INTAKE NOISE CANCELLATION USING A MANIFOLD BRIDGING TECHNIQUE

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ABSTRACT

Automobile manufacturers have expended considerable efforts to attenuate the many noise sources perceived within the passenger compartment with varying degrees of success. Given that these dominant noise sources have been attenuated, induction noise has become more noticeable. The present study investigates the feasibility of using a non-conventional noise cancellation technique. The investigation has attempted to improve the acoustic performance of the induction system by introducing a bridge between the exhaust and intake manifolds. The effectiveness of such a technique is investigated using Ricardo WAVE, a computational engine simulation technique that uses a one-dimensional finite-difference formulation. Graphical results using $1/12^{th}$ octave frequency spectra and three dimensional colour maps of both an unmodified and a bridged engine are presented for both steady state and transient engine cases. A sound quality analysis is also presented using the psychoacoustic metrics of Loudness, Fluctuation Strength and Articulation Index. While a reduction in overall sound level was achieved, an additional benefit of this technique proved to be in the realized sound quality of the induction noise with the implementation of the manifold bridge. This investigation continues with verification of the theoretical model to experimental measurements on a dynamometer.

RÉSUMÉ

Les constructeurs d'automobiles ont fait des efforts considérables pour atténuer les nombreuses sources de bruits perçus dans l'habitacle, avec un succès variable. Ces bruits dominants ayant été atténués, on a fini par percevoir plus clairement le bruit de l'induction. La présente étude examine la possibilité d'annuler ce bruit par une technique innovatrice. On a tenté d'améliorer la performance acoustique du système d'induction en établissant un pont entre les tubulures d'admission et d'échappement. L'efficacité d'une telle technique est testée à l'aide de Ricardo WAVE, technique de simulation virtuelle de moteurs qui emploie une formulation de différence finie à une seule dimension. Des résultats graphiques utilisant des spectres de fréquence de 1/12 octave et des plans en couleur à trois dimensions, d'un moteur non modifié et aussi d'un moteur équipé d'un pont, sont présentés pour des moteurs tournant de manière continue et de manière intermittente. Une analyse de la qualité du son est présentée également, qui utilise les mesures psychoacoustiques de Force, d'Amplitude de fluctuation et d'Index d'articulation. Une réduction du niveau global de bruit a effectivement été accompli, et cette technique a aussi donné un avantage supplémentaire: la qualité du bruit d'induction s'est améliorée, grâce à l'installation du pont. On poursuit la recherche en vérifiant le modèle théorique au moyen de mesures expérimentales faites sur un dynamomètre.

1. INTRODUCTION

There are many sources of noise in the modern day automobile which include the combustion process of the engine, exterior wind noise, tire noise as well as exhaust and intake noise. Manufacturers have responded with stiffer and better acoustically insulated bodies, better aerodynamics, improved tire technology and quieter mufflers. This has resulted in an overall reduction of sound pressure levels in the passenger cabin, which has also resulted in a more acute awareness of other sources of noise, specifically induction noise. Given this, greater emphasis is being given to the study of induction noise and what can be done to lessen its impact on the consumer.

Traditional acoustic attenuators of intake noise include simple Helmholtz resonators or perhaps adaptive passive systems which allow the acoustic resonator volume to react according to engine RPM. However, due to the growing popularity of smaller vehicles with increasing limits on under hood space, it is becoming more difficult to facilitate the installation of these traditional methods of intake noise attenuation.

As a result, new design concepts for induction noise attenuation are being investigated by automotive engineers. One such approach is active noise cancellation where preliminary work to date has shown promising results.

The objective of this work was to investigate the feasibility of attenuating automotive induction noise using active noise cancellation through the implementation of tuned exhaust noise feedback through the intake system. This process typically utilizes a computer controlled speaker as the negating noise source. However here, the reduction of acoustical energy in the intake system would be realized by using exhaust noise as the effective dynamic noise source instead of a speaker. This is accomplished with the introduction of an open physical bridge inserted between the exhaust and intake manifold of the engine.

2. ACOUSTICS OF AIR INDUCTION SYSTEMS

The noise emitted from an automotive induction system is the result of a combination of two processes. The first process is the propagation of pressure pulses generated when the intake valve opens to the cylinder which has a pressure greater than atmospheric. A second pulse occurs when the intake valve closes [1]. The repetition of these pulses result in the oscillation of intake air at the natural frequency of the inlet passage column. The frequency of these pulses is further reduced to approximately 80 to 150 Hz in the engine firing range due to the influence of the intake system ducting, silencers and air cleaner package. Figure 1 illustrates these oscillations with respect to the timing of the inlet valve.

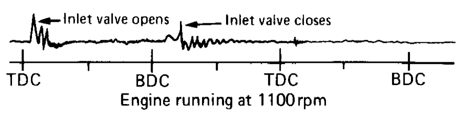

Figure 1: Inlet Noise Oscillogram[1].

The second process is flow generated, or gas flow noise, which is the result of turbulence of the mean flow traveling across the valve seat. This high velocity flow generates a high frequency broad spectrum noise above 1000 Hz. This noise, however, is effectively attenuated by the air cleaner and transmission path between the engine and passenger compartment. Consequently, this flow generated noise does not warrant concern [2]

For the purpose of this study, the propagation of noise in the intake system is assumed to be a one-dimensional wave. It has been shown in the past that the consideration of this noise as plane acoustic propagation has been able to provide reliable prediction of automotive intake noise [3].

2.1 The One-Dimensional Wave Equation for Pulse Noise

A brief derivation of the one-dimensional wave equation is presented I this section. The wave equation illustrates the propagation of acoustic pressure fluctuations. The acoustic variables of interest are pressure p, density ρ, and particle velocity u.

Consider a fixed volume of a duct, as shown in Figure 2, with a cross section S and an arbitrary length dx in the x direction.

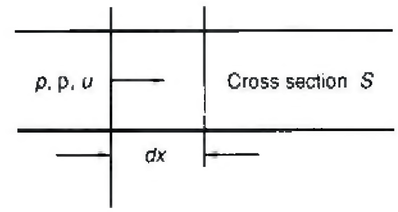

Figure 2. Control Volume with Wave Propagation through a Duct [4]

The total density of the fluid in the control volume in the duct is given as:

$$\rho_{tot} = \rho_o + \rho$$

where, ρ_o is the initial uniform fluid density and ρ is the change in the fluid density caused by the acoustic wave. Applying the continuity equation to the control volume, the rate of mass inflow to the control volume is given as:

$$\rho_{tot}uS - \left[\rho_{tot}u + \frac{\partial(\rho_{tot}u)}{\partial x}dx\right]S = -\frac{\partial(\rho_{tot}u)}{\partial x}dxS$$

Any increase of mass in the control volume must be balanced by the net inflow of mass given above. After simplification, we have:

$$\frac{\partial(\rho_{tot})}{\partial t} + \frac{\partial(\rho_{tot}u)}{\partial x} = 0$$

Linearizing the above equation and considering only the first order terms, the mass continuity reduces to,

$$\frac{\partial\rho}{\partial t} + \rho_0\frac{\partial u}{\partial x} = 0$$

One can also apply the principal of momentum conservation to the same fluid element and using the argument above, express the net pressure in the fluid as:

$$p_{tot} = p_o + p$$

where, p represents the acoustic pressure. The net force acting on the element in the x direction is then:

$$p_{tot}S - (p_{tot} + \frac{\partial p_{tot}}{\partial x}dx)S = -\frac{\partial p_{tot}}{\partial x}dxS$$

Linearizing the above equation and considering only the first order terms, the momentum continuity reduces to,

$$\rho_0\frac{\partial u}{\partial t} + \frac{\partial p}{\partial x} = 0$$

Differentiating the linearized equation of mass conservation with respect to time and the linearized equation of momentum conservation with respect to position we get:

$$\frac{\partial^2 p}{\partial x^2} - \frac{\partial^2 \rho}{\partial t^2} = 0$$

The acoustic wave propagation is assumed to be adiabatic where a change in pressure is directly related to a corresponding change in density by a proportionality constant of the square of the speed of sound 'c'. Substitution of this into the previous equation gives the one-dimensional wave equation for the propagation of acoustic pressure fluctuations in the manifold duct.

$$\frac{\partial^2 p}{\partial x^2} - \frac{1}{c_0^2}\frac{\partial^2 p}{\partial t^2} = 0$$

This relationship illustrates the way that acoustic pressure fluctuations act with respect to the co-ordinate distance x and with respect to time.

3. CONVENTIONAL CONTROL METHODS

Before presenting the non-conventional noise cancellation technique, review of some of the traditional methods of intake noise attenuation and active noise control (ANC) are discussed below.

3.1 Passive Attenuation

Automotive intake noise is traditionally attenuated through the application of passive control techniques. These techniques are usually the simplest and least expensive form of attenuation but do not always yield the best results. Two primary categories of passive noise control are: 1) path redirection of the acoustic energy; and 2) reduction of the acoustic energy flow, usually through either absorption with acoustic insulation or by changing the acoustic impedance of the power output, perhaps through the use of a sudden cross section change.

The Helmholtz resonator is one of the most common passive noise control technique used in automotive induction systems. When acoustic energy travels down a tube or pipe, a specifically chosen attached volume can be used to attenuate the traveling noise. This technique is particularly effective when the unwanted noise consists of a narrow frequency band and the volume, or resonator, is tuned to the target frequency. A schematic of a typical resonator in an automotive induction system is given in Figure 3.

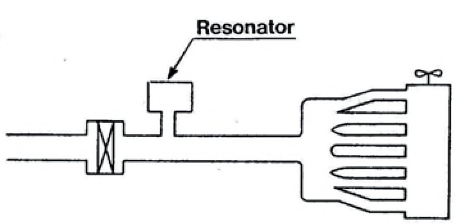

Figure 3. Resonator in Automotive Intake System [5].

Another common passive noise control technique is the use of an expansion chamber. These may or may not include absorptive elements. An expansion chamber without the presence of absorbing material is called a reactive muffler. Here, the performance of the muffler is dependant entirely on the geometrical shape of the expansion chamber. A cutaway of a multi-chambered muffler is illustrated in Figure 4. If sound absorbing material is integral to the attenuating abilities of the muffler, it is referred to as a dissipative muffler. Such designs are generally best suited for controlling frequencies higher than 500 Hz.

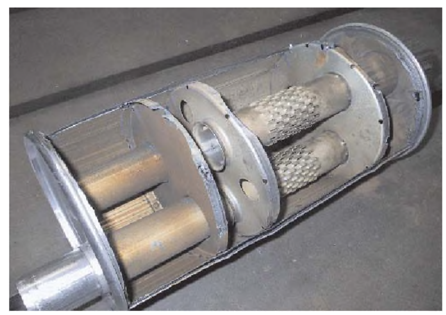

Figure 4. Cutaway of Multi-Chambered Muffler.

3.2 Active Attenuation

Active noise control (ANC) attenuates unwanted noise by canceling the unwanted noise waves through the introduction of a second set of noise waves which are equal in amplitude but opposite in phase to the undesired acoustic signal. The traditional method of ANC most often utilizes loudspeakers to generate a sound field to cancel the existing sound field.

The most commonly found types of active noise control systems are the adaptive feedforward, the adaptive feedback and the wave synthesis system. A typical adaptive feedforward active noise control system is shown in Figure 5.

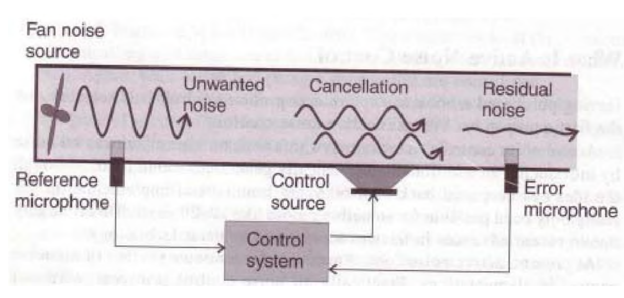

Figure 5. Schematic of Adaptive Feedforward Active Noise Control System [6].

Noise propagation through an automotive intake system can be considered synonymous to the propagation of a sound field through a duct. Here, a duct is simply considered to be an enclosure where one of the dimensions of the enclosure is very long. This enclosure, most often, terminates into open space where in the case of an automotive induction system, the termination would be the air intake opening.

One application of ANC in an automotive intake system (McLean paper 2001-01-1613) used a source coupling technique to control the automotive intake noise. This was accomplished by placing a conventional loudspeaker co-axially inside the air intake. Here, the speaker diaphragm was co-planar with the termination of the air intake duct. A sketch of the speaker system inside the intake duct is shown in Figure 6.

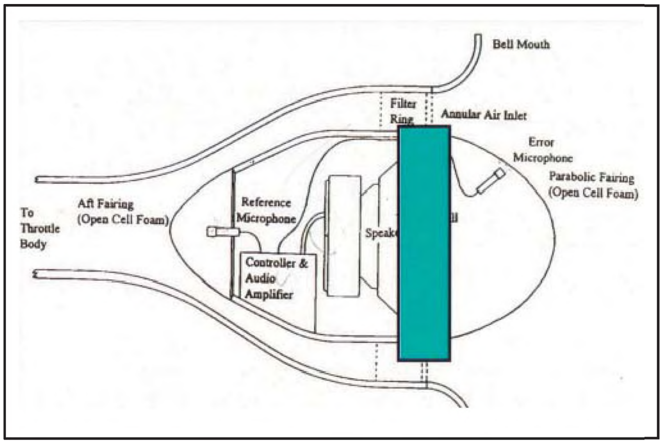

Figure 6. Sketch of Air Induction ANC via Source Coupling [7]

It was found that placing the speaker in the co-axial configuration provided much more attenuation when compared to configurations where the speaker was aligned with the pipe axis, with the speaker pointed into the pipe inlet or where the speaker and pipe axes were orthogonal.

4. THEORETICAL MODEL

The results of this work are the culmination of a theoretical modeling investigation and hence, a model of the original unmodified engine first had to be created.

4.1 Modelling Software

The modelling software program used for this investigation is called Ricardo WAVE. "WAVE is a computer-aided engineering code developed by Ricardo to analyze the dynamics of pressure waves, mass flows and energy losses in ducts plenums and the intake and exhaust manifolds of various systems" [8]. This is accomplished by applying a one-dimensional finite difference approach of the theoretical thermo-fluid equations of the working fluids of the defined system. First, a representation of the subject components must first be synthesized to a rendering of the subject engine. In order to facilitate this, specific information about the engine must be obtained. This information is comprised of three categories: i) geometric data; ii) engine data; and iii) the operating conditions.

The geometric data required to completely model the engine includes the physical dimensions of the intake and exhaust manifold duct lengths, port sizes and air box volumes. These lengths and volumes play an important role in the determining the performance characteristic of the engine. Given this, the geometric dimensions used in the model must be as accurate as possible to ensure the most accurate numerical results. Even the component materials and surface finishes will influence the effects of wall friction. Without these considerations, actual induced flow losses may not be realized in the modeled results.

The engine data is the quantitative information associated with the engine block and cylinder head. This would include valve diameters, timing and lift profiles, and complete port flow coefficients. The bore, stroke, connecting rod length and pin offset, along with the compression ratio, firing order and frictional details are required information of the engine block.

Information of the operating conditions of the engine is also required for the simulation model to reach steady state condition. The better the initial operating information is, the more capable the simulation will be able to quickly and accurately reach its final results. Some of the operating conditions required are the inlet and exhaust wall temperature, operating speed, head, piston and cylinder temperatures. Further requirements include the ambient conditions and combustion information.

4.2 Model Design

In order to analyze an engine with WAVE, it must first be created with the preprocessor WAVEBUILD. This canvas provides the ability to create and synthesize all of the building blocks representing the various ducts, volumes and other engine component. WAVEBUILD also allows for the input of the required physical data and operating conditions of the engine.

Figure 7 provides an illustration of the unmodified model of the engine used in this investigation. By unmodified,

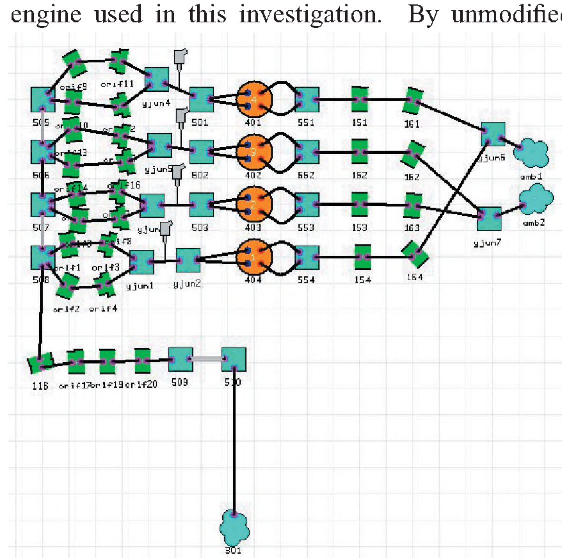

Figure 7. Unmodified WAVEBUILD Engine model.

it is meant that this engine is the original design prior to the implementation of any manifold bridge. The engine modeled is based on a Toyota 4A-GE used in the North American MR2 Mark I and Corolla GTS applications. The engine configuration is a 16 valve inline 4 cylinders with a displacement of 1587 cc and a compression ratio of 10:1. This model provided the reference acoustical performance information of the motored engine that was used to evaluate the results of the implementation of the manifold bridge.

Figure 8 is a schematic of the front end of the intake system illustrated with a corresponding WAVE model for the shown components. The inlet snorkel, which is opened to the ambient conditions, is attached to the airbox where the air filter is housed. The airbox is modeled as two cylindrical volumes joined at the air filter element which is represented by the hatched line in the figure. The air filter is modeled as a zero length duct, with a perforated obstruction. The zip tube exits the airbox and is connected to the throttle body. The diameter of the throttle is set as a variable and can be adjusted to represent different engine loading conditions.

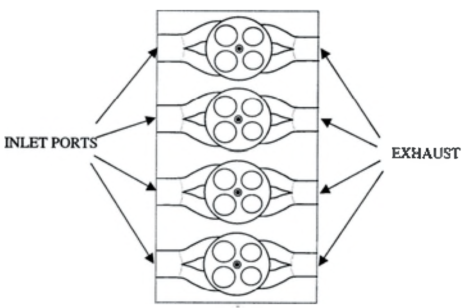

Figure 8. Sub-system of Intake Front End [9].

Figure 9 is a representation of a simplified unmodified intake manifold. The actual intake manifold used in this study is more complex in that each cylinder has duel runners, however, this representation provides a good understanding of the modelling procedure used. This simplified manifold consists of a plenum with four runners. The inlet to the plenum is the throttle body discussed above. The runners are modeled by several individual ducts so as to accurately represent the changes in cross sectional area and bends.

The next sub-section of the engine model created

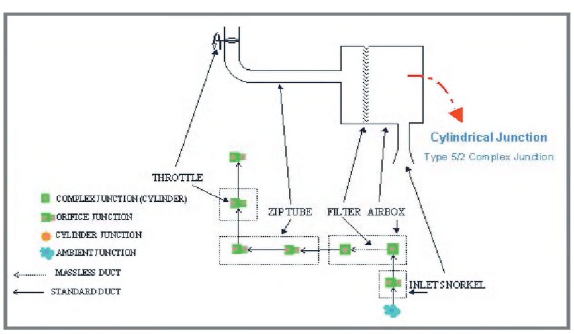

Figure 9. Intake Manifold System [9].

is the cylinder head as represented in Figure 10. As already stated, the engine is a 16 valve, four-cylinder engine with four valves per cylinder. The diameters and lift information of the intake and exhaust valves is input into the model. All losses related to the ports are taken into account through the specification of flow coefficients.

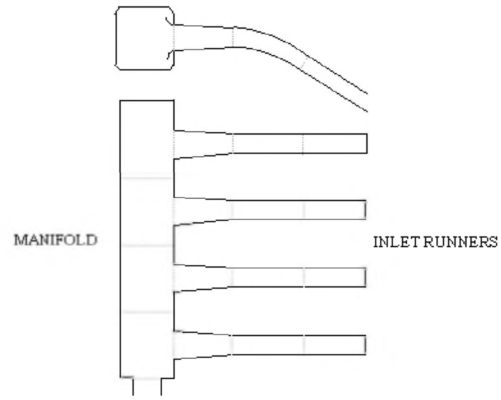

Figure 10. 16 Valve Cylinder Head [9].

The configuration of the exhaust manifold for this engine is a four to two arrangement. In other words, there are four exhaust runners which join to become two runners which subsequently meet to become a single outlet. A more simplified four to one is shown in Figure 11. Care must be taken in the modelling process to ensure that the angles and dimensions of the ducts are carefully represented in the model as they play an essential role in determining the dynamic behaviour of the exhaust system.

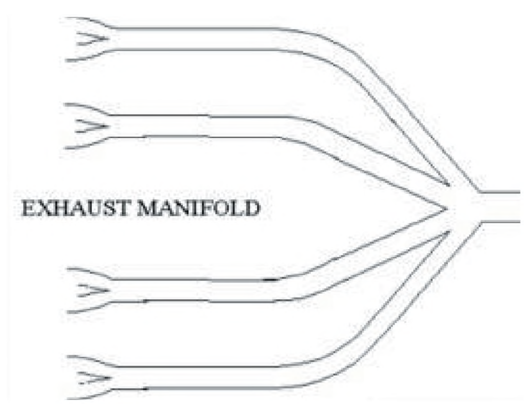

Figure 11. Exhaust Manifold System [9].

4.3 Model Outputs

The purpose of this work is to validate the feasibility of using a manifold bridge to improve the acoustical performance of an automotive intake system. The success of this investigation is determined by both the realized attenuation due to the implementation of the bridge as well as any improvement in the measured sound quality. This is accomplished through application of several psychoacoustic metrics.

The traditional acoustical parameters reported in this paper to measure the attenuation are all measured at a position 0.1 meters from the intake opening of the engine. This is an industry standard. Here, the overall linear and A-weighted sound pressure level and frequency spectra are measured for various steady rpm's of the engine's operating range. Colour maps are also determined for transient runs over the rpm range of the engine.

In addition to the traditional acoustical parameter, several psychoacoustic metrics were employed to measure the effectiveness of the manifold bridge. The metrics used included loudness, sharpness, fluctuation strength and roughness and are described below.

Zwicker Loudness is a standardized metric that describes the human perception of loudness instead of simply a reported sound pressure level. This value takes into account the temporal processing of sounds as well as audiological masking effects [10]. The unit of loudness is sones and is given across Bark, or critical, bands, as opposed to frequency bands.

Sharpness, which has units of acum, describes the high frequency annoyance of noise by applying a weighting factor on sounds above 2 kHz. This overall measurement is useful for such sounds as broadband sources, wind or rushing air noise and gear meshing or grinding sounds. Given that a high frequency component of intake noise is created by the intake air traveling across the valve seat at a high velocity, sharpness is an appropriate metric for the evaluation of the merits of the manifold bridge.

Fluctuation strength and roughness are both metrics used to describe the annoyance of modulating sounds depending on the frequency of the modulation. The fluctuation strength focuses on sounds which modulate at frequencies between 0.5 Hz and 20 Hz, with 4 Hz being the most annoying fluctuation. The unit of amplitude for fluctuation strength is the vacil. Roughness focuses on noise which is modulating at frequencies between 20 Hz and 300 Hz, with the most annoying modulation being 70 Hz. The unit of amplitude for roughness is the asper. When sounds modulate faster than 300 Hz, the human ear is not be able to distinguish this from a normal pure tone. Examples of modulating sources include beating sounds, sirens and fan blades.

4.4 Optimization of the Model

In order to achieve both the greatest attenuation and improvement in sound quality as a result of the insertion of the proposed manifold bridge, the physical parameters of the bridge needed to be determined. In other words, the configuration of the bridging runners as well as their lengths and diameters needed to be calculated.

Several bridging configurations were investigated and evaluated with respect to overall noise attenuation. The first included a single bridge from the exhaust manifold output to the intake manifold plenum. Secondly, bridging ducts running from exhaust manifold runners to their corresponding intake manifold runners were looked at. In other words, a bridging duct attached to the exhaust runner associated with cylinder number one would be similarly attached to the intake runner, also for cylinder number one. The third configuration looked at the outcome of attaching the bridging runners to each of

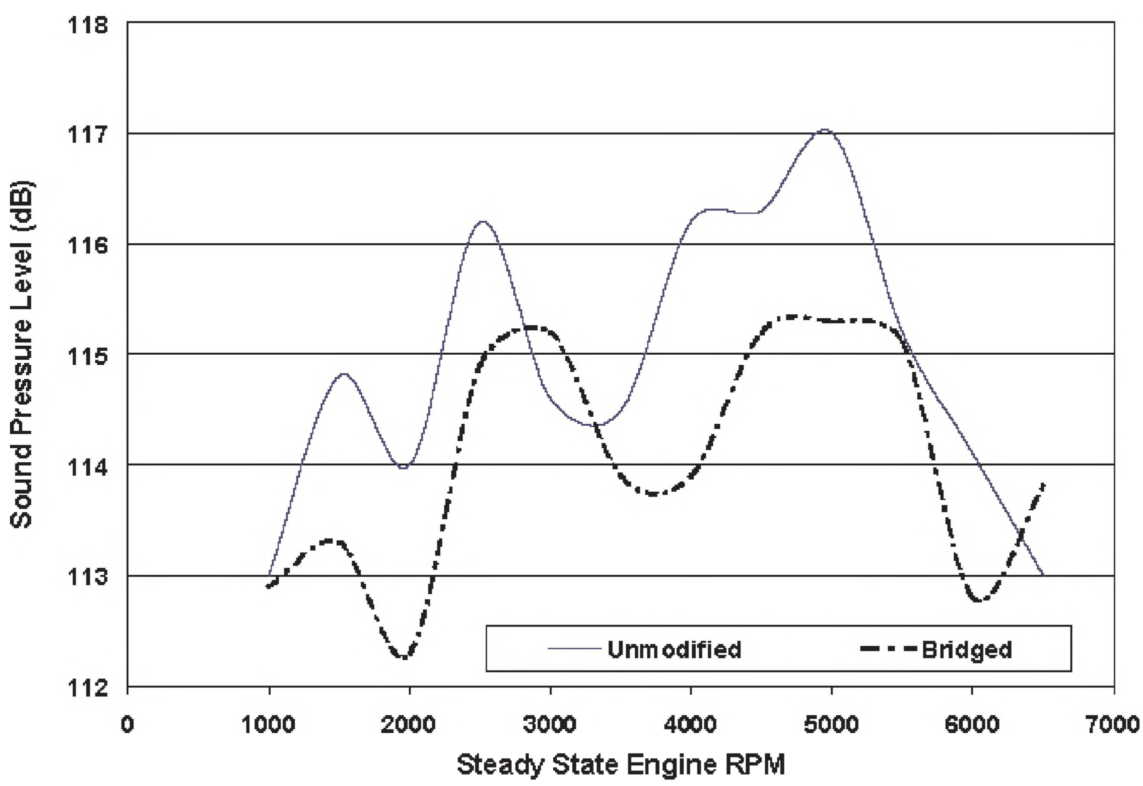

Figure 12. Steady State Intake Noise of Unmodified and Bridged Engines.

the four exhaust manifold runners and routing them to the corresponding intake runners which were associated with cylinders that were 180 degrees out of phase with respect to the firing order of the engine.

Using the results from the three alternatives described above, it was determined that the second approach provided the best attenuation at the measurement position outside the air induction inlet. That is, the configuration where the bridging ducts were linked from the exhaust manifold runners to their corresponding intake manifold runners achieved the greatest noise reduction. Using this configuration, the physical parameters were further optimized. The length of the duct was varied from the minimum physical length possible to a maximum of approximately double. While it was found that additional length for some runners proved to have additional attenuating characteristics, the investigation was limited to double the original length for practical purposes. The diameter of the bridging ducts were also varied from 18 to 32 mm in order to find an optimal cross sectional area. This range was again restricted to practical sizes.

5. DISCUSSION OF RESULTS

Once the feasibility range of the physical characteristics was determined, more detailed acoustic analyses were carried out. Specifically, the noise attenuation between the original and bridged engines was determined for steady state conditions for engine speeds from 1000 to 6500 rpm. Similar analyses were performed for transient runs between the same rpm range. Sound quality analyses were also carried out on the steady state and transient cases. Figure 12 is an illustration of the modeled sound pressure levels for the steady state conditions for engine speeds ranging from 1000 to 6500 rpm. The two shown curves represent the sound pressure levels for the case of the unmodified engine along with the case of engine modified with the manifold bridge. It can be seen that for the entire operating range of the engine, with the exception of between approximately 2800 to 3400 rpm, the bridged engine is quieter then the original unmodified engine. It was found that an attenuation of up to approximately 2.5 dB at around 4000 rpm was realized with the implementation of the bridge over the unmodified engine. Only at a speed of approximately 3000 rpm was the bridge found detrimental to the acoustic performance of the engine with an increase in sound level of approximately 0.8 dB.

Figures 13 and 14 further emphasize the apparent attenuation for steady state conditions between the original and modified engines. These figures show the frequency spectra of the two engines at engine speeds of 2000 rpm and 4000 rpm respectively. Again, it can be seen that the curves representing the bridged engine demonstrate lower amplitudes for much of the frequency spectrum.

In addition to the steady state simulations, acoustic tests were also carried out on transient simulation runs of the two engine models. Like the steady state simulations, the transient runs were from 1000 to 6500 rpm. It was found that an overall sound reduction of 5.6 dB was achieved with the implementation of the manifold bridge. Figures 15 and 16 are colour map representations of the induction noise during these transient simulations. Figure 15 shows the frequency of the intake noise for the rpm range of the unmodified engine. Here, the various colours represent the amplitude of the predicted sound pressure level.

Similarly, Figure 16 illustrates the same for the engine modified with the manifold bridge. The acoustic shortcomings of the unmodified engine are incontrovertibly obvious. The yellow and orange streaks representing the fundamental and subsequent harmonic frequencies are more apparent with more red showing on the map of the unmodified engine. This shows higher amplitudes of sound at the fundamental frequencies which are obviously associated with the speed of the engine. Also, the bridged engine simulation has less of the higher sound pressure level represented by the green colour. Similarly, it has more of the lower sound pressure represented by the mid and dark blue shades.

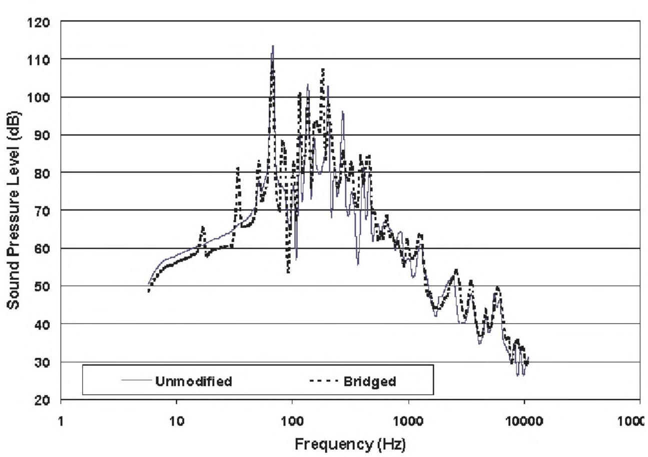

Figure 13. FFT of Both Modeled Engines at 2000 rpm.

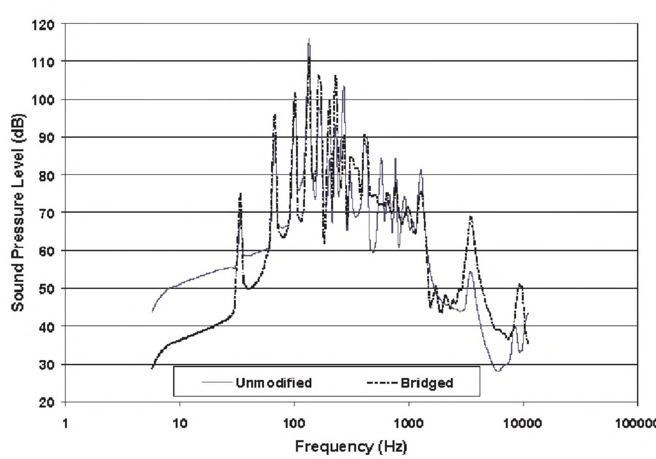

Figure 14. FFT of Both Modeled Engines at 4000 rpm.

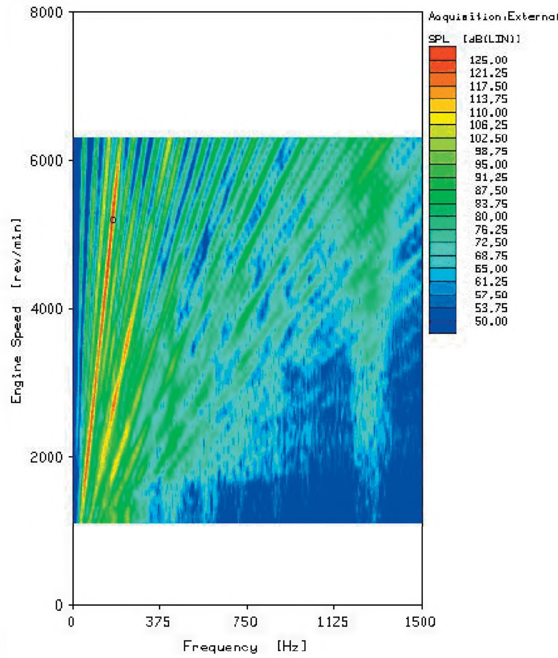

Figure 15. Colour Map of Intake Noise of Unmodified Engine.

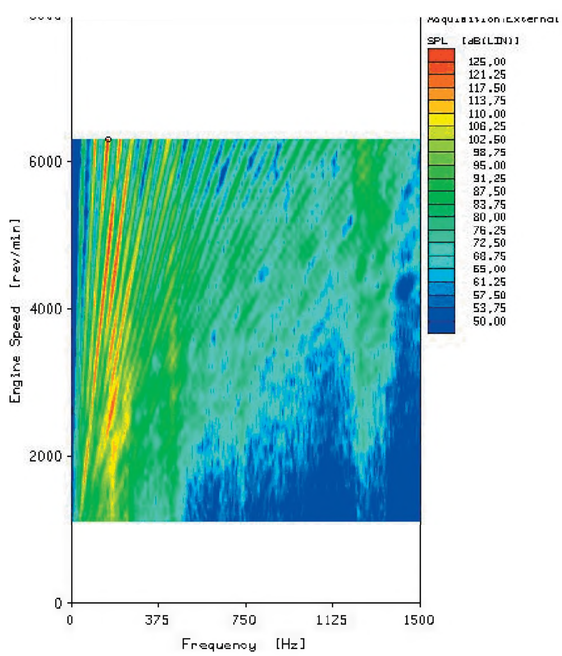

Figure 16. Colour Map of Intake Noise of Bridged Engine.

In addition to the traditional methods of acoustic analysis, psychoacoustic metrics were applied to quantify the differences in sound quality of the two engines. The results of this sound quality analysis are illustrated in Table 1.

Sound Quality Results		
	Unmodified	Bridged
Loudness (sones)	155.07	120.30
Sharpness (acum)	0.53	0.65
Fluctuation Strength (vacil)	1.80	1.44
Roughness (asper)	2.51	3.49

Table 1. Sound Quality Comparison of Unmodified and Bridged Engine.

Perhaps the largest difference between the two engines is shown by the loudness reduction of approximately 35 sones. While this metric is spectrally influenced in that it loosely follows the attenuation realized by the A-weighting curve over the frequency range of interest, one advantage of the overall loudness is that it also includes the influence of masking and temporal effects. Given this, the advantages of the bridged engine are more apparent.

An increase in sharpness is shown with the bridged engine which indicates that an increase in the frequency content of the signal is present in the range above 2000 Hz.

The decrease in fluctuation strength from 1.8 to 1.44 vacil shows that a drop in low frequency modulation is present. This concept is reinforced by the fact that there is an increase in roughness with the bridge engine results. This suggests that the loss of low frequency modulation was just simply a shift to a higher frequency, somewhere above 20 Hz.

6. CONCLUSIONS

For the conditions investigated, it has been shown that the implementation of the manifold bridge has a positive influence on both the amplitude and the sound quality of induction noise. While this investigation was limited to the presentation of the theoretical modelling results, experimental verification is also being pursued. The focus of this presentation was the realized acoustical results of the bridge implementation and did not report on some of the other engine performance criteria. It should be realized that the addition of the manifold bridge will affect some of these other criteria, particularly the influence of the additional exhaust gas recirculation. However, if this was found to be detrimental to the engine performance, the exhaust and intake systems could be isolated from each other through the addition of a membrane or a dual walled bladder system. This investigation, however, does demonstrate the merits in pursuing further refinements of this unique noise control approach.

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