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Virtual design and performance prediction of a silencing air cleaner used in an I.C. engine intake system*

HAO Zhi-yong (郝志勇)[†1], JIA Wei-xin (贾维新)[†‡1], FANG Fang (方 芳)[2]

([1]College of Mechanical and Energy Engineering, Zhejiang University, Hangzhou 310027, China)

([2]Tianjin Internal Combustion Engine Research Institute, Tianjin University, Tianjin 300072, China)

[†]E-mail: haozy@zju.edu.cn; jiawx@zju.edu.cn

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Abstract: This paper reports results of the authors' studies on the virtual design method used in the development of low noise intake system of I.C. engine. The resulting high pass-by noise at level above the legislative target at full throttle when engine speed was around 5200 r/min necessitated a BEM-aided redesign task, following the typical process of design and development of an intake system. During the initial design, based on the acoustic theory and the requirements (1. The air flux of the redesigned should equal to or exceed the value of the original flux; 2. The filtering area must not be degraded), and considering the constraint of space in the engine compartment, total volume and rough internal dimensions were determined. During the detailed design, the exact internal dimensions of the air cleaner were determined, and an effective method was applied to improve the acoustic performance at low frequency. The predicted sound power of the intake system indicated that the objective of reducing the overall engine noise by minimizing intake system noise was achieved.

Key words: Virtual design, Acoustic performance, Silencing air cleaner, Boundary element method (BEM)
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INTRODUCTION

The primary function of an intake system is firstly to efficiently channel fresh air to the engine, and secondly to minimize intake noise emissions. There are a number of current approaches for developing a more realistic method to improve intake system design. The objectives include more effective silencing performance to meet increasingly severe legislative targets for reduced noise on the one hand, with optimized engine performance and fuel economy accompanied by improvements in vehicle quality on the other hand (Davies, 1996).

A typical procedure followed during the design and development of an intake system for a vehicle engine is shown in Fig.1. The design process includes a careful tuning of all components of the intake system that influence noise emission with optimized matching of these to the engine operational and breathing characteristics influencing pollutant emission, performance and economy. As Fig.1, starting with an existing or notational system layout, an integrated assessment of the various performances is performed. This information may then be used appropriately to assess current system performance in terms of the various design objectives, to provide rational basis for systematic optimization of the design by implementing appropriate modifications to its constituent elements.

The BEM (boundary element method) widely used in the design of intake and exhaust system (Bilawchuk and Fyfe, 2003; Wu *et al*., 2003; Selamet and Radavich, 1995) can be used to compute the interior, exterior, or both fields simultaneously and only requires that the perimeter of the air cleaner be divided into elements; and the ease in imposing the boundary condition is another attraction. In this paper BEM is

[‡] Corresponding author

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used to predict the air cleaner's transmission loss (TL) and noise emission.

The redesign of the original intake system arises in connection with a high pass-by noise with level above the legislative target at full throttle with engine speed around 5200 r/min. The spectral characteristics of the noise signal are normally dominated by an extensive sequence of discrete tones that are harmonically related to the engine firing frequency (Fang, 1978) which is 173 Hz corresponding to 5200 r/min and the inline 4-cylinder 4-stroke engine. In many instances the bulk of the acoustic energy from the primary source is distributed among the lower frequency components that may be difficult to control. Thus frequency range this paper referred to is from 0 to 1 kHz. In this frequency range, as the influence of filter paper on acoustic performance of integrated system is trivial, so filter paper is disregarded. The noise is emitted from the inlet of the air cleaner, and the outlet of the air cleaner system (containing the air channeling pipe) connects to the inlet of the engine. So the pressure at the inlet of the engine or the outlet of the air cleaner is the boundary condition for the BEM.

Traditional intake system with a function of noise reduction normally has two parts: air cleaner and silencer. Due to the constraint of space in the engine compartment, the redesigned air cleaner combines the effect of cleaning and silencing. In this work, a so-called silencing air cleaner was redesigned, with geometrical structure determined by predicted TL and sound power emission by BEM. Also, in order to minimize the sound power of the intake system at low frequency, a bypass pipe was added to the air-channeling pipe. Analysis of the resulting acoustic performance showed that the method is feasible for the goal of reducing the overall engine noise by minimizing the intake system noise.

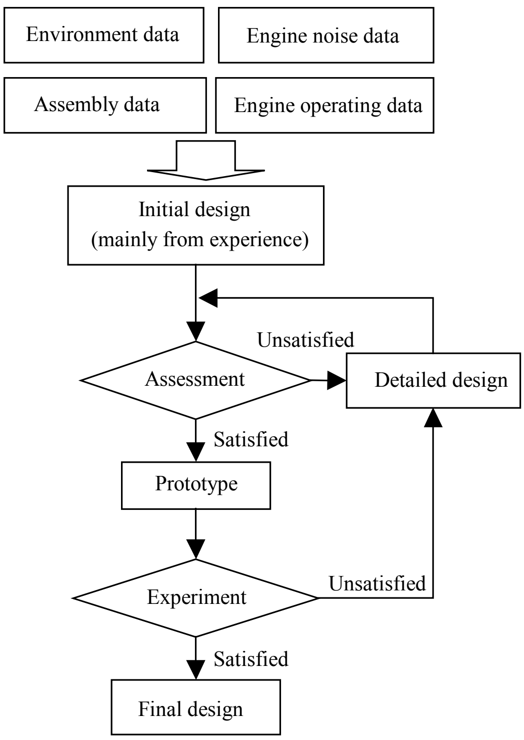


Fig.1 Typical process of design and development of an intake system

DESIGN OF THE SILENCING AIR CLEANER

Original air cleaner evaluation

This original air cleaner of mainly cleaning design paying little attention to minimizing noise emission is shown in Fig.2a. Fig.2b is its BEM mesh.

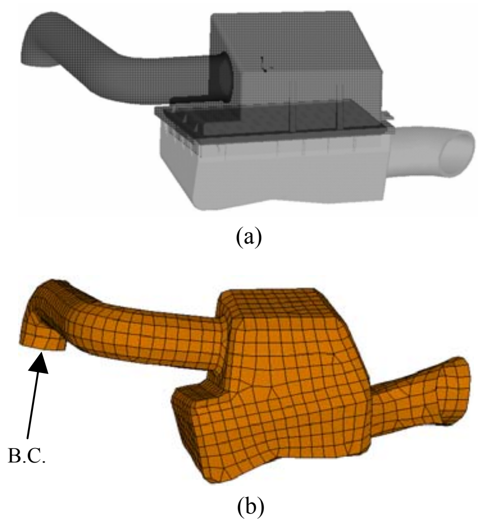
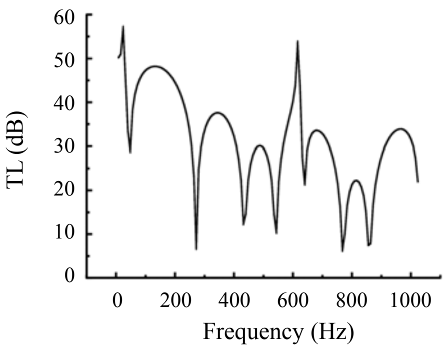


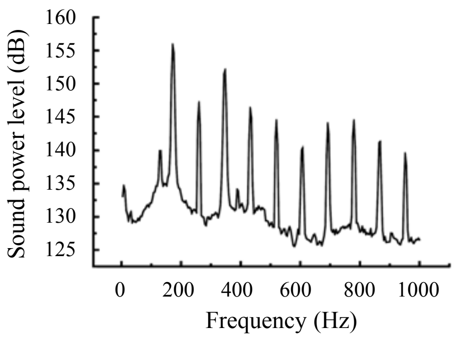
Fig.2 CAD model (a) and BEM mesh (b) of original air cleaner

During calculation of TL, the outlet section of the air cleaner is given a unit velocity amplitude to model a sound source, all other surfaces are modeled as "acoustically hard" by default (Bilawchuk and Fyfe, 2003). The sound power of the outlet section can be calculated from the formula: $P_{w1}=S\rho_0 c_0|u|^2/2$, where S is area of the outlet section, ρ_0 is density of the medium, c_0 is speed of sound, u is vibration velocity of the medium. The sound power of the inlet port P_{w2} was calculated by BEM, so $TL=P_{w1}-P_{w2}$ (Fang, 1978). All the subsequent TL predictions have the same boundary condition.

The predicted air cleaner performance is shown in Fig.3a. The sound power level at the engine intake port (noise source) is shown in Fig.3b. The peaks of the noise source signal are harmonically related to the engine firing frequency of 173 Hz. As the bulk of the acoustic energy from the engine combustion distributed among the low frequencies from 100 Hz to 800 Hz is too high, the transmission loss at the frequency range of 220 Hz to 1 kHz is so low that the noise emission cannot be minimized, so a silencing air cleaner with a higher TL at 200 Hz to 1 kHz is required.



(a)



(b)

Fig.3 Transmission loss of original air cleaner (a) and sound power level at engine intake port (b)

Initial design of the silencing air cleaner

If there is sufficient space, a complex structure can be assigned to minimize the intake noise emission. So we must make good use of the limited space. In this work CAD software Pro/E was used to envelop the rest of the space of the other automotive components in the engine compartment. Then the total air cleaner volume required is obtained from the enveloped space. The next step is to choose appropriate silencer units and their dimensions. Considering the effect of air cleaning, two requirements must be satisfied: (1) The air flux should equal to or exceed the value of original flux; (2) The filtering area must not be degraded. Based on the requirements and the theory of the silencer units, an initial layout was determined, as shown in Fig.4. The air cleaner is separated into three expansion chambers by two baffles, the right baffle has a hole in the middle, and the left baffle has four holes at its four corners respectively. The filter paper is placed at the center of the middle expansion chamber. The diameter of the hole in the right baffle is determined by the first requirement; because of the complexity of this air cleaner, the diameter of the hole is bigger than the original one. The second requirement (filtering area limitation) relates to the diameter of the hole in the first baffle and the length of the second chamber, after letting the filtering area be equal to the original one, then the length of the second expansion chamber can be determined. So the geometrical structure of the air cleaner is set by two variables: the length of the first chamber L_1 and the diameter of the holes in the second baffle D, shown in Fig.4.

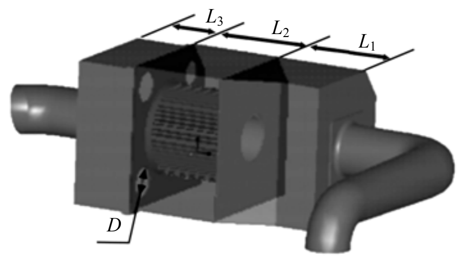


Fig.4 Layout of the initial design of the silencing air cleaner

The overall acoustic behavior is a summation of the behavior of all constituent components of the three chambers. In order to provide a wide bandwidth of continuous attenuation spectrum, the attenuation minimum in their individual contributions should not occur simultaneously.

According to Eq.(1):

$$f_n=nc/4l \text{ (Hz)}, n=1,3,5,\ldots \quad (1)$$

where f_n is the frequency corresponding to attenuation maxima, l is the length of the chamber, c is speed of sound (Du, 1981).

One should avoid such coincidences by satisfying the following inequality:

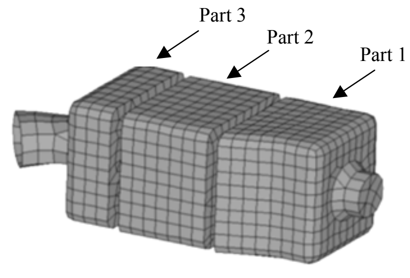
$$L_1/i \neq L_2/j \neq L_3/k, \quad i,j,k=1,2,3\ldots \tag{2}$$

where L_1, L_2 and L_3 are the lengths of the expansion chambers respectively in the silencing air cleaner. The best situation is that the attenuation peaks of one expansion chamber correspond to the minima of the other chambers.

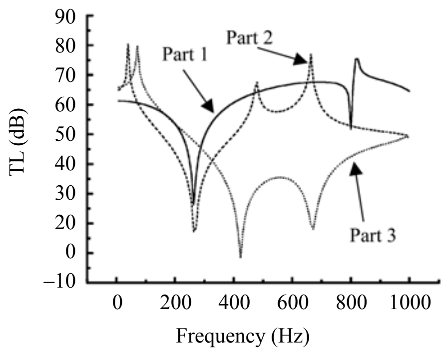
Choosing L_1=165 mm, L_2=120 mm (determined by the two requirements) and L_3=80 mm (determined by overall length and L_1), by Eq.(1), one can achieve the first maximum attenuation of the first chamber at the frequency of about 515 Hz, that of the second chamber at 708 Hz, and the third chamber at 1062.5 Hz. The first maximum attenuations of the three chambers occur alternately, thus providing a continuous effective attenuation spectrum.

In order to investigate the individual acoustic performance of the three chambers, the integrated air cleaner was separated into three parts, one of which is a silencer unit of expansion chamber, just at the place of the two baffles, shown in Fig.5a. BEM run was performed to calculate the individual acoustic performance of the three chambers. The information is shown in Fig.5b, showing that the first expansion chamber has significant effect in minimizing the intake noise emission. The first chamber has high and continuous attenuation at the frequency band of 300~1000 Hz, and the acoustic performance of the second chamber which has high attenuation at the frequency range of 400~800 Hz is a shade worse than that of the first one. The third chamber has the worst performance, but at the frequency of about 230 Hz, where the first and the second chamber have minimal attenuation, it has higher attenuation, thus providing compensation effect for the first and second parts. Also, at higher frequency, the third part possibly represents good acoustic performance, which is not illustrated in this figure.

Note that the acoustic interactions that exist between such single parts also introduced uncertainties, even when their individual performance can be appropriately represented or modeled. Clear understanding of the acoustic performance of the single part gives significant information for initial design (Lan et al., 2001). The integrated system with changes in some areas is calculated by the method of BEM in the detailed design to gain the best acoustic performance.



(a)



(b)

Fig.5 Three parts of the air cleaner (a) and TL of the three parts (b)

Detailed design

In the initial design, the external geometrical dimension is decided, and silencer unit with a rough dimension is chosen. While the position of the baffles and the radius of the four holes in the second baffle can be varied to achieve better acoustic performance, the detailed design should be performed by BEM. Fig.6 is the BEM mesh of the air cleaner.

Fig.7a compares the predicted TL of air cleaner while varying the length L_1, which represents the position of the baffles. In this case, the parameter L_1 of the BEM samples is 145 mm, 155 mm, 165 mm, 175 mm respectively. One can observe that at lower frequency the four lines agree very well, and that at frequency band of 500 Hz to 1000 Hz, the amplitude of TL with L_1=145 mm is much higher than that of the other samples, especially at around 500 Hz where the other samples have bad TL, so the position of baffles is determined by the dimension L_1 (L_1=145 mm).

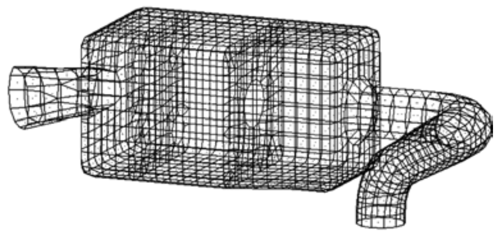


Fig.6 BEM mesh of the silencing air cleaner

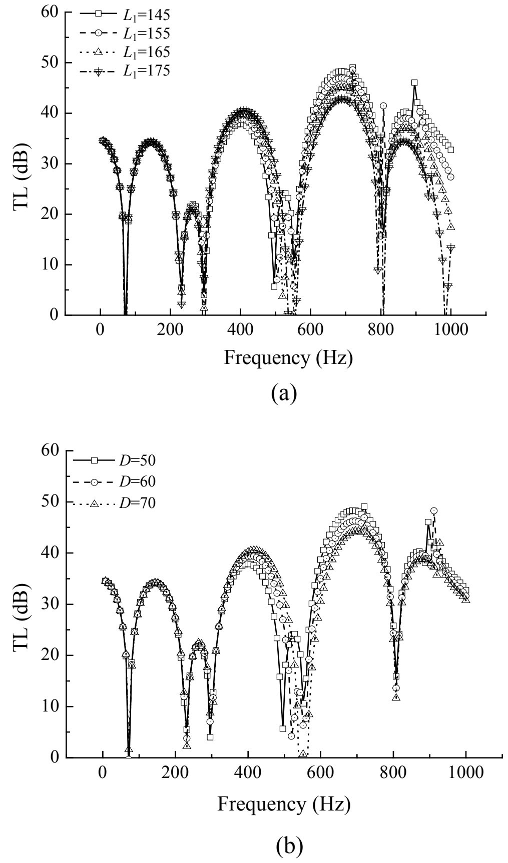


Fig.7 TL respectively when L_1=145, 155, 165, 175 mm (a) and TL respectively when D=50, 60, 70 mm (b)

Now maintaining the position of baffles at the position of L_1=145 mm, while varying D representing the diameter of the four holes at the second baffle, the comparison of the predicted TL of air cleaner with D=50 mm, 60 mm, 70 mm respectively is illustrated in Fig.7b. Basically, the three lines have the same tendency as that in Fig.7b at frequency band of 500 Hz to 1000 Hz, the amplitude of TL with D=50 mm is much higher than that of the other samples. So the diameter of the holes is chosen to be 50 mm.

Noise emission prediction

Until now, we have accomplished total design of the silencing air cleaner. In order to make a comparison with the original, prediction of the noise emission was carried out.

To verify the practical performance of the redesigned air cleaner, the sound pressure at the engine intake port was measured at full throttle with engine speeds at 5200 r/min at engine test bed as the boundary condition or noise source of the original and the redesigned air cleaner, then predicting the noise emission from the inlet of the air cleaner.

When measuring this sound pressure, the air cleaner was removed, and the engine noise was shielded off. In addition, due to the difficulties in measuring the pressure at the engine intake port because of the air flow influence on noise signals and the induced microphone disturbance that deteriorate the engine performance, the measurement location of the microphone was at the place 200 mm to the intake port, and at 45° to the normal of the intake port section (Park *et al.*, 2002). The boundary condition applied at the outlet of the air cleaner system should be extracted from the noise signals at the measurement location mentioned above based on acoustic theory.

Assume that the engine intake port is a round plane cylinder radiator (Xu, 2003), with characteristic of direction.

The sound pressure of the point p_0 at the axis of the intake port is the maximum compared to the other points with the same distance to the center of the intake port section O, it is:

$$p_0 = \frac{j\rho_0 c_0 k}{2\pi r} \mathrm{e}^{-\mathrm{j}kr} \iint_s u_0 \mathrm{d}s ,$$

where $\rho_0 c_0$ is the acoustic impedance of medium, r is the distance from point O to p_0, $u_0=u_{0m}\mathrm{e}^{\mathrm{j}\omega t}$ is vibration velocity of the radiator surface, k is the wave number.

Direction function:

$$R(\theta) = \frac{p_\theta}{p_0} = \frac{2}{a^2}\int_0^a \rho J_0(k\rho\sin\theta)\mathrm{d}\rho = \frac{2J_1(z)}{z},$$

where p_θ is the sound pressure of the point p at angle θ to the normal of the radiator, and at the same distance to the center of the radiator as p_0, $z=ka\sin\theta=\dfrac{2\pi}{\lambda}a\sin\theta$,

a is the radius of the radiator, J_0, J_1 is zero order and first order Bessel function respectively.

All the discussion above is on condition that a long distance from radiator to p_0 is assumed. Next let us pay some attention to the Fresnel region $r \le a^2/\lambda$, the near field and the far field boundary of the round cylinder radiator. The engine intake port radius is a=40 mm; for the frequency of 1 kHz, the radius of the Fresnel region is $r \le 4.7$ mm. As the distance from the measurement location to the radiator is 200 mm, the characteristic of the round cylinder radiator is similar to that of the far field.

As the vibration velocity on the radiator surface is uniform, then $\iint_s u_0 \mathrm{d}s = u_0 s$, and the sound pressure amplitude at the radiator axes can be obtained as:

$$p_{0m} = \frac{\rho_0 c_0 ks}{2\pi r} u_{0m} = \frac{\rho_0 c_0 s}{\lambda r} u_{0m} .$$

So the amplitude of sound pressure at arbitrary point is:

$$p_{\theta m} = p_{0m} R(\theta) = \frac{\rho_0 c_0 s}{\lambda r} u_{0m} R(\theta).$$

Then the amplitude of the radiator vibration velocity is:

$$u_{0m} = \frac{c_0 r}{\rho_0 c_0 sfR(\theta)} p_{\theta m}.$$

From the property of plane wave propagation (Du, 1981), the sound pressure at the radiator surface is:

$$p_m = \rho_0 c_0 u_{0m} = \frac{c_0 r p_{\theta m}}{sfR(\theta)}. \quad (3)$$

The expression above can be used to obtain engine intake port sound pressure for modeling the boundary condition of the air cleaner system. In order to enhance the precision of p_m, $p_{\theta m}$ is taken as the average sound pressure of four points in this paper.

The medium particle velocity calculated using Eq.(3) is imposed as the air cleaner's boundary condition. Then, the BEM run is performed and the inlet section sound power level is determined as shown in Fig.8. In the frequency band of 400~1000 Hz, the air cleaner emitted sound power is dramatically reduced, compared with the original one, but is still high at around firing frequency of 173 Hz, so extra measures should be introduced.

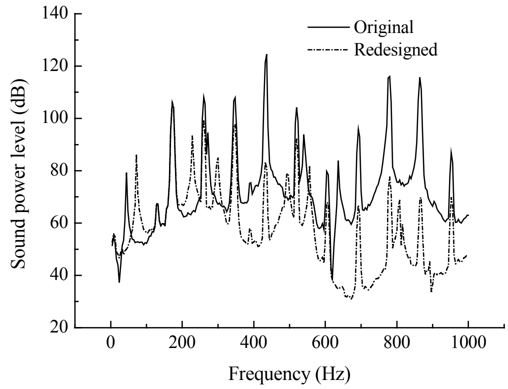


Fig.8 Sound power level of the original and the redesigned air cleaner's intake noise

EXTRA METHOD TO MINIMIZE SOUND POWER

One potential problem in using the previous layout is that at the frequency of 173 Hz the level of sound power emitted from air cleaner's inlet is not as low as that of other frequency bands according to Fig.8. The vacant space around the pipe connecting the air cleaner and the engine intake port suggests some other possible improvements of the air cleaner system. This paper uses a bypass pipe to achieve this goal.

Fig.9a presents the method with a flexible bypass pipe fixed on the air-channeling pipe. The theory of bypass pipe is illustrated in Fig.9b (Hwang *et al.*, 2003).

So the length difference of the two paths corresponding to noise reduction frequency f can be expressed as:

$$L_d = n\lambda/2 = cn/2f, \quad n = 1,3,5\cdots$$

Increasing of TL at frequency of around 173 Hz being the goal determines that dimension L_d=980 mm. Fig.10a compares the predicted acoustic performance between the previous layout with no change on pipe

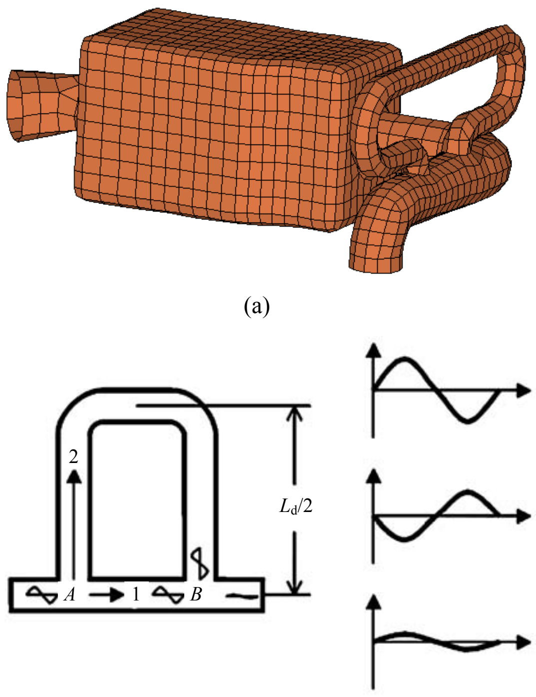


Fig.9 Air cleaner with bypass pipe (a) and theoretical effect of bypass pipe (b)

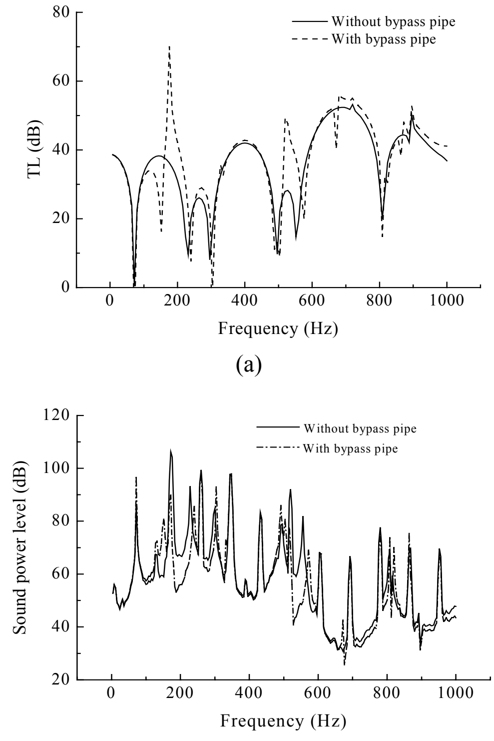


Fig.10 TL (a) and sound power level (b) of air cleaners with and without bypass pipe at the main pipe

and the present layout with flexible bypass pipe. The acoustic performance of the bypass pipe layout improved dramatically at frequency of around 173 Hz, and at frequency of 519 Hz, corresponding roughly to thrice of half wavelength, the TL of bypass pipe layout increased lightly, as expected.

The resulting sound power level is illustrated in Fig.10b. It can be seen that sound power level of the without-accessory air cleaner at 173 Hz is the highest, while that of the with-bypass-pipe air cleaner at this frequency is greatly decreased. The overall sound power level from 0 to 1 kHz, for the without-bypass-pipe is 110.2 dB, and for the with-bypass-pipe air cleaner is 105.0 dB.

Note that the engine sound power level without intake system noise from experimental testing is 112.2 dB, and the silencing air cleaner with bypass pipe is 105.0 dB, so the redesigned silencing air cleaner feasibly reduced the overall engine noise by minimizing the intake system noise.

CONCLUSION

This paper reports the designing of a BEM-aided silencing air cleaner, following the procedure shown in Fig.1. During the initial design, based on the acoustic theory and the requirements (1. The air flux of the redesigned should equal to or exceed the value of the original flux; 2. The filtering area must not be degraded), and considering the constraint of space, the total air cleaner volume and the rough internal dimensions of the air cleaner were determined. During the detailed design, the exact internal dimensions of the air cleaner were determined, after which an effective method was applied to improve the acoustic performance at frequency of 173 Hz.

The use of the boundary element method to help in acoustical engineering design is increasing rapidly. The study results in this paper provided guidelines for engineering application of silencing air cleaner.

Flow effect was not considered in this study. Although the mean flow may not have significant effect on the acoustic performance, it may have some effects on the air cleaning performance. The effect of filter paper was ignored, although it may influence the acoustic performance of the air cleaner at high frequency. Future study should include the flow effect on the air cleaning and the influence of the filter paper on the acoustic performance at high frequency.

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