.

From these results of the order estimation and the experiments, the following two matters were clarified.

1. The pressure fluctuation affecting the inner walls of muffler cavity has a superior power as the noise source of flow induced noise in comparison with the share of jetting flow. (2) In the case with pulsating flow, the power of flow noise source was predominant on the inlet and side walls of the muffler cavity, while on the exit wall in the case with steady flow.

REFERENCES


FIGURE 1. Experimental set up

FIGURE 2. Power level of flow induced noise

FIGURE 3. Pressure fluctuation spectrum
Effect of Tabs on Reduction of Feed-back Resonance Generated in a Short Enlargement Muffler with High Speed Flow

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Effect of the tabs on the acoustic characteristics of resonance generated in short enlargement muffler (cavity) was examined of 0.3 and 0.5. As the results, tabs are very effective to decrease feedback resonance generated in the muffler. The reduction depends on de-correlate of the azimuth coherence vortices. For an area blockage of tabs to the nozzle exit greater than 3 %, feedback resonance is reduced considerably.

INTRODUCTION

When the jet vortex hits directly against the training edge of a short enlargement muffler, then the pressure wave is generated, and the wave returns to the leading edge of the muffler. This phenomena is repeated itself many times, then feedback resonance occurs. To decrease feedback resonance, many techniques inclining the downstream wall of the muffler have been tried. But the problem of these methods is that the muffler structure becomes too long.

Tabs have been shown to produce stream-wise vortices on both sides of the tabs. This device has been applied for reducing jet noise. Therefore, it is considered that the application of tabs to the reduction of feedback resonance generated in the enlargement muffler is effective.

The present paper examines the effect of tabs on the reduction of feedback resonance.

Results and Discussion

Schematic diagram of experimental apparatus is shown in Fig.1. High speed air flow is exhausted from a nozzle diameter of 20mm attached to a plenum chamber in which 5 screens are set. The aperture of the trailing edge is the same with the nozzle diameter. Cavity configuration is L(0 to 80mm)×H60mm×W60mm and its section is made of clear Plexiglas to visualize the flow. Triangular tabs are fitted to a circular nozzle being the inlet shape of the enlargement muffler. The heights of the tabs are 2 or 4mm, and tabs number is 1, 2, 3, 4 and 5. The area blockage of each tab is about 0.6% of the nozzle exit area for the tab height of 2mm, and is about 2.5% of the nozzle exit area for the tab height of 4mm. The cavity length L was varied from 0 to 80mm by moving the trailing edge.

The sound pressure spectra of the noise generated aerodynamically in the muffler was measured at a position of R=15cm and θ=60º by a 1/4 inch microphone.

The sound pressure spectra without a tab showed a lot of the complicated discrete frequencies. The dominant frequency is called cavity tone frequency in the present experiment. The cavity tone frequencies for Mach number 0.5 are shown in Fig.2 as a function of cavity length L. In the same figure, the calculation values of both resonance frequencies for cavity length direction and lateral direction are shown. The frequency modes for the lateral direction f_h and for the cavity length f_l are defined as n a_0/(2H) and n a_0/(2L) respectively, where n is integer, a_0 sound speed, H cavity height and L cavity length. The frequency mode of f_h is seen at around L=30mm and 55mm. The frequency mode of f_l is seen at around L=15, 40 and 65mm. These frequency modes were also produced at a Mach number of 0.3. The acoustic feedback mode (Rossiter’s equation) and the cavity length mode shows a similar frequency, therefore it is difficult to separate these frequencies. Tam & Ahuja stated that the feedback doesn’t occur below Mach number of 0.4. But in the present experiment the feedback was produced at Mach number 0.3. These phenomena might depend on the difference of experimental apparatus whether the box is installed or not.

Varying the shear layer by fitting tabs to the nozzle, jet vortex structure changes. As the vortex becomes smaller than before fitting the tabs, the hit of the vortex is alleviated. The pressure wave produced by the hit is decreased, and then noise generation is reduced.

The effect of tabs on the overall sound pressure level (SPL) is shown in Fig.3 as a function of the cavity length. With no tabs, an intensive sound pressure level of 140dB is generated. For tabs of h=2mm and n=5 which corresponds to blockage area ratio 3.1%, the sound pressure level is 115dB, that is, a noise reduction of 25dB is found. According to photographs of the flow visualization, the hit of the vortices against the trailing edge generates the discrete frequency tone. When the
vortex collapses before hitting against the trailing edge, the discrete tone does not appear. In the case of h=2mm and nt=3, it is found that the vortices hit against edge through the photographs of the flow visualization. Therefore, the lateral frequency mode is observed a little, but the frequency mode of length direction is not seen. The tabs decrease the acoustic resonance of cavity length direction more effective than that of lateral direction. By increasing tab numbers and tab height, the cavity tone is decreased. The appropriate tab numbers and tab height should be selected. In the present experiment, for the tab height of 2mm, tab numbers not less than 4 are proper, and for the tab height of 4mm, 2 tabs is good. The blockage of their tabs area to the nozzle area is 2.5% and 5%, respectively. Moreover, an additional experiment for the tab height of 3mm from 2 to 4 tabs was conducted. Noise reduction of 15dB (overall) for 2 tabs (blockage 2.8%) was obtained. Then, the blockage above 3% is effective to decrease the cavity tone.

To examine the reason reducing the cavity tone, the coherence function of the velocity fluctuation in the shear layer was measured using 2 hot wires. In this case, the side of the enlargement muffler was removed to do the measurement of coherence easily. As the results, the values of coherence function of the velocity fluctuation with tabs were lower than that without tab. It is found that tabs de-correlate the azimuth coherence vortices.

Conclusions

Tabs are very effective to decrease the cavity tone generated in the short enlargement muffler. The reduction depends on de-correlation of the azimuth coherence vortices.

REFERENCES

A Dynamical Countermeasure Method for the Non-stationary Wind-induced Noise based on Generalized Regression Model and Its Field Experiment

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In this paper, the problem of detecting the acoustic signals contaminated by random background and wind-induced noises is discussed from the statistical viewpoint. For evaluating the effects of wind-induced noise, we will pay our special attention to the measurement of wind velocities which cause the wind-induced noise. By grasping the statistical relationship between wind velocities and wind-induced noise in the form of generalized regression model, a new type wide-sense digital filter for detecting the acoustic signals under wind-induced noises is proposed from Bayesian viewpoint theoretically and experimentally.

INTRODUCTION

A wind-induced noise influences the acoustic measurements in the outdoor. The wind-induced noise fluctuates randomly owing to the temporal changes of wind velocity at the observation point and shows arbitrary probability distribution forms of non-Gaussian type. It is necessary to establish some systematic countermeasure methods for the wind-induced noise especially when measuring the low frequency acoustic signals, because the wind-induced noise includes components in a low frequency range which can not be reduced with use of the usual methods (such as the usage of wind screen and others). This is due to the fact that the physical mechanism of wind-induced noise is not known and the stationary property of wind-induced noise cannot be assumed.

Here, we will pay our attention to the fact that the wind-induced noise is generated from the temporal change of wind velocity, which can be measured separately when observing the acoustic data. The wind velocity changes relatively slower than the changes of the wind-induced noise and, in addition to the amplitude of wind velocities, the directions and the frequency components of them are also important. Then, for predicting the whole probability form of wind-induced noise, we should observe the velocities in different times and/or positions, and then consider the nonlinear multivariate correlation informations between the wind-induced noise and the wind velocities.

In this paper, from the above viewpoint, a new trial of dynamical countermeasure method for the wind-induced noise will be proposed by considering how to use effectively the multivariate information of wind velocities. More concretely, we will expand first the multivariate moment generation function hierarchical-ly and derive the multivariate form of generalized regression model of the wind-induced noise on the set of wind velocities. Next, by predicting various statistics of the wind-induced recursively, we evaluate the true acoustic signals dynamically under the wind-induced noise from Bayesian viewpoint.

Finally, the experimental confirmation of the proposed method has been confirmed by applying it to the actual data of field measurement.

THEORETICAL CONSIDERATIONS

The Generalized Regression Model of Wind-induced Noise on Wind Velocities

It is well-known that there is the averaged relationship between a wind-induced noise $v_1$ and a wind velocity $u^2$ (power) : $v_1 = \tau \cdot u^2$, (1)

where $n$ : a priori known integer, and $\tau$ : known proportional constant. But the wind-induced noise cannot be explained only with use of the averaged relationship in Eq.(1). Then, we will consider the deviations from Eq.(1) : $\varepsilon_1 = v_1 - \tau \cdot u^2$. (2)

Then, we will detect and utilise the correlation information embedded under $\varepsilon_1$ as much as possible, by using the information on the wind velocities powers. Here, we consider the $m$ observations of wind velocities $u^2_{1k}, u^2_{2k}, \ldots, u^2_{mk}$ ($u^2_{1k}$ is the present velocity $u^2_{2k}$ at the different time or positions) at the different time or positions. For grasping the whole probability form of the deviations of a wind-induced noise without minimum information losses, the joint probability function $P(\varepsilon_1, u^2_{1k}, u^2_{2k}, \ldots, u^2_{mk})$ should be considered.

By introducing the joint characteristic function

$M(\theta_1, \theta_2, \ldots, \theta_m) = \langle \exp[\varepsilon_1 \theta_1 + u^2_{1k} \theta_2 + \ldots + u^2_{mk} \theta_m] \rangle$

associated with the joint probability function on wind-induced noise and wind velocities, the hierarchically expanded joint characteristic function can be obtained as follows:
\( M(\theta, \theta_1, \theta_2, \ldots, \theta_m) = \prod_{j=1}^{m} \exp(\kappa_{0, \ldots, 1, \ldots, \theta_j} + \kappa_{0, \ldots, 2, \ldots, \theta_j}^2/2) \sum_{i_1 + \ldots + i_m = 0}^{\infty} g_{i_1, \ldots, i_m}(\theta) \) with use of the background noise and the wind velocities. The correlation information among the wind noise and the wind velocities can be reflected in each expansion coefficient hierarchically. By using the inverse Laplace transformation of the moment generating function, the statistical prediction of arbitrary moments of a deviation term \( R_k \) can be derived in the form of function on the observation wind velocities:

\[
< x^N_k | u_k^2, u_{k+1}^2, \ldots, u_{mk}^2 > = \sum_{i_1 + \ldots + i_m = 0}^{\infty} ( \frac{\kappa_{0, \ldots, 2, \ldots, 0}}{\sqrt{\kappa_{0, \ldots, 2, \ldots, 0}}} )^{i_1} / \sum_{i_1 + \ldots + i_m = 0}^{\infty} g_{i_1, \ldots, i_m}(\theta) \]

\[
(g_{ii, \ldots, ij}(\theta))^{(i_1, i_2, \ldots, i_m = 0)} \]

\[
(\kappa_{0, \ldots, 2, \ldots, 0})^{i_1} / \sum_{i_1 + \ldots + i_m = 0}^{\infty} g_{i_1, \ldots, i_m}(\theta) \]

\[
\prod_{j=1}^{m} H_{ij} \left( \frac{\kappa_{0, \ldots, 2, \ldots, 0}}{\sqrt{\kappa_{0, \ldots, 2, \ldots, 0}}} \right)^{i_j} / \sum_{i_1 + \ldots + i_m = 0}^{\infty} g_{i_1, \ldots, i_m}(\theta) \]

where \( g_{ii, \ldots, ij}(\theta) = (\frac{\partial}{\partial i} \frac{\partial}{\partial j})^N \prod_{j=1}^{m} H_{ij} \left( \frac{\kappa_{0, \ldots, 2, \ldots, 0}}{\sqrt{\kappa_{0, \ldots, 2, \ldots, 0}}} \right)^{i_j} / \sum_{i_1 + \ldots + i_m = 0}^{\infty} g_{i_1, \ldots, i_m}(\theta) \).

That is, the generalized multivariate regression model can be realized for predicting the arbitrary moments of the wind noise with use of the wind velocities.

**A Wide-Sense Digital Filter for Detecting Acoustic Signals Embedded under Wind-induced noise**

In the same analytical viewpoints, a dynamical signal detection method under the background and the wind noises can be derived. Here, based on the analysis based on the additive property of energy quantities and the physical mechanism, the system and observation equations for the acoustic systems are formulated as:

\[
x_{k+1} = F_k(x_k, u_k), \quad y_k = H_k(x_k, v_k),
\]

where \( x_k \) denotes the unknown acoustic signal at a time stage \( k \), and \( y_k \) denotes the contaminated observation. \( v_k \) and \( u_k \) are the background noise and the wind noise.

For detecting the acoustic signal \( x_k \) with use of the observation recursively, the conditioned joint moment generating function \( M^*(\theta, \theta_1) (= \exp[x_k \theta + y_k \theta_1] \mid Y_k >) \) should be considered. Here \( Y_k \) is the set of past observations \( \{y_1, y_2, \ldots, y_k\} \).

Then, by considering Bayes theorem, the unified algorithm of estimating the unknown acoustic signals under these random noises can be obtained:

\[
< x^N_k | Y_k >= \sum_{j=0}^{\infty} g^*_j(0)^*H_j \left( \frac{y_k - \kappa_{0, 1}}{\sqrt{\kappa_{0, 2}}} \right)^j
\]

\[
/ \left( 1 + \sum_{j=0}^{\infty} g^*_j(0)^*H_j \left( \frac{y_k - \kappa_{0, 1}}{\sqrt{\kappa_{0, 2}}} \right)^j \right).
\]

Finally, by combining it with the unified prediction algorithm: \(< x_{k+1}^N \mid y_k >=< F_k^N(x_k, u_k) \mid Y_k >\), the dynamical state estimation algorithm in a recursive form can be realized.

**EXPERIMENTAL CONFIRMATIONS**

The effectiveness of proposed estimation theory has been experimentally confirmed by applying it to the actual acoustic data in the outdoor field under the wind-induced noises (omitted owing to page-limit).

**CONCLUSIONS**

In this paper, for the acoustic system in an outdoor environment, a new derivation of the systematic countermeasure method for the wind noise was proposed with use of the wind velocities.

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Low Frequency Noise in Automotive Interiors
due to Flow through Window Openings

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We investigate how the acoustic field created by air inflowing through the side windows of automobiles of various sizes depends on the vehicle speed. Extremely large sound pressure levels are found near the frequency of the fundamental mode of the vehicle’s Helmholtz resonance (about 20 Hz). Much smaller peaks are also found at the frequencies corresponding to higher harmonics. A positive, tight correlation exists between the peak frequency and the vehicle speed. The peak sound pressure levels increases with the vehicle speed, up to a critical speed, and declines afterwards. In absolute terms, much larger sound pressure levels were found inside a sub-compact car, as compared to a four-door sedan and to a station wagon.

INTRODUCTION

Interior noise in the low frequency region is usually of minor concern inside contemporary cars. However, the presence of open sunroofs and/or side windows can easily result in very high amplitude, low frequency noise, which strongly interferes with the driver’s concentration and wakefulness, causing considerable discomfort and even possibly impairing his ability to provide safe driving [1].

This contribution reports the first results of an ongoing investigation aimed at characterizing the low frequency sound pressure field in vehicle interiors, as a function of vehicle type and speed, as well as of the opening sizes, number and locations. Here we focus on the relation between the peak sound pressure level and frequency and the vehicle speed.

METHODS

Three vehicles were tested, including one sub-compact city car (SC), one sedan (S), and one full size station wagon (SW). Measures were taken positioning a microphone with extended sensitivity to low frequency waves (Brüel & Kjær type 4193) at a distance of about 10 cm from the right ear of the driver. Experiments were carried out between March and December 2000, with different vehicle speeds, ranging from 80 to 140 Km/h, and a fully open rear right window. Typical test durations were set at around 90 seconds.

Narrow band FFT spectra were calculated with bandwidths 0 – 100 or 0 – 200 Hz and resolutions 0.5 Hz. One third octave band spectra were synthesized from narrow band spectra. Data analysis was performed using the software Kyplot (version 2.0) to calculate best fitting models.

RESULTS

Spectral shape

Figure 1 shows the FFT spectrum measured inside the SC vehicle at 100 Km/h with fully open right rear window. Despite its specific character, this spectrum can be taken as generally representative of frequency spectra collected in this study, since they all share the same basic features.

FIGURE 1. Frequency spectrum of vehicle SC

After an initial area of low signal at very low frequencies (up to 8 - 12 Hz), the sound pressure level shows a very fast rise, culminating with a sharp peak at frequencies f1 in the range 16 – 22 Hz depending on the various parameters affecting the system. This is followed by an equally sharp decline. Beyond 30 Hz, a more shallow downward trend sets in, which extends to about 80 – 100 Hz. Superposed on this “noise floor” are the various peaks due to the higher harmonics, at
frequencies n×f₁. Their amplitudes are much smaller (at least 25 – 30 dB) than that of the first peak.

The exciting mechanism is provided by the aerodynamical noise due to flow instabilities in the boundary layer between the vehicle interior and the outer flow [2]. These instabilities results in vortex shedding and the convection of discrete vortexes over the opening determines the acoustic excitation of the cavity. Acoustic emission is triggered by the interaction of the vortexes with the downstream edge of the opening, and shows a characteristic frequency

\[ f_a = \frac{N_s \cdot V}{d} \]

where \( N_s \) is the Strouhal number, \( V \) is the vehicle speed and \( d \) is the opening streamwise length. The actual emission takes place at a frequency dictated by the acoustical response of the vehicle interior. The latter is very well approximated by a Helmholtz resonator [3]. Under the circumstances found in vehicles, the fundamental proper mode has a frequency

\[ f_h = \frac{c_s}{2\pi} \left( \frac{2}{\pi^{1/2}} \frac{A^{1/2}}{V} \right)^{1/2} \]

where \( c_s \) is the sound speed, \( A \) is the window opening area, and \( V \) is the interior volume. For the typical values assumed by these parameters in tested vehicles, \( f_h \) is of order 20 Hz.

Peak frequency and velocity

Table 1 reports the peak frequencies and the peak narrow band sound pressure levels in vehicles SC and SW. The very large difference between the sound levels is likely due to a combination of a reduced inflow and a broader emission spectrum is vehicle SW. The non-monotonic trend of the SPL with velocity is possibly related to the matching of the instability characteristic wavelength and the opening length.

<table>
<thead>
<tr>
<th>Velocity (m/s)</th>
<th>Vehicle SC</th>
<th>Vehicle SW</th>
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<tr>
<td>f-Peak (Hz)</td>
<td>SPL (dB)</td>
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<td>100</td>
<td>18</td>
<td>126.9</td>
</tr>
<tr>
<td>110</td>
<td>19</td>
<td>126.8</td>
</tr>
<tr>
<td>120</td>
<td>19.5</td>
<td>127.5</td>
</tr>
<tr>
<td>130</td>
<td>20</td>
<td>128.2</td>
</tr>
</tbody>
</table>

Table 1. Peak sound pressure levels and peak frequencies

Figure 2 shows the peak frequency as a function of velocity for all three tested vehicles. A very clean, systematic rising trend can be observed as a function of vehicle speed, in agreement with the results of a previous study [4].

This trend is very robust, as it appears to be present in all vehicles of different shapes and sizes. A power law \( f \propto V^\alpha \) has been found to provide good fitting to data. Results indicate similar trends (\( \alpha \approx 0.5 \)) for the SC and the S vehicles. A matched pair t-test shows however that SPL values in the S vehicles are significantly larger (t = 6.54, P < 0.05) than in the SC vehicle. A more shallow slope characterizes the SW vehicle. The same behavior is replicated by the higher harmonics, although they prove at times hard to locate due to unfavourable signal to noise ratios. Finally, the spectrum becomes more and more spread out as velocity increases, with lower resonance peaks and higher high frequency "noise floors", pointing to a rising trend of damping in the system.

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REFERENCES

Prediction of airplane interior noise due to flow over small steps
Part 2: Non-resonant sound transmission

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Flow over small steps on the exterior of an airplane causes aerodynamic pressure fluctuations that are up to 30 dB greater than those under a turbulent boundary layer on a surface without pressure gradients. Although of small extent, regions with such excitation may contribute substantially to the interior noise in airplanes, particularly if well-radiating structures like windows are excited. The aim of this investigation is the prediction of interior noise of airplanes resulting from flow over such steps, common on airplane exteriors. This requires derivation of the relations governing the vibration and acoustic radiation of thin-walled structures in the case of excitation by such pressure-fluctuation fields. Existing interior-noise prediction procedures in the framework of statistical-energy analysis are modified to reflect the peculiarities of these fields, which differ from those under a turbulent boundary layer in terms of correlation scales and non-uniformity scale. This paper is limited to the transmission due to non-resonant excitation of the structure. A paper delivered at Inter-noise 2001, The Hague, Netherlands describes the somewhat greater resonant contribution to the radiated field. This work is the result of cooperation between TsAGI, Moscow and Boeing, Seattle.

INTRODUCTION

Relations are found for prediction of structural-acoustic radiation and airplane interior noise due to excitation by pressure fluctuations in flow over steps. In reference \cite{1} the relations associated with the resonant behavior of the structure were obtained. The aim of the present investigation is to obtain the relations associated with non-resonant behavior of the structure and are derived by studying a model task. The results were obtained in the course of scientific research contracts between the Boeing Company (USA) and the Central Aerohydrodynamics Institute (Russia).

TASK STATEMENT AND SOLUTION

METHOD

We examine a thin infinite panel, with mass but no stiffness, separating two half spaces (\Omega_1 and \Omega_2). We use an orthogonal system of coordinates \{x_1, x_2\} in the plane of the plate, completed by a coordinate \(x_3\) positive in half-space \Omega_2. The non-uniform pressure-fluctuation field excites the plate from half-space \Omega_1. The perturbation of the air in \Omega_1 and \Omega_2 is assumed small and non-vortical. The speed of sound (c_0) is taken to be equal in \Omega_1 and \Omega_2 but the air density in the general case different (\rho_1 and \rho_2, respectively).

The task is prediction of the spectral density of the sound pressure in half-space \Omega_2. This task is solved by techniques analogous to those used for acoustic radiation from a plate excited by a random pressure-fluctuation field \cite{2}.

From the multiplicative presentation of the space-correlation spectra \cite{2} we will obtain relations for the frequency/wavenumber spectra of the non-uniform pressure-fluctuation field:

\phi_\Omega(k_1, k_2, \omega) = \Phi_\Omega(\omega)\phi(k_1, k_2, \omega)\phi_2(k_2, \omega) \quad (1)

\phi(k_1, k_2, \omega) = \frac{2}{(2\pi)^2} \left[ 2a(a+c)+(k_1-k_1')(k_1+k_1') \right] + \frac{(a+c)(a-c)-(k_2-k_2')(k_2+k_2')}{[2a(a+c)+(k_2-k_2')(k_2+k_2')]^2} \left[ k_1+k_1' \right]^2 + \frac{2a(a-c)+(k_1-k_1')(k_1+k_1')}{[2a(a-c)+(k_2-k_2')(k_2+k_2')]^2} \left[ k_2+k_2' \right]^2 + \frac{(a+c)^2-(k_2+k_2')(k_2+k_2')}{[a+c]^2+(k_2+k_2')^2} \left[ k_1+k_1' \right]^2

\varphi_\Omega(k_2, \omega) = \frac{\Lambda_2}{\pi^2 \Lambda_1 (\hat{a} L)^2} \quad a = 1/2L \quad c = \sqrt{\Lambda_1 \Lambda_2} \quad \gamma_i = \omega U_{ih}

Here \Phi_\Omega(\omega) is the spectral density of the pressure-fluctuation field in the region of maximal intensity, \(L\) the scale of non uniformity of the pressure-fluctuation field, \(\Lambda_1\) the longitudinal correlation scale, \(\Lambda_2\) the transversal correlation scale, \(U_{ih}\) the phase velocity of the pressure-fluctuation spectral components, \(\omega = 2\pi f\), and \(f\) the frequency in Hz.

In this case the following relation for prediction of the sound pressure spectral density at (\(x_1, x_3\)) in \Omega_2 is obtained:
\[ \Phi_{p2}(x_1, x_2, \omega) = \frac{\pi k_0^3}{4} \int \int \int \Phi_0(\lambda_1, \lambda_2, \omega_0) \exp \left[ \pm ik_0 (x_1 + \lambda_1^2 - \lambda_2^2 x_2) \right] \times \left[ \frac{1}{\sqrt{1 - \lambda_1^2 - \lambda_2^2 - i \nu_1}} \right] d\lambda_1 d\lambda_2 d\omega_0 \]  

Here  
\[ \lambda_1 = k_1/k_0 \quad \lambda_2 = k_2/k_0 \quad \omega_0 = \omega/c_0 \quad \tau_1 = [c_0(p_1 + p_2)/m \omega]^2 \quad \tau_2 = \tau_1 \rho_0 \rho \]

The three-dimensional integral can be calculated numerically. However, when the distance between observation point and step (R) is large in comparison with the sound wavelength \((k_0 R > 1)\), it is possible to give an asymptotic analytical relation:

\[ \Phi_{p2}(\omega) = \frac{\pi \tau_2}{2 R} \Phi_0(\omega) \kappa_1(\omega) \frac{k_0^2 \Lambda_2}{1 + \left[ k_0 \Lambda_2^2 \right]^2 (1 + \tau_1)} \times \left[ \frac{1}{\sqrt{1 + [k_0 \Lambda_2^2]^2 - \tau_1}} \sqrt{1 + \tau_1} \right] \]

(3)

When the component \( \Phi_1(\kappa_1, \kappa_2, \omega) \) of a frequency-wavenumber spectrum in (2) varies slowly in the range of integration (which is true in almost all cases for fields at small steps), it is possible to give another analytical relation for small distances between observation point and step \((k_0 R << 1)\). This relation is not presented here due to space limitations.

The increase in sound pressure level related to the inertial behaviour of the structure, when excited by interaction of the turbulent boundary layer with a step, is predicted by the following relation:

\[ \Delta L = 10 \cdot \log \left[ 1 + \Phi_{p2}(\omega) / \Phi_{p2}(\omega) \right] \]

(4)

Here \( \Phi_{p2}(\omega) \) is the sound pressure spectral density predicted according to (2) for the case of flow over small steps and \( \Phi_{p2}(\omega) \) is the same quantity for the case of non-gradient turbulent boundary-layer flow on a smooth surface. This last quantity is predicted according to the well known relation (see [2]):

\[ \Phi_{p2}^{(BL)}(\omega) = \frac{\pi k_0^3}{4} \ln \left( 1 + \frac{1}{\tau_1} \right) \Phi_{BL}(\omega) F(\alpha) \]

(5)

Here \( \Phi_{p2}^{(BL)}(\omega) \) is the spectral density of turbulent boundary layer pressure fluctuations [3] and \( F(\alpha) \) is a function of dimensionless parameters [2].

As an example, function \( \Delta L \) presented in Fig. 1 was predicted for the case of excitation of a flat flight-deck window pane at different values of the step height and at an observation-point distance of \( R = 0.1m \). The initial parameters of the calculation were the same as in case of resonant energy transmission [1].

Comparison of figure 1 [1] with figure 1 of this paper shows that the increase in sound pressure levels inside the airplane, due to the inertial behaviour of the structure, are smaller than those due to its resonant behaviour.

**REFERENCES**

Determination of Source Impedance of Fluid Machines by Lumped Element Model

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Fluid machines (internal combustion engines, pumps, fans) with inlet and outlet systems form complicated acoustical systems. To understand sound generation and to design properly the pipe or duct system for the fluid machine, it is necessary to know the source impedance of the machines. In most cases there is no general theory to calculate it and the impedance has to be determined experimentally. However, this is time consuming and costly and mathematical models for the source impedance would be a help. In the paper as an example, a complete lumped-impedance model for a centrifugal fan is derived and validated by experiments. The model can give satisfactory results at low frequencies, where the wave propagation effects can be neglected. However most of the acoustical energy of fluid machines is generated at low frequency region. In order to verify the derived lumped-impedance model, measurements were carried out on centrifugal fans. By comparison with measured curves the predicted source impedance were wound to agree well up to approximately 400 Hz.

INTRODUCTION

Fluid machines such as pumps, fans etc., can generate high sound levels in connected pipe and duct systems. The sound power generated by the source depends both on the impedance of the duct system and the source. The impedance of the duct system can usually be easily calculated but very few investigations can be found concerning the calculation of the impedance of the ducted fans. Yeow [1,2] used a lumped impedance-element approach to study the influence of the fan’s structural parts impedances on the radiated sound power. Lavrentjev [3] used similar model and scaling laws to determine the 1-port source data for fans. The aim of this paper is to develop a complete source impedance model for centrifugal fans.

MATHEMATICAL MODEL

The lumped model of the fan consists of four different basic types of elements. In the first type of element the kinetic energy of the gas dominates over the potential energy (due to compression) and the element is characterised by the inertia of the gas (mass type element, $Z_L$). In the second type of element the potential energy of the gas dominates over the kinetic energy of the flow (spring type element, $Z_C$). The third type of element is the resistive element, which causes a pressure drop when a gas flow passes (resistive element, $Z_R$). The fourth element needed is a termination element, which can describe the radiation from the fan inlet opening ($Z_{\text{term}}$).

Taking into account the elements described, a typical centrifugal fan (Figure 1) can be split into lumped-elements as illustrated in Figure 2 and 3.

![Figure 1](image1.png)

**FIGURE 1.** Geometry of the tested centrifugal fans. 1 - outlet neck, 2 - volume of the casing; 3 - channels formed by the blades of the impeller; 4 - inlet necks.

The inlet and outlet necks and the channels formed by the blade rows of the impeller can be modelled either as mass type element (Figure 2) or spring type element (Figure 3).
The choice depends on the relative lengths of the blade rows.

Since blade rows can also be expected to introduce certain losses, they will include also resistive elements. Quantities applied in the elements of models and characterizing fans are purely geometrical (volumes and lengths) except resistive elements which have to be determined by theoretical experiments.

**EXPERIMENTS**

Models described here were used successfully for different fans. As an example, a model, presented in Figure 3 was applied for the centrifugal fan (13 backward curved blades, $n=2800$ 1/min). Source impedance was determined experimentally for two flow rate cases (100% and 20%) by using 2-microphone method and 1-port model approach. The comparison between model and measurement results is shown in Figure 4 and Figure 5. As one can see the discrepancy between predicted and measured curves are small.

**CONCLUSIONS**

By using the lumped impedance model the centrifugal fans impedance can be determined. For different types of fans different models of impeller may be applied.
On reduction of acoustic disturbances in supersonic separated flow by means of passive flow control

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Effect of passive control on acoustic disturbances caused by flow unsteadiness in a shock wave/turbulent boundary layer interaction region at supersonic speeds was studied experimentally. The shock wave was generated by blunted fin mounted on a wind tunnel sidewall. The cavity covered by perforated cover and located in the interaction region was used as a passive control device. Tests were conducted at Mach numbers from 2 to 4 and unit Reynolds numbers from $9.6 \cdot 10^6$ to $51.2 \cdot 10^6$. Spectra and integral levels of wall pressure fluctuations were measured during experiments as well as static pressure fields were measured. It was demonstrated that passive flow control can sufficiently decrease acoustic disturbances caused by flow unsteadiness in the interaction region.

INTRODUCTION

The shock wave/boundary layer interactions occurs in supersonic flows around such elements of modern aircrafts as inlets, control surfaces, pylons etc. Flow in interaction region has a complex structure and can sufficiently effect on aircraft aerodynamics [1]. In addition to these, separation is a cause of flow unsteadiness, and its presence leads to a high level of aeroacoustic loads on a vehicle elements [2]. Present paper concerns the flow around control surface-fuselage junction at supersonic speeds. The main object was to determine the effectiveness of passive control at Mach numbers $M_\infty = 2 - 4$ for reduction of flow unsteadiness in interaction region. Flow was studied experimentally due to its complexity. The blunted fin was used as a model [3]. The shock generated by fin interacts with a turbulent boundary layer on a flat plate (wing tunnel sidewall). Due to sufficient pressure overrun in a shock wave, the flow decelerates and main separation zone occurs. Inside an interaction zone the horseshoe vortex, developing downstream forms.

EXPERIMENT

Experimental facility and model

Experiments were carried out in supersonic low-turbulent blowdown wind tunnel with cross section $0.21 \times 0.20$ m\textsuperscript{2}.

Wind tunnel equipped with removable nozzles for freestream Mach number variation. Unit Reynolds numbers are varying by means of plenum chamber pressure variation. Tests were conducted at Mach numbers $M_\infty = 2, 3, 4$ and plenum chamber pressure from $2 \cdot 10^5$ Pa to $2 \cdot 10^5$ Pa, while unit Reynolds numbers were from $9.7 \cdot 10^6$ m\textsuperscript{-1} to $51.2 \cdot 10^6$ m\textsuperscript{-1}. Dimensionless $D/\delta$ parameter (where $D$ is a bluntness diameter of leading edge, and $\delta$ is a boundary layer thickness) was changed from 0.287 to 0.417.

Model (Figure 1) was consisted of shock wave generator and passive control device. As a shock wave generator the rectangular blunted fin was used. The fin was 120 mm long, 10 mm wide and 10 mm thick. The bluntness diameter was 10 mm. Fin was mounted on a remote controlled drive, which allowed to move the fin parallel to wing tunnel axis of symmetry with a minimal step of 5 mm by means of micrometer screw. Passive control device consisted of plug with cavity mounted in a port on a wind tunnel sidewall. Cavity diameter was 155 mm, and cavity depth was 114 mm. Cavity can be covered flush by solid or perforated cover. The thickness of cover was 4.5 mm. When model without control device was studied, the cavity was covered by solid cover. In case of passive control the perforated cover was used. Perforated cover have had 4256 holes 0.8 millimeters in diameter. The perforation ratio was 6.25%.

The pressure coefficient was calculated by static pressure measurements in a single orifice, relative to which the fin was shifted. The plug was turned in such a way that measuring orifice was shifted relative to the axis of symmetry of the model by 9.89 and 14.0 mm. Due to this, the measurements of $C_p$ in three sections of $Y = const$ plane were performed. Piezoresistive pressure transducer Endevco type 8514-10 with case diameter of 1.65 m-
m was used for studying the effect of passive control on wall pressure fluctuations. Transducer was mounted in a miniature plug placed in bypass cavity. Pressure fluctuations spectra were measured at \( Y = 0 \) section with a step of 5 mm. These were measured directly during the experiment in third-octave frequency bands by spectra analyzer. Analyzer was consisted of type 2607 measuring amplifier, type 1614 filter and type 2305 plotter by "Brüel & Kjær". In present experiments 100 Hz - 25 kHz frequency band was studied. Using measured spectra the integral level of pressure fluctuations \( L_\Sigma \) was determined, which characterized the magnitude of acoustic disturbances.

**RESULTS**

**Spectral characteristics of the static pressure fluctuations**

The flow in separation region is unsteady. It means that place and structure of shock waves system, in which the shock decomposes near the wall, is time-dependent. This unsteadiness produces acoustic disturbances. Pressure fluctuation spectra, measured along the model symmetry axis in section \( Y = 0 \) indicates the strong flow alternation in interaction region, while spectra measured in undisturbed flow are characterized by low amplitude in whole frequency range. Example of the wall pressure fluctuations spectra measured in 100 Hz - 25 kHz frequency band in separation zone \( x/D = -2.5 \) are presented on Figure 2. The pressure fluctuations spectra are characterized by wide-band power spectra with high energy concentration in low-frequency region, conditioned by unsteady shock wave displacement. Air bypass employment leads to reduction of magnitude of disturbances in interaction region. It should be noted that for plot presented, the maximal reduction of pulsation amplitude happens in low frequency domain. This leads to redistribution of wall pressure fluctuations amplitudes by frequencies. Mentioned above indicates that unsteadiness caused by shock displacements becomes lower.

**Integral level of pressure fluctuations**

Plot of integral level of pressure fluctuations on the wall surface in section \( Y = \text{const} \) for solid and perforated cover is presented on Figure 3. Integral level of wall pressure fluctuations is characterize the magnitude of acoustic disturbances. It should be pointed out that level of acoustic disturbances caused by shock wave/boundary layer interactions is extremely high and achieves 180 decibels at Mach 2.

**FIGURE 3.** Example of an integral level of wall-pressure fluctuations, \( M_\infty = 2 \).

**CONCLUSIONS**

Experimentally studied interaction of shock wave, generated by blunted fin, with turbulent boundary layer at supersonic speeds.

For reduction of separation intensity the passive flow control was used. Passive control device was consisted of cavity covered by perforated surface with perforation ratio was 6.25%.

It was demonstrated that passive control allows to decrease the length of separation zone, to relieve the pressure on interaction region, and to lower acoustic disturbances in separation zone.

Maximal reduction of integral pressure fluctuations level was up to 5.7 decibels (at Mach 3).

Effectiveness of flow control device was demonstrated at all Mach and Reynolds numbers traced.

**REFERENCES**