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A Comprehensive Methodology to Design a Helmholtz Resonator for Two-Wheeler Induction Noise Reduction

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Summary
In the recent past there is a steep increase in customer's preference for powerful vehicles that are yet quieter. It is very evident that careful attention needs to be given during design phase to meet such contradictory requirements. These challenges are even more complex in case of two wheelers because of limited design space, cost restrictions and high power-to-weight ratio. The major noise sources in a two wheeler include induction system, exhaust system and engine radiated noise. There has been a great focus on reduction of noise from sources like exhaust and radiated engine by researchers in the past while very little attention was given to induction noise. Traditionally in case of two-wheelers, an induction system is designed giving paramount importance to engine performance with respect to power and torque developed. Induction noise arising out of such a design is dealt by changing material properties and stiffness of panels — as this design is already frozen for engine performance attributes and no freedom exists for optimization of entire induction system. This approach will have a very limited success in the case of aero-acoustic noise, as design changes mentioned above will have least impact on reduction of induction noise. This paper aims to address these concerns by designing a suitable Helmholtz resonator that helps in reduction of induction noise with very little changes to the layout. A Matlab program is developed to arrive at a design with maximum Transmission Loss (TL) of resonator. This paper also highlights the challenges involved in incorporating a Helmholtz resonator for induction noise reduction in case of two-wheelers — challenges that are not dealt by previous researchers in detail according to authors' knowledge. In this paper, a successful methodology is proposed for design and development of an intake resonator for reduction of induction noise. This methodology is proven using objective noise measurements on three different vehicles; achieving at least 5 dB(A) noise reduction at snorkel and 1.5 dB(A) noise reduction at pass-by location. The effect of wall flexibility of resonator is studied to optimize resonator's performance. The side effects of introduction of a resonator on engine performance are studied using representative metric like wheel force. Finally, a subjective-to-objective correlative measure for induction noise — a 5 dB(A) reduction at inlet will improve subjective rating by 3 points — is established.

PACS no. 43.50.-x

1. Introduction

Major noise sources in two-wheelers are induction, exhaust system and engine radiated noise. It can be found out from noise source ranking that engine radiated noise is of significant importance followed by exhaust and intake systems [1]. Hence researchers addressed engine radiated noise reduction in greater detail. Sekita *et al.* [2] demonstrated how the crank noise radiated from the engine of a motorcycle often spoils the product perception because of its annoying sound. In their paper, a new method using virtual modelling of a single-cylinder engine of a motorcycle that succeeds in identifying the mechanism behind the generation of annoying noise is proposed. A combined experimental and numerical approach was adopted to obtain the necessary accuracy. Vamsi Krishna *et al.* [3] proposed a novel approach to reduce crank noise by optimizing the crankcase geometry using topology optimization. The topology optimization was carried out to identify the structurally weaker regions yielding to higher surface velocities that in turn potentially radiate higher noise levels. These regions were stiffened using ribs thus by adding weight only at appropriate locations resulting in minimum weight penalty. Noise simulation and experimental results showed 2 dB(A) reduction in both idle and rev-up noise levels with the modified design. Jones [4] developed a computer model based on linear acoustics that predicted tail pipe noise. He *et al.* [5] made a modular muffler design where the length and radius of the muffler pipe, ex-

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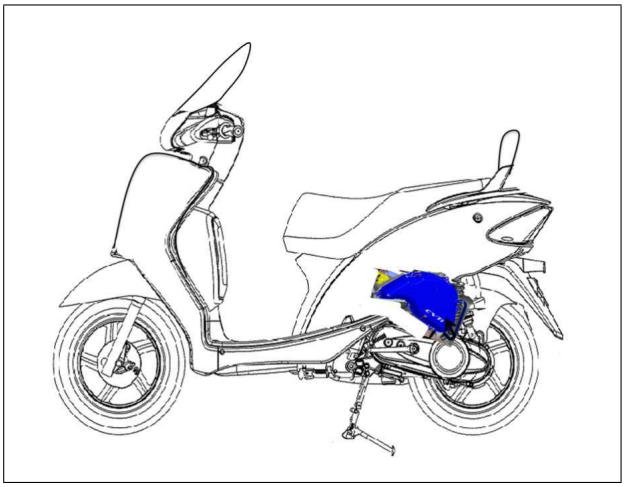

Figure 1. Schematic diagram of vehicle fitted with the induction system.

pansion cavity, size or shape of the muffler tailpipe can be varied to understand noise reduction. Ghodake *et al.* [6] proposed a novel method of coupling a one dimensional GT Power [7] engine model to a three dimensional Computational Fluid Dynamics (CFD) model using ANSYS Fluent [8]. Inlet pressure wave modelling is carried out using GT-Power and is provided as an input boundary condition to ANSYS Fluent. Though the proposed method overcomes challenges of simulating the entire domain in three dimensional space, it is still cumbersome and it may warrant a complete redesign of an induction system. Shaw *et al.* [9] demonstrated that Boundary Element analysis (BEA) can be used to predict the frequency response of ducted systems and is particularly useful in three-dimensional systems. In their paper engine simulation is done to obtain the inputs to the BEA for the prediction of radiated noise levels while including the frequency or order content of air induction noise. Shugong [10] also demonstrated the use of boundary element methods for predicting induction system noise. These above discussed methodologies will be of help only to evaluate the effect of material properties and structural design modifications. However, experience shows, in most cases, induction system noise is result of improper snorkel breathing. This problem is addressed by modeling entire fluid domain in three dimensional space using CFD to gain insight of physical phenomenon and making appropriate design changes. This approach will not be of great help if induction system is already finalized w.r.t engine performance and only induction noise needs to be reduced without altering engine performance characteristics. In this paper a novel methodology is demonstrated for reducing induction system noise using a Helmholtz resonator, without causing major changes to the layout. To the best of the authors' knowledge, there are no studies that describe any methodology to incorporate a resonator to reduce induction system noise in two-wheelers. The present paper discusses the methodology of design and development of a resonator system applicable to a two wheeler. The proposed design had reduced noise levels — achieving at least 5 dB(A) noise reduction at snorkel and 1.5 dB(A) noise reduction at pass-by location — of three different vehicles of the same model. The effect of wall flexibility of resonator is also studied to obtain the best possible noise reduction using a resonator.

The side effects of introducing a resonator on engine performance parameters are studied in a holistic manner without getting into nuances, as such a detailed study is out of scope of the present investigation. Wide Open Throttle (WOT) wheel force and air-to-fuel ratio are measured on a chassis dynamometer and are used as a metric for qualifying the resonator-fitted design with respect to engine performance. A subjective testing of both standard and resonator fitted design is performed using six different subjects in actual usage condition. It was found out that subjective rating of modified design is better than that of standard design. Finally, an attempt is made to establish a subjective-to-objective correlation for induction noise thus substantiating the findings of noise source ranking in the case of a two wheeler.

2. Experimental Procedure

The vehicle, on which induction noise experimental studies are carried out, is equipped with a 110 cc engine which has an idling speed of 1500 rpm and a maximum engine speed of 8000 rpm. It is a four stroke, single cylinder and air cooled engine. It has a Continuously Variable Transmission (CVT) and an induction system with 2.8 litre capacity. This engine develops 8.5 Nm torque at 8000 rpm and power of 8 bhp at the same rpm. This induction system is already optimized for engine power and torque and hence there is no freedom to redesign to optimize considering both noise and engine performance parameters. The schematic arrangement of the vehicle along with induction system is shown in Figure 1.

The objective noise measurements of the vehicle are carried out by running the vehicle on a chassis dynamometer in a semi-anechoic chamber. The vehicle is tested at the following conditions.

- The near field engine noise levels — to quantify induction noise — is measured 5 cm from the inlet of the induction system during a wide open throttle condition running the vehicle at maximum rated engine rpm.
- The near field engine noise level is measured at a 1 m distance from the vehicle and also at 5 cm from the inlet of induction system during gradual acceleration, from 3000 to 8000 engine rpm, while the vehicle is run on a dynamometer in a semi-anechoic chamber with the exhaust ducted outside the room.

3. Results of standard induction system

The noise levels are shown below for the testing conditions described in the above section. Figure 2 shows the noise spectrum for constant speed motoring condition at maximum engine speed. As can be seen from the figure, intake

Table I. Comparison of pass-by and induction noise levels dB(A) between standard and modified exhaust system designs.

Sl.No	Intake	Exhaust	Engine acceleration		Motoring	
			Pass-by	Inlet	Pass-by	Inlet
Expt-1	Standard	Standard	106	133	94	124
Expt-2	Standard	Modified	106	130	94	123

noise is having a fundamental frequency at 67 Hz and has integer harmonics whose energy content is high until a frequency of 1000 Hz. It also can be noted from the figure that the 1st, 9th and 10th harmonics are very strong compared to other harmonics. However because of A-weighting effects, the first harmonic can be neglected in evaluating induction noise perception. Hence higher acoustic pressure levels of both 9th and 10th harmonics can be assigned as the cause for objectionable induction noise.

Figure 3a shows the overall engine noise measured at a pass-by location — 1 m distance from vehicle — during engine accelerating from 3000 to 8000 rpm. Overall engine noise is represented using a single metric — RMS of spectrum — which has exceeded the internal reference value. Similarly from Figure 3b, induction noise RMS value is 130 dB(A) and it too has exceeded reference value. However it is not very evident at this point , the amount of reduction in induction noise levels that is required to achieve the targeted noise levels at pass-by location.

It is hypothesized that induction system design that results in noise, is independent of engine parameters. Hence engine parameters like ignition timing, compression ratio, combustion chamber size are not varied during this study. Hence only the effect of exhaust system on the induction noise is studied in a great detail. Table I shows the experiments designed in order to understand the influence of exhaust system on induction noise levels. In the first experiment, combination of standard induction and exhaust systems is used for evaluating the induction noise. In the second experiment a modified exhaust system whose mean (r.m.s) Transmission Loss (TL) is higher by 10 dB — highest TL that can be achieved with the given layout — along with the standard induction system is used for evaluating the induction noise. As can be seen from the results, modified muffler has reduced inlet noise levels by 1 dB(A) during constant speed motoring test done at maximum rated rpm. It also reduced inlet noise levels by 3 dB(A) during engine acceleration condition without causing any noise reduction at pass-by location. Hence the modified muffler is unlikely to give any subjectively perceivable reduction in induction noise levels. It can be concluded from these experiments that induction system design alone is responsible for higher induction noise levels.

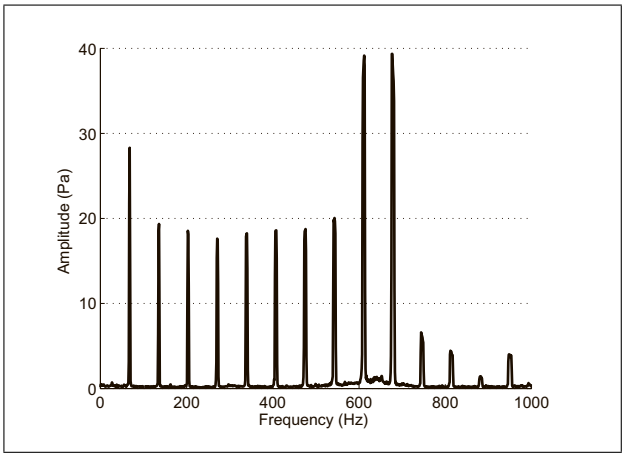

Figure 2. Noise spectrum at inlet during constant speed motoring condition.

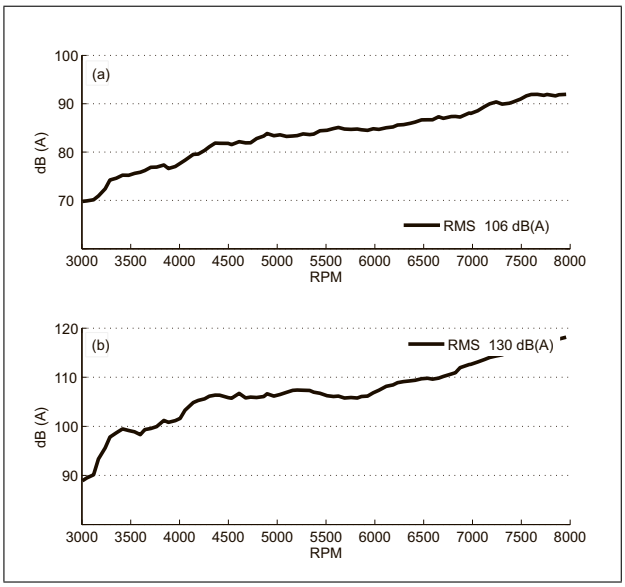

Figure 3. Overall noise measured at pass-by location and at inlet during acceleration condition.

3.1. Experimental Transmission Loss evaluation

The transmission loss of a system can be evaluated by different experimental methods namely decomposition method , two-load method and two-source method as suggested by Tao *et al.* [11]. In the present work, two-load method as shown in Figure 4 is used –as it is relatively easier– to obtain four poles of the system. The transmission loss of the system is calculated using these four-poles as suggested by Munjal [12].

The transfer matrix elements of the entire induction box (with and without the Helmholtz resonator) are determined by changing the loads at the termination, induction snorkel, thus generating four equations with four unknowns. In order to avoid unstable results, two completely different end conditions are used. In test 1 (load 1), the

terminal load is the radiation impedance at the snorkel ($Z_a = R + jX$). In test 2 (load 2), the terminal load is rigid and is completely reflective ($Z_b = R$). The simultaneous solution of these four equations are given by Tao *et al.* as

$$A = \frac{\Delta_{34}(H_{32a}H_{34a} - H_{32b}H_{34a}) + D_{34}(H_{32b} - H_{32a})}{\Delta_{34}(H_{34b} - H_{34a})}, \quad (1)$$

$$B = \frac{B_{34}(H_{32a} - H_{32b})}{\Delta_{34}(H_{34b} - H_{34a})}, \quad (2)$$

$$C = \frac{(H_{31a} - A_{12}H_{32a})(\Delta_{34}H_{34b} - D_{34}) - (H_{31b} - A_{12}H_{32b})}{B_{12}\Delta_{34}(H_{34b} - H_{34a})}, \quad (3)$$

$$D = \frac{(H_{31a} - H_{31b}) + A_{12}(H_{32b} - H_{32a})}{B_{12}\Delta_{34}(H_{34b} - H_{34a})} B_{34}, \quad (4)$$

where H_{ij} is transfer function,

$$H_{ij} = \frac{p_i}{p_j}, \quad (5)$$

where $\Delta_{34} = A_{34}D_{34} - B_{34}C_{34}$.

The transmission loss of system can be calculated using four-poles obtained by equations (1)–(4) as

$$\mathrm{TL} = 20\log_{10}\left\{\frac{1}{2}\left|A + \frac{B}{\rho c} + \rho c\, C + D\right|\right\} + 10\log_{10}\frac{S_i}{S_0}. \quad (6)$$

The transmission loss is calculated as per Equation (6) for two different induction box conditions (with and without resonator) and represented in Figure 5. It can be seen from the figure that ripples exist in the frequency range of 100–200 Hz. This phenomenon is due to the end terminal loads as $ka \ll 0.5$ in this frequency range.

The lower transmission loss of induction system can potentially result in higher induction noise levels. Figure 5 shows the transmission loss superposed with the spectral composition of induction noise in the same plot. It can be seen from the results that transmission loss has a trough and is very low in the frequency range of 500 to 650 Hz and has highest peak at 830 Hz. Hence, it is possible to conclude that the trough in the transmission loss curve is the reason for higher induction noise levels resulting in poor sound perception. A Helmholtz resonator is developed to address the trough in transmission loss curve, as there is no freedom to change induction system since it is already frozen for engine performance factors.

4. Concept resonator design

It is seen from Figure 5, highest induction noise levels are recorded for 9th and 10th harmonics. Hence a resonator design that is aimed to increase transmission loss in this frequency range is arrived at based on a Matlab program. A Matlab program is developed to maximize the transmission loss in the frequency range of 500 to 650 Hz. Resonator tuning frequency [13] is calculated based on the equation

$$f = \frac{c}{2\pi}\sqrt{\frac{A}{V\,l'}}, \quad (7)$$

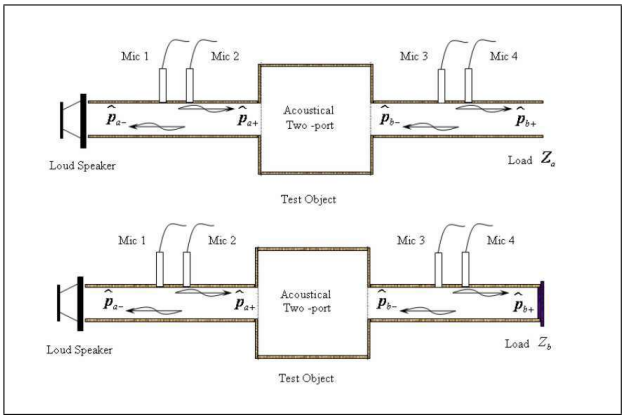

Figure 4. Schematic diagram of Two load method.

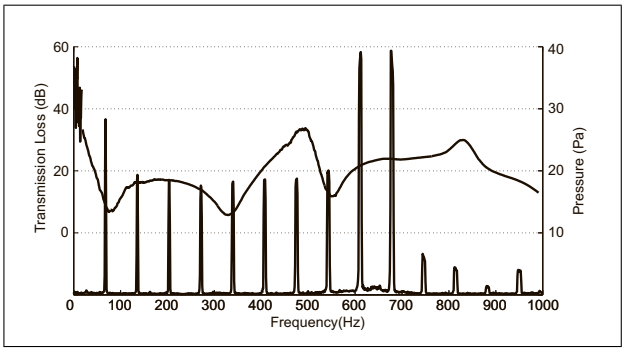

Figure 5. Comparison of transmission loss and induction noise spectral content measured in constant speed motoring.

where f is frequency, A, V and l are neck cross-sectional area, volume and corrected neck length ($l = l' + 1.45r$) of resonator while c is speed of sound in air.

The impedance of a Helmholtz resonator, neglecting damping effects, is denoted [12] as

$$z = \frac{-j\rho c^2}{\omega V} + \frac{j\rho\omega l}{A}.$$

In the above equation, the first term represents elastic impedance and second term represents inertial impedance of resonator. The state variables pressure and mass velocity at the upstream and downstream of the tube outlet fitted with a resonator is given as

$$\begin{bmatrix} p_{out} \\ v_{out} \end{bmatrix} = \begin{bmatrix} 1 & 0 \\ \frac{1}{z} & 1 \end{bmatrix} \begin{bmatrix} p_{in} \\ v_{in} \end{bmatrix}.$$

The transmission loss of resonator can be calculated after obtaining the four poles from transfer matrix [12] as

$$\mathrm{TL} = 10\log_{10}\left\{1 + \left(\frac{\frac{c}{2A_{out}}}{\frac{\omega l}{A} - \frac{c^2}{\omega V}}\right)^2\right\}.$$

The transmission loss for a typical combination of volume, area and length, is shown in Figure 6.

There can be different combinations of area, volume and length that can be used to tune to the same resonator

Table II. Comparison of Transmission Loss peak amplitude (only resonator) for different volume, area and length combinations.

Sl.No	Volume(cc)	Area (mm^2)	Length(mm)	Frequency(Hz)	TL peak(dB)
1	119	531	33	629	80
2	130	177	10	631	78
3	137	284	15	636	87
4	156	531	25	631	83
5	160	573	26	635	104
6	180	491	20	631	87
7	200	491	18	631	86

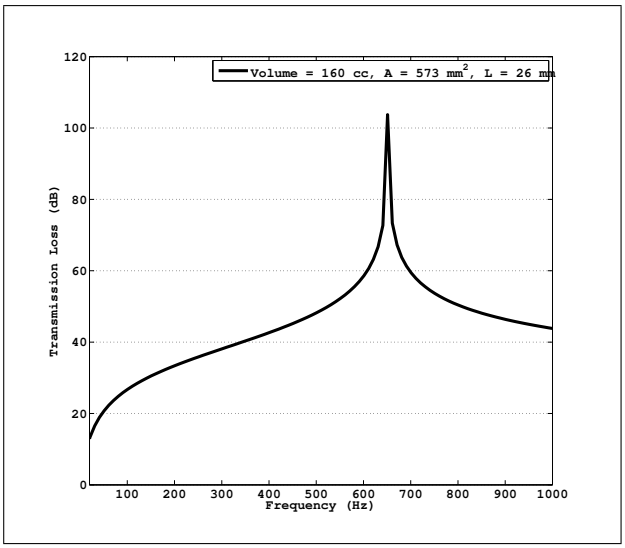

Figure 6. Transmission loss of resonator.

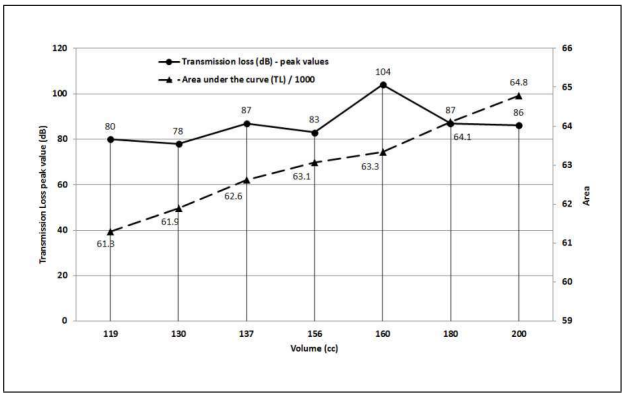

Figure 7. Comparison of (only resonator) transmission loss.

frequency as shown in Equation (7). However the effect of these combinations on the resonator performance (amplitude reduction at tuned frequency) can be significantly different as the propagation of acoustic wave in resonator cavity is dependent on the relative magnitudes of the wavelength, the neck and duct diameters [14]. The amplitude reduction attained at resonator-tuned frequency is dependent on l/d ratio of acoustic volume. Hence the following combinations (shown in Table II) of volume, area and length are studied to understand the effect on resonator's peak amplitude.

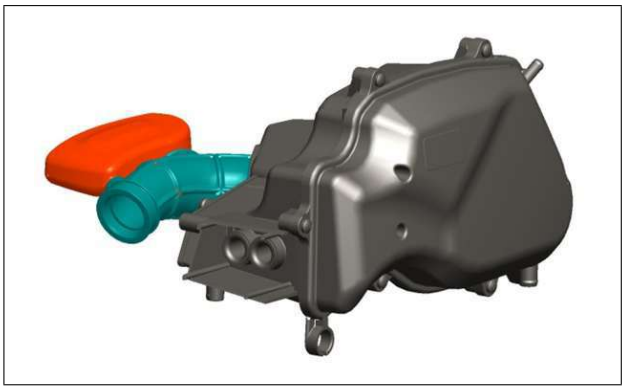

Figure 8. Induction system fitted with concept resonator.

As can be seen from Table II, transmission loss has highest peak value of 104 dB for combination of 160 cc volume, 573 mm^2 area and 26 mm neck length. It may be inferred from the result that there is highest impedance mismatch for this configuration. As can be seen from the Figure 7, transmission loss has highest peak for 5th combination shown in Table II along with higher area under curve compared to other combinations — also one of crucial parameters for reduction of noise levels at other frequencies other than the resonator-tuned-frequency. Hence this combination is selected to design a resonator that can potentially attenuate noise levels for 10th harmonic in the frequency of 500-650 Hz.

4.1. Experimental comparison of Transmission Loss

The transmission loss of modified induction system that is fitted with resonator is evaluated and the same is compared with that of standard system. The experimental procedure adopted for evaluating the transmission loss is the two load method that is described by Tao *et al.* [11]. It may be noted that resonator is fitted on the tube outlet i.e on cleaner side of induction system and also parallel to ground condition. This arrangement scheme results in benefits like dust entry prevention from the outside environment and also to prevent fuel accumulation — that flows in reverse direction from carburettor in case of negative back-pressure. This concept resonator made up of mild steel with thickness of 1.5 mm is shown pictorially in Figure 8. Figure 9 shows comparison of transmission loss curves for both standard and modified (fitted with resonator) induction system in the frequency range of interest. As can be seen from the

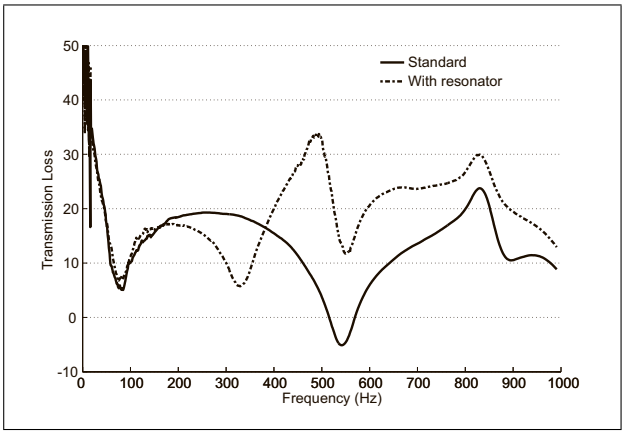

Figure 9. Comparison of transmission loss for standard induction system and with induction system fitted with resonator.

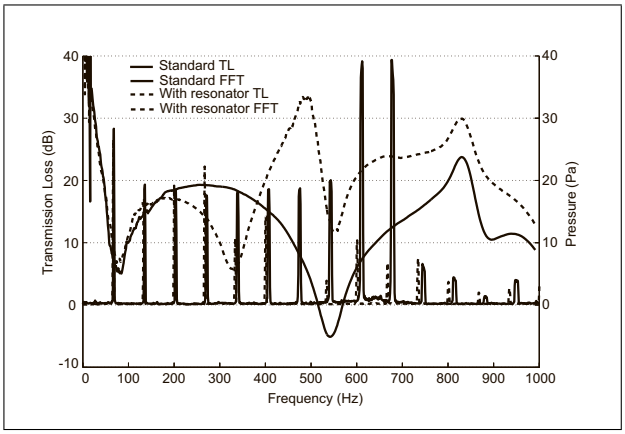

Figure 10. Comparison of transmission loss and sound pressure levels for standard induction system and with resonator-fitted system.

figure that the modified induction system transmission loss is higher compared to that of standard induction system. It can be inferred from this result that peaks in the frequency range of 400 to 650 Hz in design equipped with resonator against troughs of standard induction system is result of appropriate “tuning" of resonator and hence this design has potential to suppress noise content in the troubling frequency range.

The associated space constraint resulted in the resonator neck to be eccentric relative to the resonator chamber (cavity). This configuration allows higher order modes to get excited and that could be one of the reasons, along with the mean flow effect which has been neglected in the study, for the frequency shift in actual experimental results compared to the prediction [15]. The acoustic coupling between the induction box and the resonator has not been taken into account at the time of tuning and it has been dealt only during experimentation.

5. Results of induction system with resonator

The induction system along with a concept metallic resonator is assembled on an engine. This vehicle is evaluated for noise reduction under similar testing conditions that are described in section 2. Figure 10 shows noise measurement results along with transmission loss of standard and resonator fitted designs in constant speed motoring condition. It can be noted from the figure that there is a huge reduction of sound pressure levels in the frequency range of 400 to 650 Hz in case of resonator-fitted design as there an increase in transmission loss compared to standard sound pressure levels. There is 12 dB(A) reduction in rms sound pressure levels compared to standard induction system and 8 dB(A) sound pressure level reduction at tuned frequency. Figure 11a and Figure 11b show sound pressure levels measured at inlet and also at pass-by location during engine acceleration condition for standard and resonator fitted induction systems. It may be noted from the figure that there is 6 dB(A) noise level reduction at inlet and 2 dB(A) noise level reduction at pass-by location.

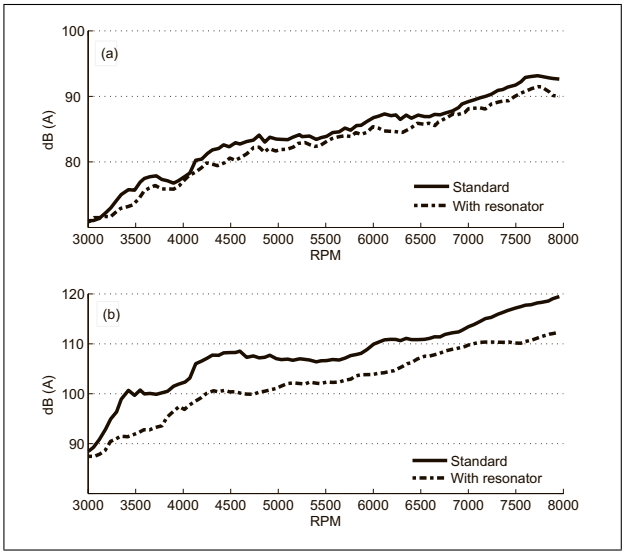

Figure 11. Comparison of sound pressure levels at pass-by and inlet location between standard and resonator-fitted system.

The concept resonator is made of mild steel with 1.5 mm wall thickness. The advantage of making the concept resonator with mild steel is that it can be readily fabricated and its effectiveness can be quickly evaluated. However for mass production, resonator part needs to be compact and light weight because of weight and cost constraints. Hence a resonator made of polypropylene with wall thickness of 1.5 mm — same thickness of concept metallic resonator — is pursued for further noise reduction trials.

The natural frequency of this resonator with 1.5 mm thickness in free-free boundary condition is computed using finite element analysis software [16] (NASTRAN), taking into account mass loading effects of accelerometer in experiments. Design iterations are carried out by varying wall thickness of resonator to increase first fundamental frequency of resonator.

The natural frequency of polypropylene resonator with 1.5 mm wall thickness in free-free boundary condition is obtained using an impact hammer test and results are

Table III. Comparison of polypropylene prototype resonator structural frequencies with different wall thicknesses.

Mode	Design -1 (1.5 mm thk)	Design -1 (with mass loading)	Design -2 (2.5 mm thk)	Design -2 (with mass loading)
1	660	449	1010	897
2	848	718	1351	1169
3	1097	1056	1526	1447
4	1131	1116	1599	1558

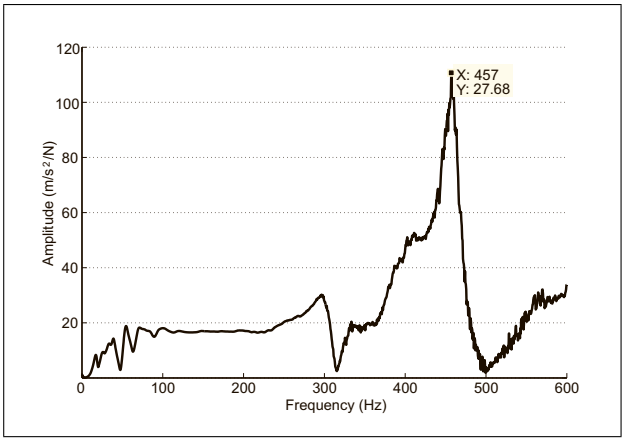

Figure 12. Experimental natural frequency of polypropylene resonator with 1.5 mm wall thickness.

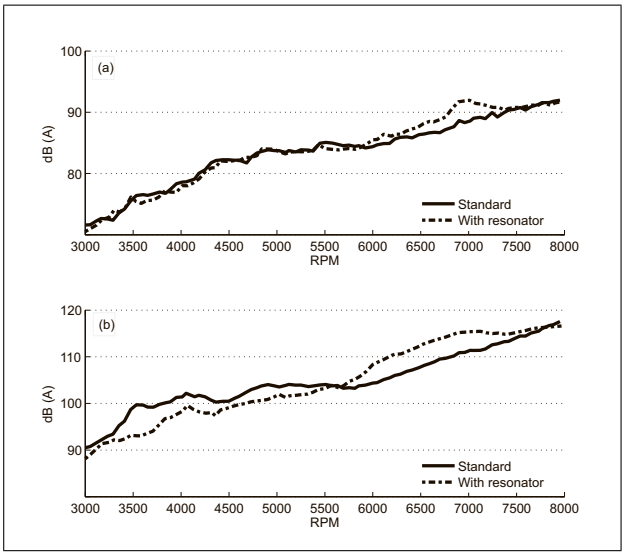

Figure 13. Comparison of sound pressure levels at pass-by and inlet location between standard and polypropylene resonator with 1.5 mm thickness fitted system.

shown in Figure 12. The first fundamental frequency obtained using simulation considering mass loading effects is 449 Hz and it matches very closely with experimental frequency of 457 Hz. Hence it can be inferred from Table III that the polypropylene resonator with 1.5 mm wall thickness has its structural frequency (660 Hz) coinciding with its acoustic frequency (635 Hz) of resonator. This polypropylene resonator is fitted on the tube outlet of the induction system and noise testing is carried out under similar testing conditions as described in section 2. It can be seen from the results that in constant speed motoring test there is 8 dB(A) reduction in rms sound pressure levels in the entire frequency range, along with 11 dB(A) sound pressure level reduction at resonator-tuned frequency. However it can be seen from Figures 13a and 13b that at pass-by location and at inlet during engine acceleration condition there is no significant reduction in noise levels. This phenomenon can be explained from the fact that although there is more reduction at tuned frequency in case of structural frequency coincides with its acoustic frequency, it is much weaker at other frequencies and emanates more noise levels thus making resonator less effective. In order to overcome the above said problem, thickness of polypropylene resonator is increased to 2.5 mm and found from simulation that first fundamental frequency is at 1010 Hz. As this structural natural frequency is well separated from acoustic resonator frequency, it can be considered as a “stiff" resonator in the frequency range of interest. It can be seen from the Figure 14a and 14b that this stiff resonator is reducing noise levels both at inlet and at pass-by location by 7 dB(A) and 2 dB(A) respectively. This improvement in noise level reduction is very similar to the one obtained with concept metallic resonator. This polypropylene resonator with 2.5 mm wall thickness is fitted on to three different vehicles and objective noise measurements are carried out under similar testing conditions — mainly under engine ac-

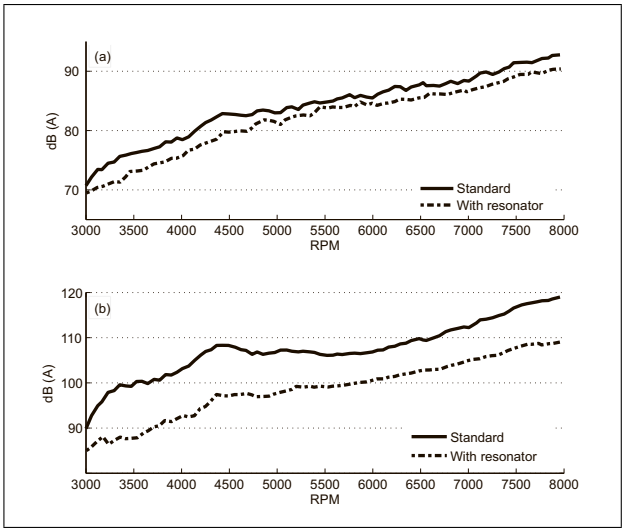

Figure 14. Comparison of sound pressure levels at pass-by and inlet location between standard and polypropylene resonator with 2.5 mm thickness fitted system.

Table IV. Sound Pressure Levels (SPL) reduction with polypropylene resonator with 2.5 mm wall thickness under engine acceleration condition.

	Pass-by	Inlet
Vehicle - 1	2	7
Vehicle - 2	1	5
Vehicle - 3	2	6

Table V. Summary of subjective testing results of polypropylene resonator with 2.5 mm wall thickness.

Sl.No	Subject	With resonator	Standard
1	Subject - 1	6	3
2	Subject - 2	6	3
3	Subject - 3	7	4
4	Subject - 4	7	4
5	Subject - 5	7	5
6	Subject - 6	7	4
	Average	6.7	3.8

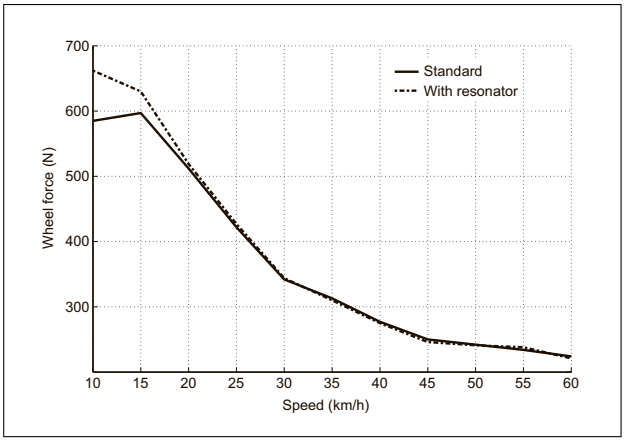

Figure 15. Comparison of wheel force between standard and polypropylene resonator.

celeration conditions. It can be seen from Table IV that this resonator is reducing on an average about 6 dB(A) at the inlet and 2 dB(A) at the pass-by location.

5.1. Subjective testing

The improvement obtained by objective noise measurements needs to be corroborated by subjective testing results. Hence a subjective testing on both standard and polypropylene resonator with 2.5 mm wall thickness is conducted. Six different subjects, who were not informed about the modifications (to eliminate bias in their response), were asked to test the vehicle in customer use and abuse conditions. They were asked to rate induction noise levels on a scale of 0 to 10. Any rating closer to 10 indicates that induction noise levels are lesser and are acceptable ; rating of 5 and below is objectionable for perception. Table V shows the summary of subjective testing results. It can be concluded from these results that resonator fitted induction system is much superior compared to standard induction system. It can also be inferred from the results that 5 dB(A) reduction at inlet will yield on an average three point improvement in subjective rating evaluation.

5.2. Side effect check

The side effects of introducing a resonator in induction system on engine performance is checked at gross level using Wide Open Throttle (WOT) wheel force. Vehicle is mounted on a chassis dynamometer and wheel force is measured in both standard and resonator fitted induction system cases. Figure 15 shows comparison of wheel force between standard and with resonator-fitted configuration. As can be seen from the figure that there is no significant change in wheel force with resonator installation. Hence the proposed modification will not alter engine performance parameters.

6. Conclusions

A comprehensive methodology is developed to design a resonator in case of a two-wheeler to reduce induction noise. It is shown that on an average 5 dB(A) noise reduction at inlet and 2 dB(A) noise reduction at pass-by location was possible using a resonator. This paper also demonstrated the methodology to exploit resonator concept to reduce induction noise that obviates the necessity to re-design entire induction system. The side effects of introducing a resonator in induction system on engine performance are checked at gross level using WOT wheel force and these are found to be negligible. Finally correlation between subjective and objective results is established for induction noise.

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