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Numerical and Experimental Investigation of the Acoustic and Flow Performance of Intake Systems

This paper investigates the acoustic and flow performance of an intake system using numerical and experimental techniques. The acoustic and flow performances are characterized by computing the Insertion Loss (IL) and the loss coefficient (LC) respectively. An indirect BEM formulation is used to predict the IL. The LC is computed by solving a one-dimensional fluid dynamics problem. For four simple cylindrical duct systems, numerical results for IL and LC are compared with experimental measurements. Finally, the acoustic and flow performance of an actual motorcycle intake is predicted and the results are compared to bench test results. [DOI: 10.1115/1.1471358]

1 Introduction

The intake system of an engine has three main functions. Its first and usually most identifiable function is to provide a method of filtering the air to ensure that the engine receives clean air free of debris. Two other characteristics that are of importance to the engineers designing the intake system are its flow and acoustic performance. The flow efficiency of the intake system has a direct impact on the power the engine is able to deliver. The acoustic performance is important because government regulations dictate the maximum noise level that vehicles can make during a pass-by test. The noise generated by the intake system can be a significant contributor to this pass-by noise. It may be noted that since the sound propagates from the carburetor towards atmosphere, this paper assumes the inlet is at the carburetor and the outlet is at atmosphere.

The intent of this paper is to investigate methods for predicting the acoustic and air flow performance of an intake system. Transmission Loss (TL) is one of the measures used for predicting the acoustic performance of an intake system. For ducted systems, TL is a transfer function of the sound power present at the inlet divided by the sound power present at the outlet. Although TL is adequate for measuring acoustic performance for simple duct systems, it has one significant draw back. Because of its assumption of an anechoic termination, it does not take into account the attenuation that results from the reflection of sound waves caused by the impedance mismatch at the outlet (snorkel) of the intake system, which is an important part of the acoustic performance of an intake system. Therefore, Insertion Loss (IL) is used as a measure of acoustic performance herein. For ducted systems, IL is a transfer function of the sound power emitted from an open pipe, with no intake system installed ($W_{\text{w/o intake}}$), divided by the sound power emitted from the outlet of the intake system after it is installed on the sound source ($W_{\text{w/intake}}$). In other words, $\mathrm{IL}_{\mathrm{dB}} = 10\log(W_{\text{w/o intake}}/W_{\text{w/intake}})$.

Both the Finite Element Method (FEM) and the Boundary Element Method (BEM) have been used to solve acoustical problems. Some notable applications of FEM to determine the acoustic properties of mufflers include [1,2]. For a comprehensive work on the acoustics of ducts and the use of FEM, see [3]. The application of BEM for muffler acoustics includes work of Tanaka et al. [4]. A method for measuring IL experimentally has recently been presented in [5]. For a comprehensive review on the use of BEM for acoustic modeling, see [6].

This paper provides a comparison between numerical and experimental measurements of acoustic and flow performance of intake systems. For four simple intake systems, the indirect BEM is used to predict IL whereas the loss coefficients are obtained by solving a one-dimensional fluid dynamics problem. The numerical methods presented herein are used to predict the IL and flow performance of a newly developed Harley-Davidson Twin Cam 88 intake system, and the numerical results are compared to experimental test results.

2 Numerical Modeling

Figure 1 presents the schematic of a simple intake system consisting of three cylinders. The first cylinder (inlet) with a length of L_1 and diameter of D_1 mounts up to the carburetor and provides the interface to the expansion chamber of length L_2 and diameter of D_2. The third cylinder (snorkel or outlet) has a length L_3 and diameter D_3.

For this research, the intake designs detailed in Table 1 were evaluated for their acoustic and flow performance. The designs in Table 1 are a subset of designs obtained by a 1/4 factorial design of experiments.

2.1 Calculating IL Using the Indirect BEM. For a noise source or vibrating surface surrounded by an acoustic medium, the acoustic pressure (p) in the domain surrounding the noise source is determined by solving the Helmholtz equation:

$$\nabla^2 p + k^2 p = 0 \tag{1}$$

Here $k=\omega/c$ is the wave number of harmonic waves of frequency ω and c is the speed of sound. The boundary conditions can be of Dirichlet, Neumann, or Robin type.

In ducted waves, since insertion loss (IL) relates the impact the duct has on the exterior sound power, the direct collocation BEM or FEM cannot be used. Instead the indirect BEM is used to compute IL. Using classical potential theory, the discontinuity in the acoustic variables due to the presence of the duct wall is expressed as

$$\mu = p^+ - p^-$$
$$\sigma = \frac{\partial p^+}{\partial n} - \frac{\partial p^-}{\partial n} \tag{2}$$

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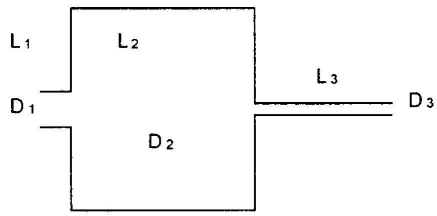

Fig. 1 Schematic of a simple intake system

In Eq. (2), μ is the difference in pressure between the positive and negative side of the "oriented" surface and is called the double layer potential. The difference in the normal derivative of pressure (σ) on the two sides is called the single layer potential.

Because of discontinuity, the pressure (p) is not differentiable near the boundary surface. However, for all smooth functions Ψ:

$$\int_V (\nabla^2 p + k^2 p)\cdot\Psi\cdot dV = \int_V (\nabla^2\Psi + k^2\Psi)\cdot p\cdot dV \tag{3}$$

The RHS of Eq. (3) can be split up as the sum of interior and exterior regions as:

$$\int_V (\nabla^2\Psi + k^2\Psi)\cdot p\cdot dV = \int_{V_i} (p\nabla^2\Psi - \Psi\nabla^2 p)\cdot dV_i + \int_{V_e} (p\nabla^2\Psi - \Psi\nabla^2 p)\cdot dV_e \tag{4}$$

which using Green's theorem and first and second layer potentials becomes:

$$\int_V (\nabla^2 p + k^2 p)\cdot\Psi\cdot.dV = \int_S \left(\sigma\Psi - \mu\frac{\partial\Psi}{\partial n}\right)\cdot dS \tag{5}$$

Equation (5) can be written in compact form:

$$(\nabla^2 + k^2)p = \sigma\delta_S + \frac{\partial}{\partial_n}(\mu\delta_S) \tag{6}$$

Here δ_S is the Dirac function. A convolution product of Green's function G with both sides of Eq. (6) yields:

$$G*(\nabla^2 + k^2)p = G*\left[\sigma\delta_s + \frac{\partial}{\partial_n}(\mu\delta_S)\right] = -p\delta \tag{7}$$

Upon further manipulation, Eq. (7) becomes:

$$p(X) = \int_S \left[\mu(Y)\frac{\partial G(X,Y)}{\partial n_y} - \sigma(Y)G(X,Y)\right] dS(Y) \tag{8}$$

Using Eq. (8), once (μ, σ) are known, the pressure anywhere in the volume can be calculated. The boundary conditions on the surface are translated into conditions for single and double layer potentials so that on each boundary surface there is only one unknown potential. The surface S is discretized and the potentials at any point on S are expressed in terms of nodal values as $\sigma = N_\sigma \cdot \tilde{\sigma}$ $\mu = N_\mu \cdot \tilde{\mu}$ and $\nabla\mu = B_\mu \cdot \tilde{\mu}$. Here, $\tilde{\sigma}$ and $\tilde{\mu}$ are vectors of nodal single and double layer potentials, N_σ and N_μ contain the shape functions and B_μ contains the cartesian derivatives of the functions in N_μ. With the functional J defined as:

$$J = \frac{1}{2}\tilde{\sigma}^T B\tilde{\sigma} + \frac{1}{2}\tilde{\mu}^T D\tilde{\mu} - \tilde{\sigma}^T C\tilde{\mu} - \tilde{\sigma}^T\tilde{f}_\sigma - \tilde{\mu}^T\tilde{f}_\mu \tag{9}$$

where matrices B, C, D require evaluation of double surface integrals and vectors $\tilde{f}_\sigma$ and $\tilde{f}_\mu$ are excitation vectors created by applied boundary conditions on the surface. A minimization of J yields the following system of equations:

$$\delta J = 0 \Leftrightarrow \begin{bmatrix} B & C \\ C^T & D \end{bmatrix}\begin{Bmatrix} \tilde{\sigma} \\ \tilde{\mu} \end{Bmatrix} = \begin{Bmatrix} \tilde{f}_\sigma \\ \tilde{f}_\mu \end{Bmatrix} \tag{10}$$

Once the solution for the unknown layer potentials is obtained, the pressure at any point inside or outside the volume is calculated as:

$$p(X) = \sum_e \int_{S^e}\left[N_\mu(Y)\tilde{\mu}\frac{\partial G(X,Y)}{\partial n_y} - N_\sigma(Y)\tilde{\sigma}G(X,Y)\right] dS^e(Y) \tag{11}$$

To determine the sound power emitted by the intake when it is installed on the open pipe ($W_{\text{w/intake}}$), a field point mesh is constructed around the outlet of the intake. Assuming plane wave propagation, the sound power (W_{inlet}) in the pipe is calculated as:

$$W_{\text{inlet}} = \rho c v^2(\pi/4)D_1^2 \tag{12}$$

where v is a known particle velocity $= 1$ m/s. The transmission coefficient of an open pipe, TC_{pipe} = incident sound power/transmitted sound power, is given as [7]

$$\text{TC}_{\text{pipe}} = \frac{(ka)^2}{\left[1 + \frac{(ka)^2}{4}\right]^2 + (0.6ka)^2} \tag{13}$$

Here k is the wave number and a is the pipe radius. Therefore, the sound power from an open pipe with no intake can be calculated as:

$$W_{\text{w/o intake}} = (W_{\text{inlet}}/\text{TC}_{\text{pipe}}) \tag{14}$$

Finally, the IL of the intake system (in dB) is given as:

$$\text{IL}_{\text{dB}} = 10\log(W_{\text{w/o intake}}/W_{\text{w/intake}}) \tag{15}$$

Typically, IL is dependent on the acoustic loading of the source by the intake system. However, by using this calculation, the calculated IL is dependent only on the acoustic performance of the intake system and is not influenced by potential loading of the modeled acoustic source. Also, from the correlation achieved between the predictions and test data, it can be concluded that the source in the test set up is of high enough impedance that it is not significantly loaded by the intakes evaluated.

2.2 Calculating LC Using One-Dimensional Fluid Flow There are 3 equations of importance when analyzing the flow of a compressible fluid in a pipe. These include the first law of thermodynamics for steady flow, continuity equation, and energy balance for a steady-state process, written as

$$\nu dp + \frac{VdV}{g} + dF + dz = 0 \tag{16}$$

In Eq. (16), p denotes the fluid pressure, z the fluid height, ν the specific volume, V the mean velocity, and F the energy used to overcome friction. Note that F is not just a function of the friction due to the air flowing along the pipe wall, it is also affected by any expansions (contractions) that the air experiences. By solving these three equations, the temperature, velocity, pressure, and density of the air can be determined at any location in the intake

Table 1 Geometric variables for four intakes evaluated

Design Number	D_1 (in.)	L_1 (in.)	D_2 (in.)	L_2 (in.)	D_3 (in.)	L_3 (in.)
1	2.00	0.25	3.00	3.00	2.00	8.00
2	2.00	0.25	3.00	10.00	1.00	8.00
3	2.00	0.25	9.75	3.00	1.00	2.00
4	2.00	0.25	9.75	10.00	2.00	2.00

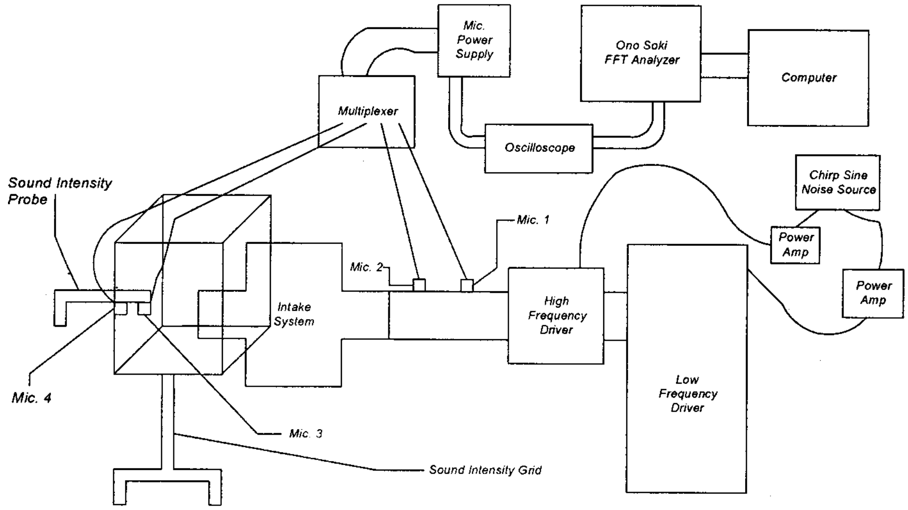

Fig. 2 TL measurement test set-up

system. Knowing the values for the pressure drop across the intake system (Δp), the air density at the outlet (ρ), and the air velocity at the outlet of the intake system (V), the Loss Coefficient (LC) is calculated as:

$$\Delta p=\tfrac{1}{2}(\mathrm{LC})*(\rho)*V^2 \tag{17}$$

3 Experimental Approach

The two set ups used to measure the IL and LC of intake systems are discussed next.

3.1 Experimental Measurement of IL. The experimental set-up used for measuring the IL of the intake system is shown in Fig. 2. The system was calibrated prior to testing. The chirp sine noise source was split into two power amplifiers with one driving a low frequency source (50 to 500 Hz) and the other driving a high frequency source (500 to 5000 Hz). This set-up allowed for gain to be adjusted on the two frequency ranges, thus increasing the signal-to-noise ratio across the whole band. The two drivers were connected to the intake system with a long pipe which insured plane wave propagation at the intake inlet. A sound intensity probe was used to obtain the sound power exiting the intake's outlet. The particle velocity (v) is related to pressure (p) as:

$$v=-\frac{1}{\rho c}\int \frac{\partial p}{\partial x}\,dt \tag{18}$$

The sound intensity (I) is defined as the average sound pressure ($\overline{p}$) times the particle velocity (v), $I=\overline{p}\times v$. Using the average sound pressure for microphones 3 and 4 as $\overline{p}=(p_3+p_4)/2$ the sound intensity is given as:

$$I=-\frac{p_3+p_4}{2\rho c\Delta x}\int (p_4-P_3)\,dt \tag{19}$$

Multiplying I by the surface area of the grid and summing the sound power over all six sides of the grid yields the total sound power emitted from the outlet of the intake system ($W_{\mathrm{w/intake}}$). Similarly, microphones 1 and 2 are used to measure sound intensity in the inlet tube as given by Eq. (19). By multiplying I by the cross-sectional area of the inlet, the sound power emitted into the inlet of the intake system (W_{inlet}) was calculated. With W_{inlet} known, by using Eqs. (13–14), the sound power from the open pipe ($W_{\mathrm{w/o\ intake}}$) was calculated. The IL is then calculated using Eq. (15).

3.2 Measuring LC Using a Test Bench. Figure 3 presents a schematic of the set-up used for determining the LC of an intake system. The setup utilized a compressor to create a vacuum in a long pipe. The current to the compressor was adjusted to provide 50, 70, and 90 ft^3/min of air through the intake system. If possible, a fourth data point was collected at the highest flow rate the compressor could deliver. The compressor was allowed to run at each measurement point for several minutes until the flow, temperatures, and pressure readings stabilized. Based on the pressure and temperature, the air density was calculated. The velocity of the air was calculated knowing the flow rate and cross-sectional area of the pipe. Finally, the LC was calculated using Eq. (16).

4 Comparison of Numerical and Experimental Results

As mentioned earlier, indirect BEM is used to predict the IL of intake systems. Figures 4–7 present the IL predictions as well as

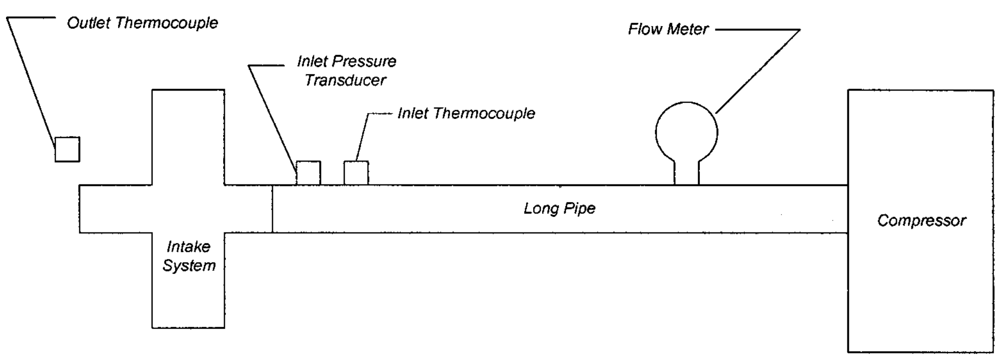

Fig. 3 Schematic of flow performance measurement set-up

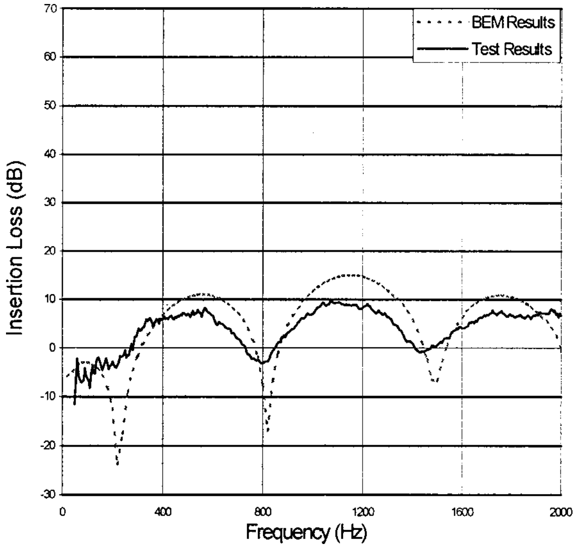

Fig. 4 IL results for intake 1

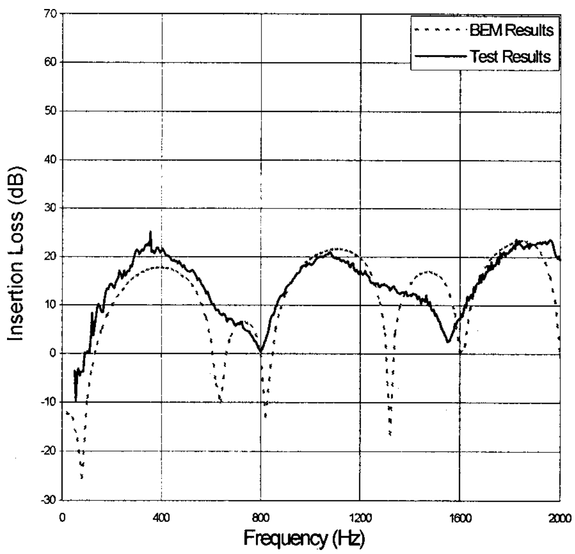

Fig. 5 IL results for intake 2

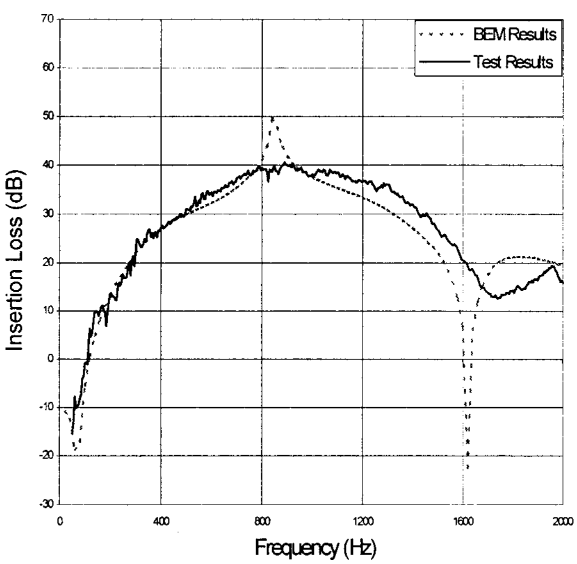

Fig. 6 IL results for intake 3

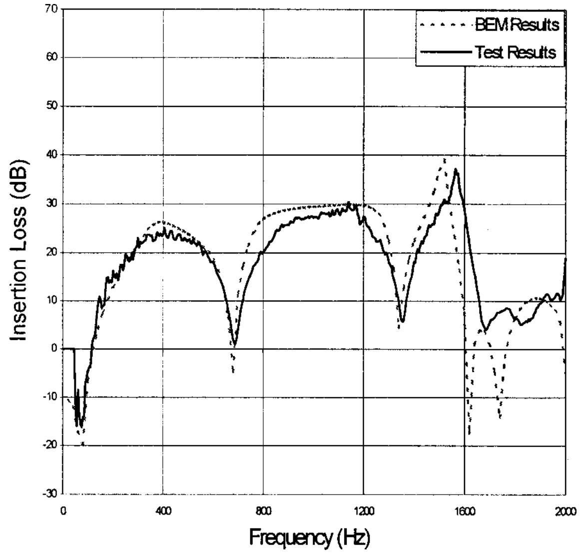

Fig. 7 IL results for intake 4

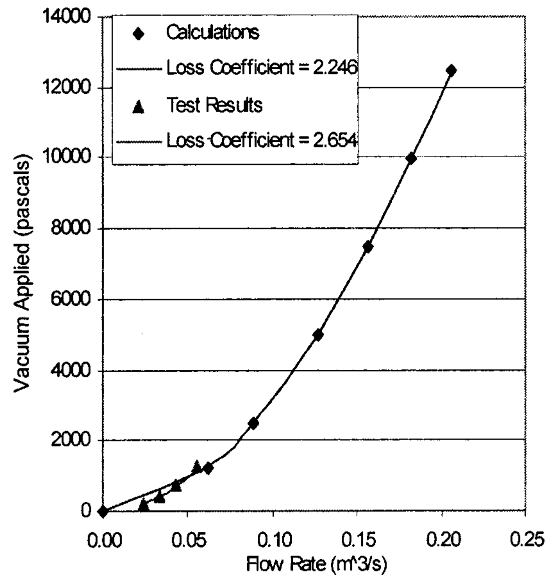

Fig. 8 Flow results for intake 1

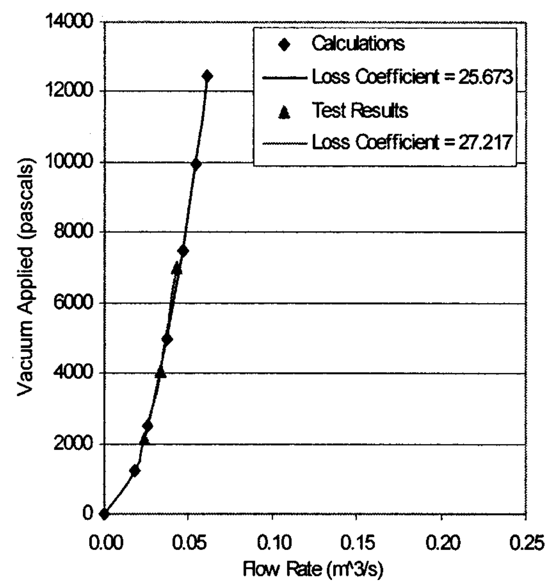

Fig. 9 Flow results for intake 2

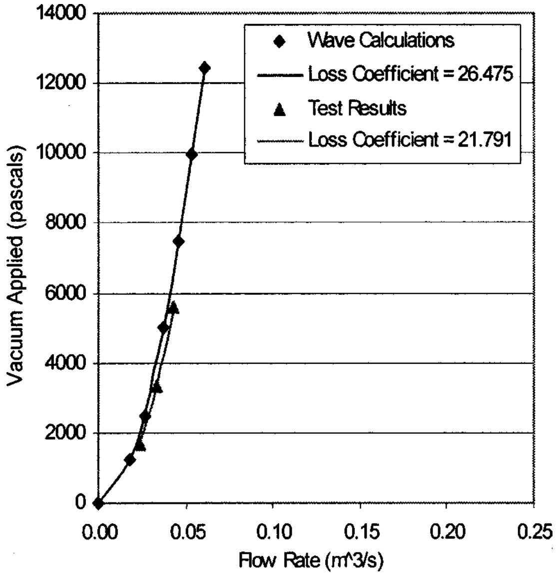

Fig. 10   Flow results for intake 3

the IL measurements for four intake systems. It may be noted that there is a lot more damping in the test curves than in the prediction curves. This could be an effect of several causes. First, the modeling technique used assumed no coupling between the air and the walls of the intake system, but in actuality there is coupling. In fact, since intakes 3 and 4 exhibited a very noticeable pumping due to the sound energy, these intakes were lagged with damping and barrier material to reduce coupling effects on the test results. Secondly, the modeling technique assumes no damping in the air itself. In reality a small amount of damping to sound waves does occur due to absorption caused by the viscosity and humidity of the air.

Figures 8–11 present the numerical air flow predictions as well as the flow bench test measurements for each of the four intakes. These figures show that the predicted and tested LC's are very well correlated with an average 3.7% difference.

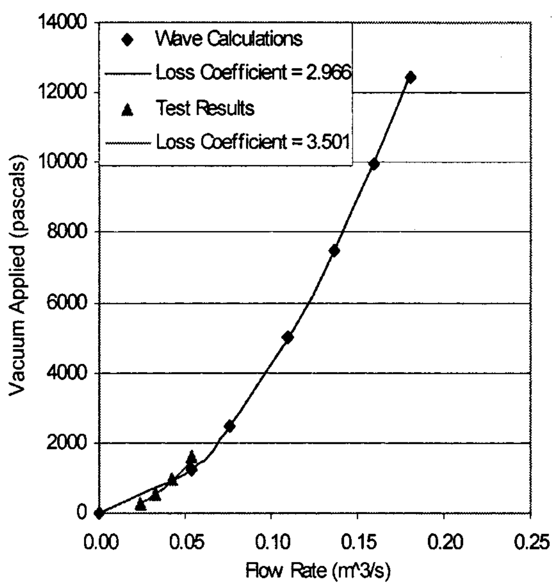

Fig. 11   Flow results for intake 4

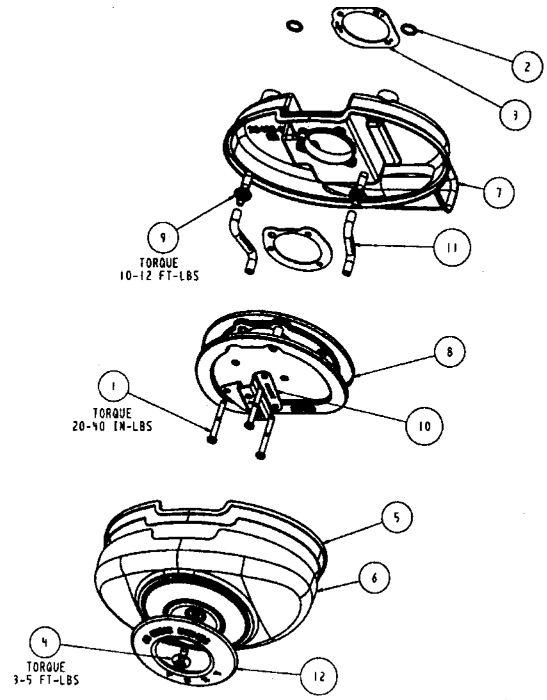

Fig. 12   HDI twin cam 88 intake system

5 The Harley-Davidson International (HDI) Twin Cam 88 Intake System

Due to very different government regulations in the markets in which Harley-Davidson sells motorcycles and the differing inlet dimension requirements of EFI and carbureted engines, the new Twin Cam 88 Harley-Davidson engine receives five different intake systems. Figure 12 presents an exploded view of the carbureted HDI intake system designed to meet stricter pass-by laws of the European Union. The asymmetric design, filter element, and flanged snorkel of the backplate make this model significantly more complex than the simple intakes previously presented.

The BEM model required for the IL calculation was generated by assembling the CAD solid model of the cover (6 in Fig. 12), and filter element (10 in Fig. 12) to the backplate (7 in Fig. 12). The inlet of the backplate was meshed over, allowing for a known

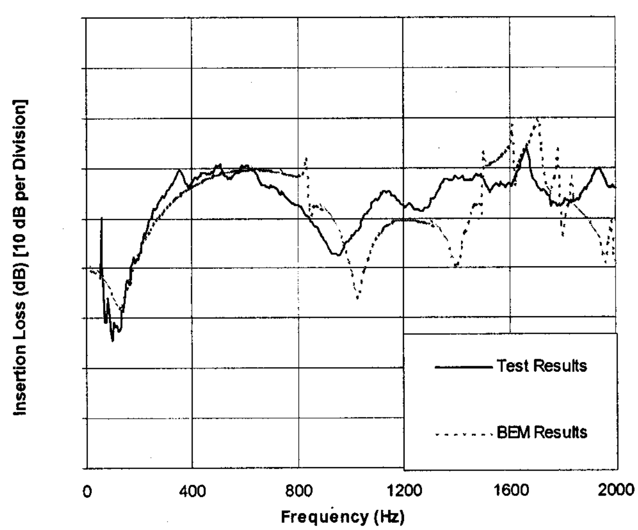

Fig. 13   Insertion loss of the twin cam HDI intake system

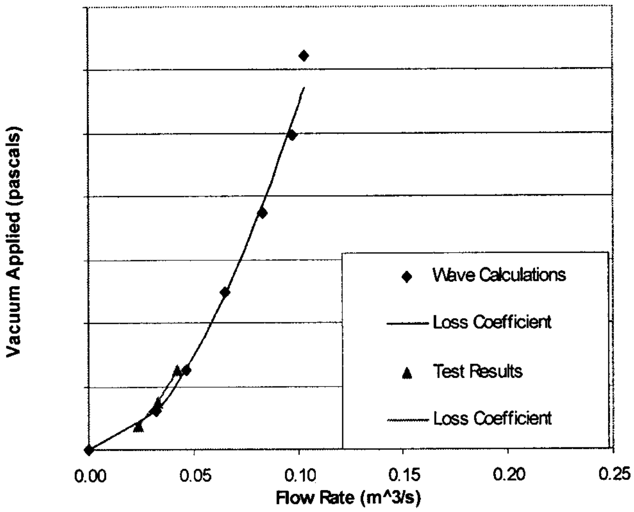

Fig. 14 Flow performance of the twin cam HDI intake system

velocity to be applied. A separate field point mesh was made over the outlet of the backplate which was used to obtain the sound power out of the intake system. Figure 13 presents the IL predictions using indirect variational BEM and test results for the Twin Cam 88 HDI intake system. The prediction provides good correlation to the test results. This follows the trend experienced with the simple intake systems. Figure 14 presents the flow performance of the HDI Twin Cam 88 intake system. The LC and flow results of the actual air cleaner are very close to the numerical predictions.

In conclusion, based on the numerical and experimental testing performed on the intake systems, it was seen that:

1) The indirect BEM provides a reliable method for predicting the IL of intake systems. The method works not only for axisymmetric intake systems, but also for more complex designs such as the HDI Twin Cam 88 intake.

2) The one-dimensional fluid dynamics solver accurately predicts the LC of axisymmetric intake systems as well as the HDI Twin Cam 88 intake system.

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