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Sound Intensity and Acoustic Source Quantification to Identify the Noise Contributions of Gasoline Direct Injection Components

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Summary
In this paper, the sound intensity and airborne source quantification methods were used to estimate the airborne noise radiating from a gasoline direction injection engine and the relative contributions of each of the engine's components to such airborne noise. In order to estimate noise contribution using airborne source quantification, the sound volume velocity for the source should be estimated. In previous methods, the sound volume velocity was estimated using the inverse method, whereas in this paper, the sound volume velocity is estimated directly. For this work, the sources of noise radiating from thegasoline direction injection engine were first identified by measuring the sound intensity of the noise sources while the engine was running idle. The sound volume velocities of these sources were estimated from the product of the area of that noise source and the directly measured particle velocity. In order to validate this method, two speakers with different frequencies were installed in an anechoic chamber and the contribution of each source was estimated. Finally, this proposed method was applied to the identification and contribution of noise sources in a gasoline direction injection engine.

PACS no. 43.58.-e

1. Introduction

Recently, the development of fuel-efficient, high-powered and environmentally friendly engines has gained prominence. In particular, efficient performance and environmental impact have come under strict legal regulations, rather than being an optional enhancement, and the importance of the engine's environmental impact has steadily grown. As a result, GDI (gasoline direct injection) engines have emerged as serious contenders, having significant advantages in fuel economy, output power and environmental impact,however, GDI engines generate increased noise due to components such as high-pressure fuel pumps, fuel injectors and fuel rails that are integral to the engine itself. Such noise has increasingly become an issue [1],therefore it is important to identify the noise sources and relative contributions for the refinement of the GDI engine.

In this paper we used the sound intensity method and the ASQ (acoustic source quantification) technique to identify noise sources and their relative contributions among GDI engine components. The Sound intensity methodhas been used for sound source identification, visualization of acoustic energy flow, and measurement of sound penetration and loss. This methodhas the advantage of being able to ignore background noise, and is therefore used frequently to identify noise sources, both in the lab as well as in the field. The sound intensity at a given point is expressed as a complex quantity obtained from the product of the sound pressure and the particle velocity [2, 3, 4, 5]. The ASQ technique is used to quantify airborne noise sources and determines the contribution level of each noise transfer path by considering the transfer path between each noise source and the response at a target point [6, 7]. Generally, when ASQ is applied, an inverse matrix method is used to obtain the sound volume velocity of the airborne noise sources [8, 9],however, there isthe inherent error in the inverse matrix process of the transfer function. In order to avoid thiserror, in this paper the sound volume velocity of noise sources is directly calculated from the product of the particle velocity and the area of the noise sources (obtained during the measurement of sound intensity). Thissound volume velocity is then used with ASQ technology to estimatethe sound pressure of the response at a target point. To identify the noise sources of a GDI engine equipped vehicle while idling, we first used a1/2 inch PU-probe (Microflown Technologies Company) to map the sound intensity of the noise sources. Afterwards, the particle velocity was measured simultaneously with the sound intensity. Noise sources were identified using the measured sound intensities, and the sound volume velocity of each source was obtained from the product of the area of that noise source and the measured particle velocity. This

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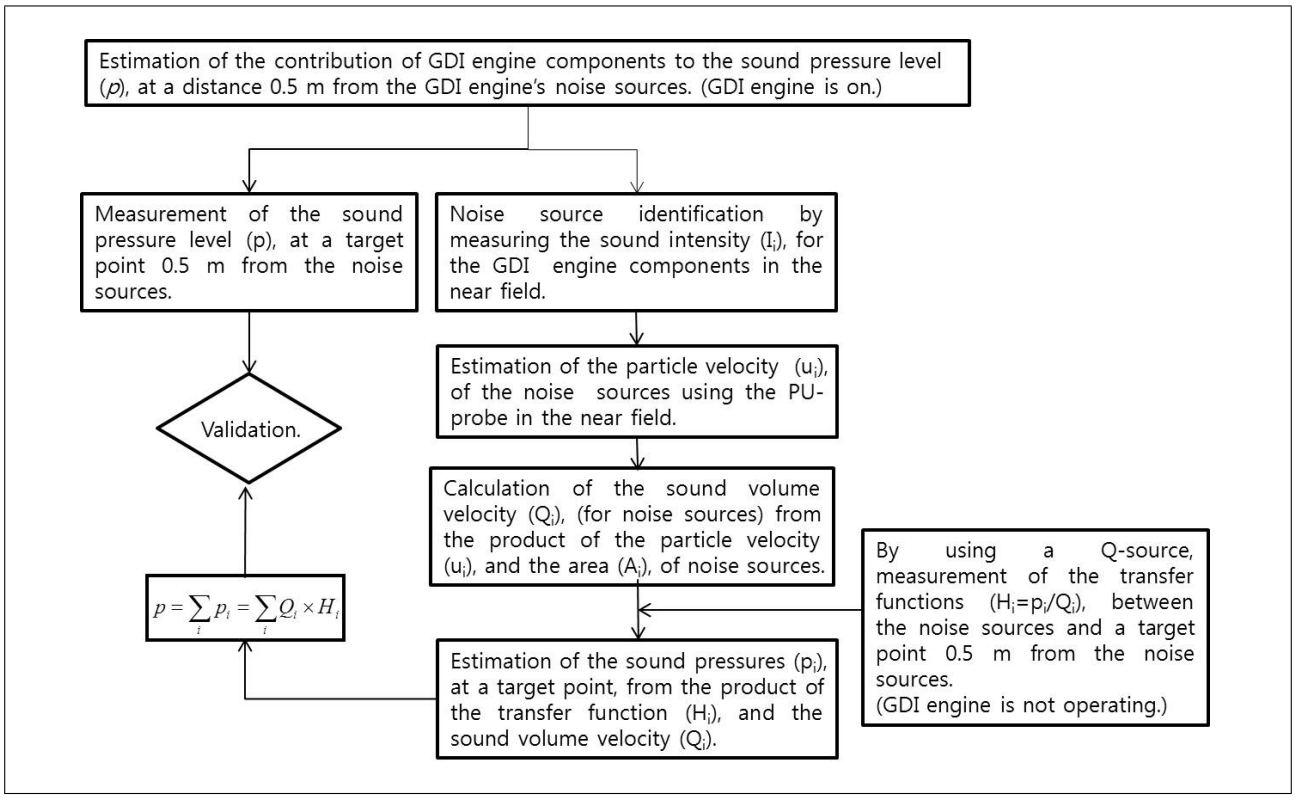

Figure 1. Block diagram of the method for identification and determination of relative contributions of noisefrom GDI engine components.

sound volume velocity was applied to ASQ and the noise contribution levels of the GDI engine components were analyzed. A summary of this method is given in Figure 1. ASQ estimates the SPL (sound pressure level) generated by each noise source, which in this case are GDI engine components. In order to apply ASQ, the transfer functions between a target point and the noise sources are required. In this paper we obtained the transfer functions from measuring the SPL at a target point 0.5 m from the GDI engine components. The SPL at the target point is generated by an acoustic source with known sound volume velocity whilst the GDI engine is idle. The transfer functions between the target point and the GDI engine sources are calculated by dividing the SPLby the sound volume velocity. Finally the SPL at the target point, generated by each noise source whilst the GDI engine is operational is obtained from the product of the transfer function (due to each source) and the sound volume velocity (of each source). The SPL estimated throughout this method is the contribution of each noise source to the SPL measured at the target point. In order to validate this method we performed a validation test in an anechoic chamber using two loudspeakers, as presented in section 3.

2. Theory of sound intensity and ASQ

2.1. Sound intensity

The sound intensity is expressed as vectors that indicate the magnitude and direction of the sound energy propagating from a given point. The sound intensity vector $\boldsymbol{I}$, in a specific direction n, is equal to the energy passing through a unit area of a plane perpendicular to the direction of transfer, as defined by equation (1), where E_n is acoustic energy of a noise source,

$$I = \frac{\mathrm{d}E_n}{\mathrm{d}t\,\mathrm{d}A} \quad [\mathrm{W/m^2}]. \tag{1}$$

The sound intensity vector $\boldsymbol{I}$ is the average of values derived from the product of the momentary sound pressure and the particle velocity at a specific position r, and is defined by equation (2) when the signal is stationary,

$$\bar{I}_r = \overline{p(r,t)\cdot\vec{u}(r,t)} \quad [\mathrm{W/m^2}]. \tag{2}$$

The cross correlation function between the sound pressure and the particle velocity is defined by

$$R_\infty(\tau) = \lim_{\tau\to\infty}\left(1/T\right)\int_0^T p(t)\,u(t+\tau)\,\mathrm{d}t \tag{3}$$

The average sound intensity in direction n can be expressed as

$$\bar{I}_r = \lim_{\tau\to\infty}\left(1/T\right)\int_0^T p(t)\,u_r(t)\,\mathrm{d}t \tag{4}$$

In order to convert the product of the sound pressure p, and the particle velocity u, into frequency ranges, Fourier transforms must be applied to the cross correlation function. The sound intensity of the cross power spectrum

$S_{pu}(\omega)$, is given by

$$S_{\infty}(\omega) = \frac{1}{2\pi}\int_{-\infty}^{\infty} R_{\infty}(\tau)\mathrm{e}^{-\mathrm{i}\omega t}\,\mathrm{d}\tau \quad [\mathrm{W/Hz}]. \tag{5}$$

This function takes the form of complex quantities that indicate the average phase relation of the pressure p, and the particle velocity u. $R_{pu}(\tau)$ and $S_{pu}(\omega)$ are a pair of Fourier transforms, as

$$\begin{aligned} R_{pu}(\tau) &= \int_{-\infty}^{\infty} S_{pu}(\omega)\mathrm{e}^{\mathrm{i}\omega t}\,\mathrm{d}\omega, \\ \bar{I}_r &= \int_{-\infty}^{\infty} S_{pu}(\omega)\,\mathrm{d}\omega. \end{aligned} \tag{6}$$

Based on this treatment, $S_{pu}(\omega)$ represents the sound intensity in the frequency domain, therefore, the cross power spectrum can be characterized by

$$\begin{aligned} \Re e\{S_{pu}(\omega)\} &= \Re e\{S_{pu}(-\omega)\}, \\ \Im m\{S_{pu}(\omega)\} &= -\Im m\{S_{pu}(-\omega)\}. \end{aligned} \tag{7}$$

In equation (7), $S_{pu}(\omega)$ can have double sided frequencies, but from a physical viewpoint it is more convenient to check using a single side frequency only, so this equation can be rephrased as equation (8), where $G(\omega)$ is the power spectrum in the positive side of the frequency domain,

$$\begin{aligned} G_{pu}(\omega) &= 2S_{pu}(\omega) \quad \omega > 0, \\ G_{pu}(\omega) &= S_{pu}(\omega) \quad \omega = 0, \\ G_{pu}(\omega) &= 0 \quad \omega < 0. \end{aligned} \tag{8}$$

Now the value of the sound intensity by component can be defined by

$$\begin{aligned} \bar{I}_r(\omega) &= S_{pu}(\omega) + S_{pu}(-\omega) \\ &= 2\Re e\{S_{pu}(\omega)\} = \Re e\{G_{pu}(\omega)\}. \end{aligned} \tag{9}$$

Using the above relations the sound intensity can be obtained.

2.2. Airborne source quantification

Assuming that the noise path is a linear system, the noise transfer path model for a GDI engine to predict the SPL at a distance of 0.5 m from the front of engine can be represented, as shown in Figure 2. Through each transfer path (expressed by their respective transfer function corresponding to each noise source), the responses at the target points are revealed. To verify the transfer characteristics from each noise source to the response point, ASQ technologywas used. By taking the product of the transfer function and the sound volume velocity from each noise source, the noise contribution level of each component can be calculated [9]. The model expressed in Figure 2 can be reiterated as the equations shown in equation (10), where Q^{oper}_{pump}, $Q^{oper}_{\mathrm{injector}}$, and $Q^{oper}_{\mathrm{engine}}$ are the sound volume velocities measured at each noise source.

$$P^{oper}_{\mathrm{pump}}(\omega) = H_{\mathrm{pump}}(\omega)\cdot Q^{oper}_{\mathrm{pump}}(\omega), \tag{10a}$$

$$P^{oper}_{\mathrm{injector}}(\omega) = H_{\mathrm{injector}}(\omega)\cdot Q^{oper}_{\mathrm{injector}}(\omega), \tag{10b}$$

$$P^{oper}_{\mathrm{engine}}(\omega) = H_{\mathrm{engine}}(\omega)\cdot Q^{oper}_{\mathrm{engine}}(\omega). \tag{10c}$$

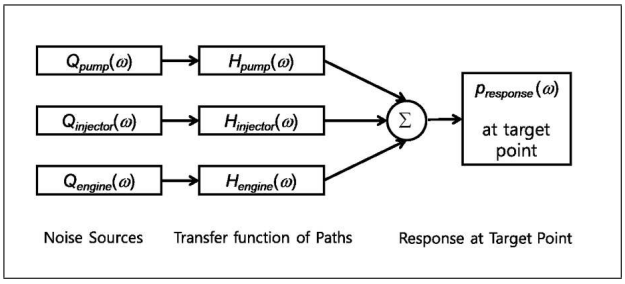

Figure 2. Modeling of the noise transfer path of a linear system for application of ASQ.

This sound volume velocity can be calculated when measuring the sound intensity. $H_{\mathrm{pump}}(\omega)$, $H_{\mathrm{injector}}(\omega)$ and $H_{\mathrm{engine}}(\omega)$ refer to the frequency response functions from each noise source to a response point and can be measured using a LMS Q-source and a B&K prepolarized Free-field 1/2" microphone (type 4189) whilst the engine is off. The $P^{oper}_{\mathrm{pump}}(\omega)$, $P^{oper}_{\mathrm{injector}}(\omega)$ and $P^{oper}_{\mathrm{engine}}(\omega)$ terms refer to the sound pressure value contributionsfrom each component of GDI engine at the target point. These sound pressures are obtained from the product of the sound volume velocity and the frequency response function. By taking the sum of the sound pressure values for each component, the total sound pressure at the target point is estimated as

$$\begin{aligned} P^{oper}(\omega) = \sum \Big[& P^{oper}_{\mathrm{pump}}(\omega) + P^{oper}_{\mathrm{injector}}(\omega) \\ & + P^{oper}_{\mathrm{engine}}(\omega) + \ldots \Big] \end{aligned} \tag{11}$$

If the sound pressure measured using a microphone at the actual target point exhibits a similar trend to the estimate obtained from equation (11), the proposed method can be used for the identification of the noise transfer path and the relative contribution of each noise source. The advantage of this method is that it can avoid the error occurred by the inverse method [9].

3. Validation of the proposed method

3.1. Validation experiments in the laboratory

In order to verify the validity of the proposed method prior to real vehicle testing, a validation experiment was conducted. This experiment used two speakers under laboratory conditions. Assuming that the speakers were noise sources of a GDI engine, they were installed in an anechoic chamber, as shown in Figure 3. In order to allow for sound intensity mapping and analysis of the relative noise contribution level of each component, the speakers were positioned in a 3 × 6 grid. The speaker on the left (as shown in Figure 3) vibrated with a 400 Hz sinusoidal signal, and the speaker on the right vibrated with a 900 Hz sinusoidal signal. When the speakers were activated, the sound intensity $\bar{I}_r$ and sound volume velocity u, was measured for each point of the grid using the PU-Probe. The sound intensity measurement used the ISO 9614-1 method, which takes individual measurements for



Figure 3. Area segments for the sound intensity mapping of the noise radiating from two loud speakers installed in an anechoic chamber.

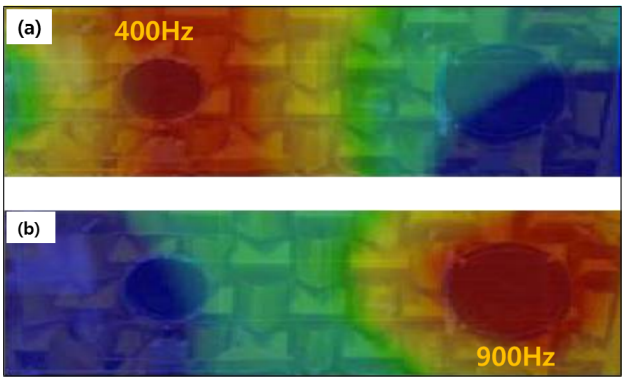

Figure 4. Sound intensity maps for the sound radiating from the two loud speakers. (a) Center frequency = 315 Hz (frequency band = 353 Hz ∼ 400 Hz), (b) center frequency = 1000 Hz (frequency band = 880 Hz ∼ 1130 Hz).

each numbered grid, in contrast to the ISO 9614-2 method, which uses continuous scanning. Once the sound intensity measurements were completed, the speaker vibration was stopped and the LMS Q-Source was set up at the target point of 0.5 m in front of the speakers (the lower frequency range of the LMS Q-source is 300 Hz). By installing microphones at each segment of the grid used for the measurement of the sound intensity $\bar{I}_r$, the frequency transfer function $H(\omega)$ between each point and the target point was obtained. The sound volume velocity $Q(\omega)$, of each segment was obtained by taking the product ofthe measured particle velocity $U(\omega)$, and the area A, of each segment. $U(\omega)$ is the Fourier transform of the particle velocity $u(t)$.

3.2. Analysis of the Sound Intensity

The sound intensity measured using the two-speaker-setup described above can be analyzed through mapping. Since noise analysis typically uses 1/3 octave bands, the sound intensity mapping also used 1/3 octave bands. Figure 4a shows the results of this analysis for the 1/3 octave band with a center frequency of 400 Hz and a range of 353 Hz-440 Hz; the sound intensity ranged from 75 dB to 98 dB. These results show that the sound propagated from the center of the left speaker that vibrated at 400 Hz. Figure 4b shows the results of this analysis for the 1/3 octave band with a center frequency of 1000 Hz and a range of 880 Hz–1130 Hz; the sound intensity ranged from 70 dB to 90 dB. The sound that propagated from the center of the right speaker vibrated at 900 Hz. These results confirm that the

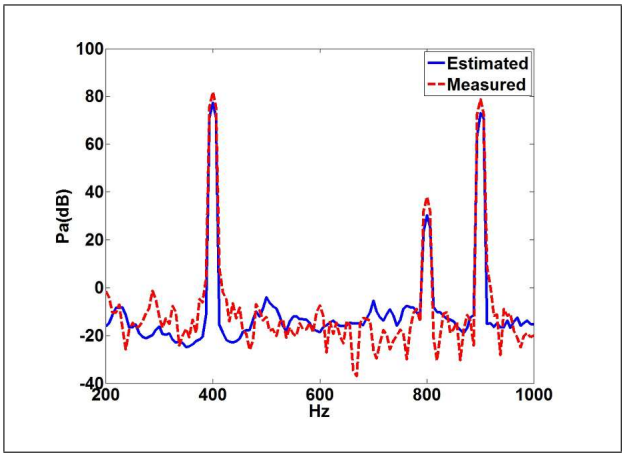

Figure 5. Comparison between the measured and estimated sound pressure levels.

noise source in the 400 Hz range is the left speaker and the source in the 1000 Hz range is the right speaker. This same method can be used to identify the noise source components in a GDI engine, as shown in section 4.

3.3. Estimating the Sound Pressure Levels of Two Noise Sources

In order to estimate the SPL at a target point 0.5 m in front of the speaker, the ASQ technique described in equation (10) was used. The estimated sound pressure radiating from each grid was calculated from the product of the frequency transfer function $H(\omega)$, and the sound volume velocity $Q(\omega)$, (associated with each segment of the grid). Then, by taking the sum of the SPL due to each grid, the estimated (total) SPL at the target point was calculated. In Figure 5 a graph comparing the measured SPL (with two speakers vibrating) and estimated SPL is shown. The difference between the estimated and measured SPL for frequencies below 300 Hz is due to the lower frequency limit of the LMS Source. At the 400 Hz, 800 Hz and 900 Hz ranges, the measured values and the estimated values exhibit similar trends and magnitudes. The SPL at 800 Hz is the harmonic component of the noise emitted by the speaker vibrating at 400 Hz. The results shown in Figure 5 demonstrate that the estimated values are actually higher for a certain frequency band of the background noise, even when its maximum estimated values are lower than those measured. This is a measurement error due to signal leakage caused by the Hanning window used for signal processing [10]. The estimated values were validated as reliable.

The noise contribution levels were analyzed by dividing the estimated values for each section. The results of Figure 6a show that at 400 Hz, the contribution level is highest at segment 11 (the center of the left speaker), whereas Figure 6b shows that the contribution level is the highest at segment 8 (the center of the right speaker) for 900 Hz. Based on these results, it is reasonable to assume that the proposed method of incorporating transfer functions to directly measured sound volume velocities can be used with-

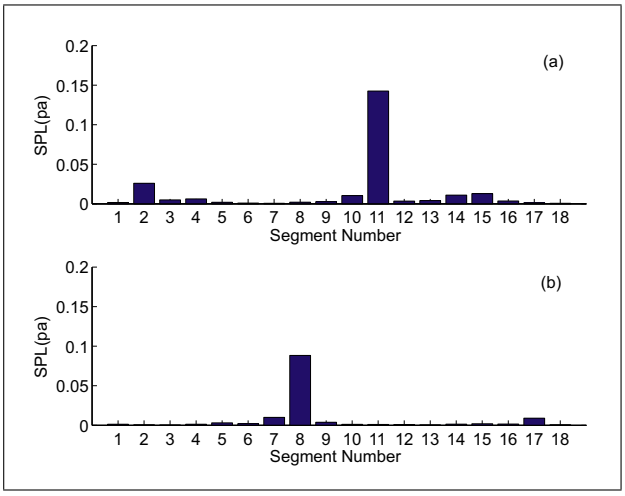

Figure 6. Noise contributions for each segment source, estimated using ASQ at frequencies of (a) 400 Hz, and (b) 900 Hz.

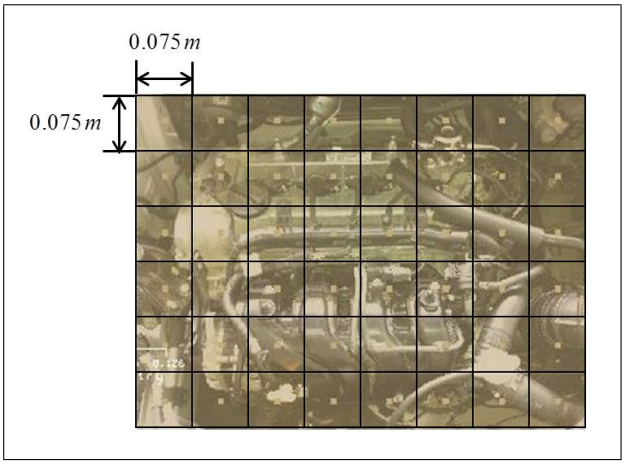

Figure 7. Segment area mapof the GDI test engine for the measurement of sound intensity.

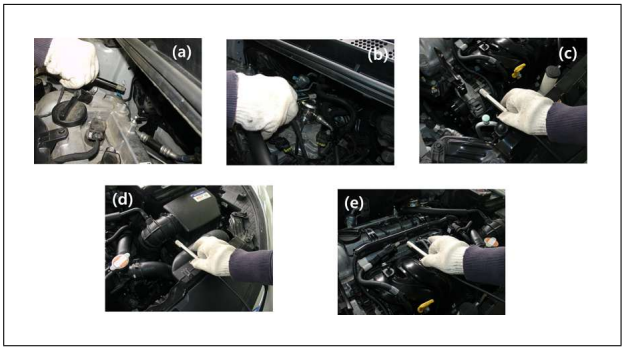

Figure 8. Photo illustration of the noise sources of the GDI test engine and the measurement procedure. (a) Ex-manifold, (b) Pump, (c) In-manifold (left side), (d) In-manifold (right side), (e) Injector.

out the inverse problem unlike the conventional practice of using the inverse matrix method on transfer functions. This resultconfirms the validity of this method for vehicle testing.

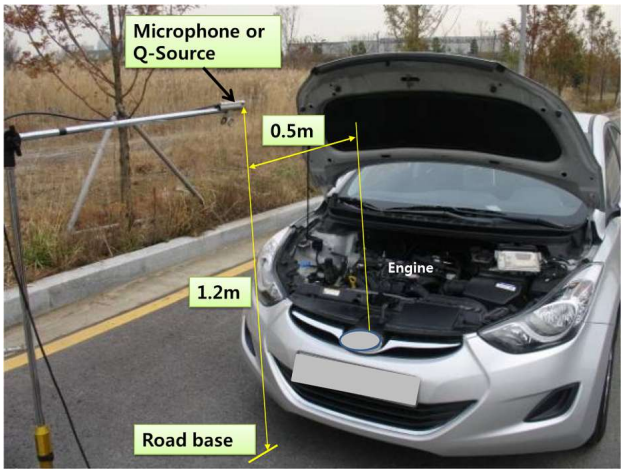

Figure 9. Position of the microphone or LMS Q-source for measurement of the sound pressure radiated from a GDI engine, or the transfer functions (between the engine noise sources and the sound pressure at the target point based on reciprocity theory) respectively.

4. Application of the proposed method to a GDI engine

4.1. Experimental Method

Now that the reliability of the method proposed here has been confirmed by a validation experiment, it can be to a vehicle equipped with a GDI engine. The experimental method was identical to that of the validation experiment and sound intensity measurements were taken with the engine running idle. As shown in Figure 7, the entire engine was set as the boundary, with the dimensions of the engine area set as 0.6 m×0.45 m with each grid segment set as 0.075 m×0.075 m. Sound intensity measurements used the discrete point method (ISO 9614-1), and data was collected at the 48 points representing each grid segment shown in Figure 7. In order to mitigate the cross-talk effect during measurements, the data was taken as close as possible to the measurement points using the PU Probe, as shown in Figure 8. Next, microphones were set up at the same points as used during sound intensity measurements, and the transfer functionsbetween each noise source and the target point were measured by installation of the LMS Q-Source 0.5 m in front of the vehicle. This process was carried out in an open (free field) environment in order to eliminate artifacts due to reflected sounds. The particle velocity of each grid segment was measured using the PU Probe. The sound volume velocity was obtained from the product of the particle velocity measured at each grid segment and the area of each grid segment. Finally, the sound pressure level was measured at 0.5 m in front of the vehicle, as shown in Figure 9. The measured SPL was used as a comparison for the estimated SPL obtained using equation (10).

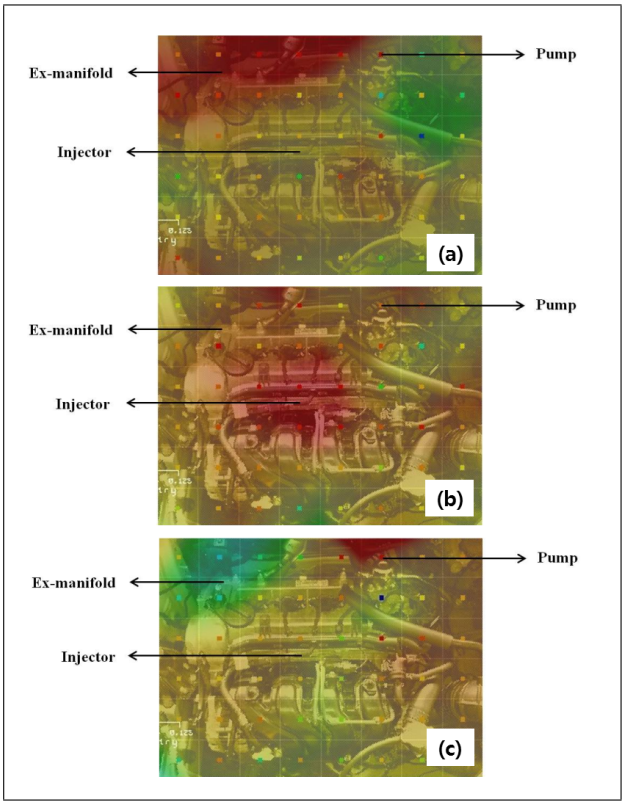

Figure 10. Mapping of the sound intensity for the noise sources of the GDI test engine at a 1/3 octave center frequency of: (a) 2500 Hz (frequency band = 2250 Hz ∼ 2825 Hz); (b) 3150 Hz (frequency band = 2825 Hz ∼ 3530 Hz); (c) 4000 Hz (frequency band = 3530 Hz ∼ 4400 Hz).

4.2. Analysis of the Sound Intensity

By measuring the sound intensity of each grid, the relative sound intensity of the GDI engine components was obtained. As with the validation experiment, the sound intensity was analyzed using 1/3 octave bands. By analyzing the sound intensity, it can be verified which components have the greatest SIL (sound intensity level) at different frequencies. When performing this sound intensity spectrum analysis, the highest SILs were found in the 2000 Hz–4500 Hz range, therefore, the 1/3 octave band analysis was carried out for only three center frequency bands at 2500 Hz, 3150 Hz and 4000 Hz. Figure 10a shows the image map for the sound intensity level in the 2250 Hz–2825 Hz frequency range where the center frequency was 2500 Hz. The SIL range is displayed from 20 dB to 50 dB. Overall, the SIL there was a high SIL around the exhaust manifold but also substantial impact around the injector. Figure 10b corresponds to the 3150 Hz–2825 Hz frequency range where the center frequency was 3150 Hz. The SIL ranged here from 25 dB to 53 dB, with the largest impact around the injector. This frequency band also shows a higher, overall,sound intensity compared to the other bands. As such, the noise generated in this frequency range has a large contribution to the overall noise level. Figure 10c corresponds to the 3530 Hz–4400 Hz range where the center frequency was 4000 Hz.

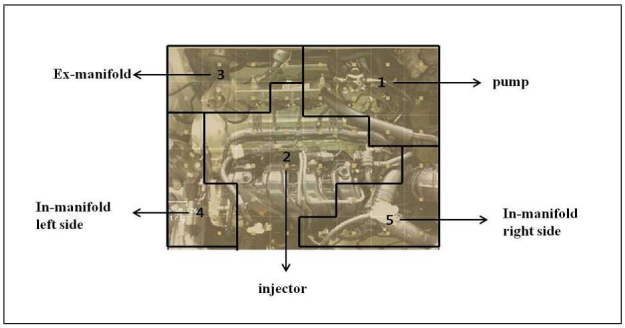

Figure 11. Segmented areas of the major noise sources in the GDI test engine.

Table I. Engine components corresponding to the number of grids they occupy.

Number	Component name	Number of grids
1	Pump	10
2	Injector	17
3	Engine 1	7
4	Engine 2	6
5	Engine 3	8

The SIL ranged from 10 dB to 40 dB and the effect around the pump is generally the largest.

4.2.1. Representative noise source components in a GDI engine

Through sound intensity analysis, the major contributions to noise from thecomponents of the GDI enginewere identified. Analysis of how the noise is transferred using sound intensity mapping, divided the main sources of noise sources in the engine. Figure 11 shows the engine compartment divided into five noise source sections, namely the pump, the injector and the engine [1, 2, 3]. Table 1lists the engine components corresponding to the number of grids they occupy.

Representative values were set for each of the noise source sections, and their transfer functions (between each noise source and the target point), and sound volume velocities, were obtained as outlinedin section 4.1. In order to estimate the sound pressures generated by each noise source, the ASQ technique is employed (equation 10). In short, by taking the product of thetransfer function and the sound volume velocity, the estimated sound pressure was obtained (for each noise source section).

4.2.2. Comparison of the measured and estimated values at the response point

The measured and estimated values of the SPL at response points must show similar values and trends, as demonstrated in Figure 12. The difference between the estimated SPL and the measured SPL for frequencies less than 300 Hz is due to the lower frequency limit of the LMS Source. The estimated values shown in Figure 12 were derived by taking the sum of all the estimated sound pressures from each noise source. As the sound pressure

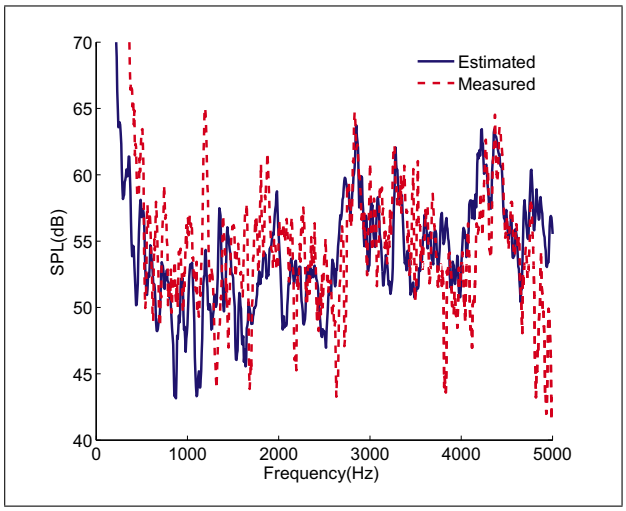

Figure 12. Comparison between the measured and estimated sound pressure levels radiating from the GDI test engine.

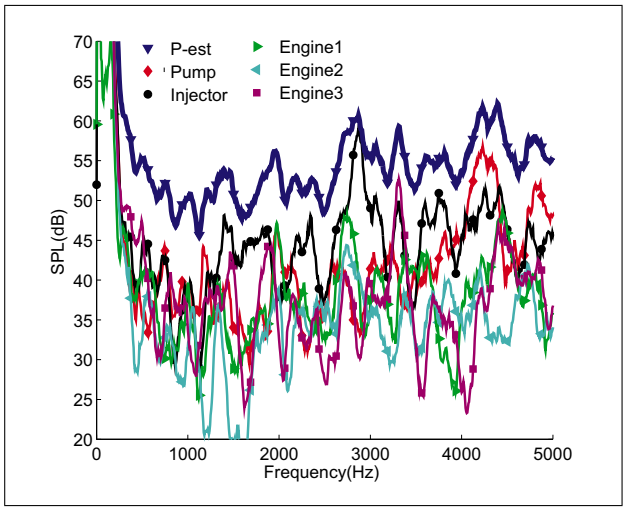

Figure 13. Contribution of the major noise sources in the GDI test engine whilst running idle.

values were minimal for frequencies under 2000 Hz and above 4500 Hz, only the 2000 Hz–4500 Hz range was analyzed in detail. According to these results, the estimated and measured sound pressure at response points exhibited similar behavior and the test to analyze noise contribution levelsfrom a source (using estimated values),is reliable.

4.2.3. Analysis of the contribution level by noise source

With the reliability of the noise contribution analysis confirmed above, the noise contribution levels based on the estimated sound pressures were analyzed. Figure 13 shows the contribution of each noise source in a spectrum format; overall the injector contribution appears to be the largest, except at 4200 Hz where the pump's contribution is larger. Since the sound pressure values from the injector and the pump are both high in the frequency range with this peak value, it can be concluded that the injector and the pump both had a large contribution on the total noise level. To quantify and express the contribution levels by noise source, analysis was carried out using 1/3 octave bands. Figure 14a shows the results for the range around a center frequency of 2500 Hz in 1/3 octave bands frequency analysis where the injector has the largest contribution. The injector had the largest impact in this frequency range in the sound intensity analysis as well. Figure 14b shows the results for the 3150 Hz range in 1/3 octave bands frequency analysis, where the injector's contribution is again the largest, as with the sound intensity analysis in this range as well. Figure 14c shows the results for the 4000 Hz range in 1/3 octave bands frequency analysis, where now the pump shows the biggest contribution. The sound intensity analysis also identified the pump's contribution as the largest in this range.

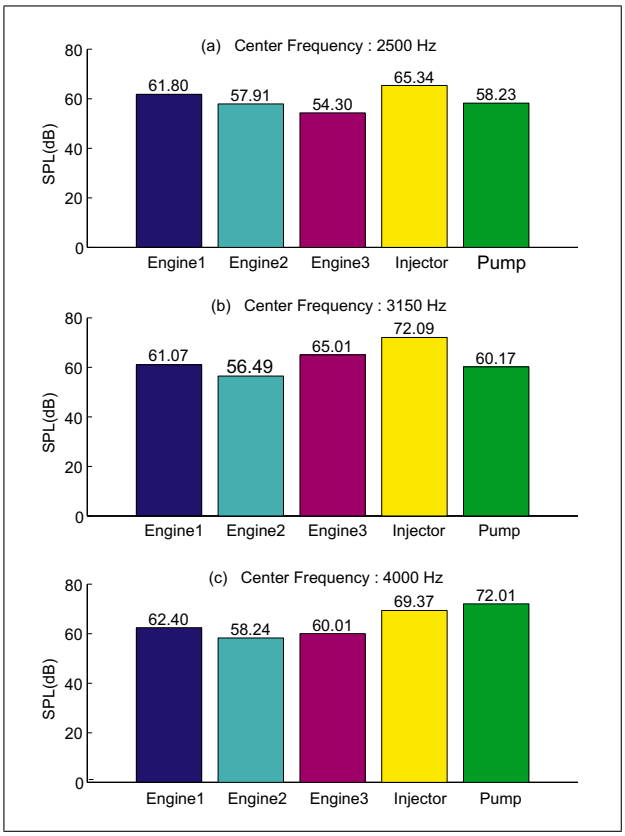

Figure 14. Contributions of the major noise sources at a 1/3 octave center frequency of (a) 2500 Hz (frequency band: 2250 Hz ∼ 2825 Hz), (b) 3150 Hz (frequency band: 2825 Hz ∼ 3530 Hz), (c) 4000 Hz (frequency band: 3530 Hz ∼ 4400 Hz).

The biggest contributing noise sources of the GDI engine were the injector and the pump. Any remaining noise (excluding the noise from the injector and the pump), was classed as "other noise", and the linear sum of noise from the three remaining segments (of the five, as shown in Figure 11), were named "engine noise". Figure 15 shows the results of analysis of the noise from the three sources in a spectrum format. The injector's contribution is prominent around 2900 Hz, whilst the engine's contribution is prominent around 2700 Hz and 3300 Hz. During sound intensity analysis, large effects were also evident around the

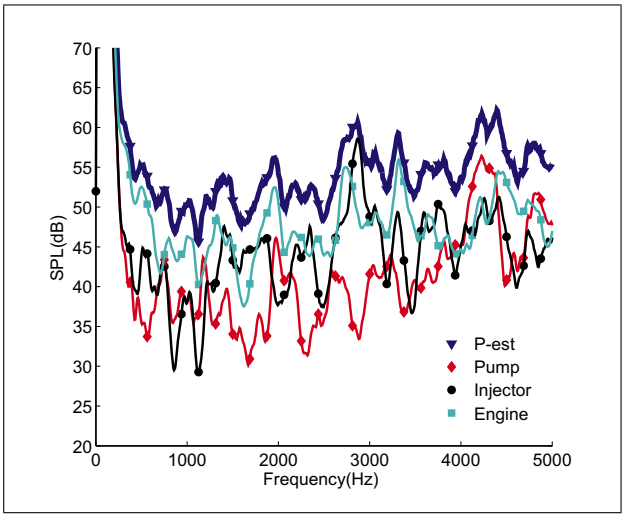

Figure 15. Contributions of the three major noise sources to the total sound pressure, atadistance of 0.5 m from the GDI test engine.

exhaust and intake manifolds. The pump's contribution is largest around 4200 Hz.

5. Conclusion

In this paper, the engine noise sources were mapped in a near sound field by measuring the sound intensities of an idling vehicle with a PU-Probe. The position of the noise sources were identified through this exercise and the engine was divided up according to those sources. The sound volume velocity of each source is obtained from the product of the particle velocity and the area of these noise sources. After measuring the transfer functions up to the normal point for the divided sections, the product of the sound volume velocity and transfer function was found, and the sound pressure for each component was estimated. Based on this method, the noise contribution of components within a given frequency range could be identified. Whilst the injector's contribution was dominant around 2900 Hz, at 4200 Hz the pump's contribution was larger. The engine's contribution was the largest around 2700 Hz and 3300 Hz. Based on these results, the noise contribution level for each component of a GDI engine can be estimated during its design, which will be useful in the reduction of noise from specific noise sources in future GDI engine designs.

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