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Technical Note

The effect of rubber dampers on engine's NVH and thermal performance

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ABSTRACT

Fins as extended surfaces are attached to the internal combustion engine surfaces for enhancing the heat transfer. However, these fins vibrate at various frequencies, which produce undesirable radiated noise. To mitigate this effect, automobile industry inserts rubber dampers between these fins. These rubber dampers reduce the fins' amplitude of vibration and thus reduce the radiated noise from the fin surfaces. Investigations on the effect of rubber dampers on the engine's NVH (Noise–Vibration–Harshness) and thermal performance using numerical (FEM and CFD) and experimental measurement have been presented in this paper. Experiments were conducted in the semi-anechoic chamber on an engine with and without rubber dampers to measure the radiated noise from the fins. It was found that rubber dampers assist in reducing engine high frequency noise signals at higher engine speeds. Modal and harmonic response analysis was carried out on various designs for NVH characteristics improvement. Prototypes of the final design was made and tested for the NVH performance. Computational fluid dynamics (CFD) simulations were performed on engine with and without rubber dampers to investigate the thermal performance. It was found that rubber dampers increase engine temperature by about 10%. Effect of rubbers dampers on the cost and environmental impact has also been discussed. This paper provides a systematic procedure to investigate the effect of rubber dampers and a method to eliminate these dampers from the engines with the same NVH and better thermal performances.

1. Introduction

A recent study by the US Energy Information Administration [1] reveals that industrial sector uses more energy than any other end-user sector, consuming about 50% of the world's total delivered energy. Over the 28-year projection, worldwide industrial energy consumption grows from 184 quadrillion Btu in 2007 to 262 quadrillion Btu in 2035, a 42% increase in the energy consumption. On the other hand, transportation sector uses 20% of the world's total delivered energy. From 2007 to 2035, growth in transportation energy use would account for 87% of the total increase in world liquids consumption. Besides limited resources and high energy needs, rise in the input raw materials cost has forced automobile sector to optimize their product so that the use of associated components/materials can be minimized or completely eliminated. In this paper, an investigation is reported that could bring about huge resources and energy saving; reduce pollution and increase productivity of the automobile industry.

Fig. 1 shows photographs of air-cooled engines of two-wheeler vehicles taken from various motorcycle brands. It can be seen that rubber dampers have been inserted between the fins. Fins are extended surfaces that are used to enhance the heat transfer from the engine surfaces to reduce the metal temperature. Fig. 1a and b shows 2-stroke (2S) engine cylinder head and Fig. 1c and d of 4-stroke (4S) engine cylinder head. Use of rubber dampers is a common practice among automotive industry. Fins on the 2S engines are generally longer and wider than 4S engines as thermal loads on engine is theoretically two times more in the former case. On the other hand, these fins vibrate as engines run at various speeds. Vibrations of these fins produce undesirable noise at various frequencies. Rubber dampers are provided between the fins to reduce the amplitude of fin vibration. The reduced vibration would lead to lower radiated noise for the fins surfaces. Also, these dampers add to the vehicle cost and pollutes environment. Scientific literatures on the effect of these rubber dampers on engine's NVH (noise, vibration and harshness) and thermal performance are sparse. NVH is the study and improving of the noise and vibration characteristics of vehicles [2]. This paper provides a systematic study of NVH and thermal effects of rubber dampers. Furthermore, a design study of the cylinder head has been provided so that the rubber dampers can be eliminated altogether while maintaining the same NVH and better thermal behavior of the engine. Horizontal engine cylinder head shown in Fig. 1a was selected in this study. Though this study is for a specific type and design of the cylinder head, the knowledge and methodology discussed in this paper

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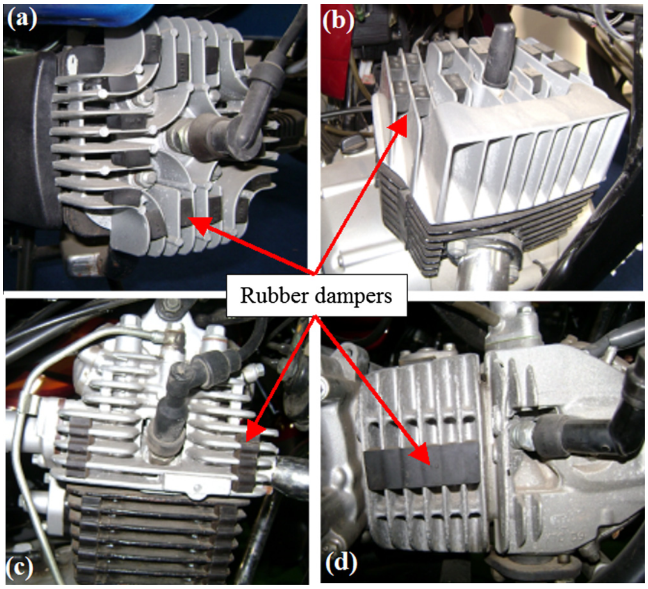

Fig. 1. Engines with rubber dampers between the fins from four different automobile manufacturers.

can be applied to any type of engine design containing fins with rubber dampers. Such studies are generally not reported in the scientific literature as investigations on the effect of rubber dampers is not considered as a part of the engine research and decision to put rubber dampers on the engine is taken during the last stages of the engine development. Once it is put on the engine and the engine is integrated with the vehicle, further investigation on the effect of dampers is hardly performed. The significance of this study will become clear from Section 4 in which the advantages of engine design without rubber dampers have been discussed. Before we present experimental noise measurement and thermal study, a brief theoretical basis is provided on the fact that noise and thermal performance of fins are related with each other.

1.1. Fundamental theories

Generally, fins are independently discussed either from the thermal point of view [3–6] or vibration point of view as a cantilever beam [7]. However, they are related when used on vibrating surface for heat transfer [3]. Total heat transfer from the heated surfaces is given by expression [5], $Q = A_c L(\eta_f + A_b/NA_f)Nh\theta_b$, where A_f is fin surface area; A_b, the exposed base surface area; A_c, the fin cross sectional area; N, the number of fins; T_b, base temperature; h, the heat transfer coefficient; η_f, efficiency of a single fin; $\theta_b = T_b - T_0$; T_b is the base temperature. Clearly, heat transfer from the surface directly depends on the fin surface area and hence fin length. The number fins depends on the available surface area and limited by heat transfer coefficient h. With large number of fins on a given surface area, the ratio A_b/NA_f decreases which in turn reduces h. In most cases where available surface area is small and rate of heat generation is high (e.g. in 2S engines), the most preferred method to enhance the heat transfer is by increasing the length of the fins.

On the other hand, lengthy fins have their own disadvantages specially when used on vibrating surfaces like IC engine. These fins vibrate at various frequencies, which produce undesirable noise. The high-frequency components (2000 Hz and above) of the cylinder pressure as a result of combustion forces are the main causes of excitation of the engine structure [8]. The amplitude of vibration, X, considering the fin as a cantilever beam fixed at one end and whole mass concentrated at other end, vary as [7], $X \sim L^3$. With increase in fin length, vibration amplitude increases (an undesirable feature) and fin base temperature decreases (a desirable feature). Putting rubber dampers between the fins makes a compromise between these two features. We show that these dampers defeats the purpose of lengthy fines i.e. they increase the engine temperature rather than reducing it. Furthermore, use of rubber dampers for NVH purpose is also disadvantageous in long run (Section 4).

The paper is organized into two main sections: one section on NVH study and other section on thermal study. In Section 2, the experimental noise measurement and finite element analysis (FEA) has been discussed. Various design modifications of the cylinder head, their modal and harmonic response analysis, and experimental verification in the improvement of noise radiated from the cylinder head is presented in this section. In Section 3, the computational fluid dynamics (CFD) modeling of engine with rubber dampers and the effect on the engine temperatures with and without rubber dampers has been discussed. A discussion on the cost and environmental impact of rubber dampers has been provided in Section 4. Conclusions are presented in Section 5.

2. Experiments and finite element analysis

2.1. Experimental noise measurement

Effect of rubber dampers on the radiated noise from the engine was studied in a semi-anechoic chamber (Fig. 2). The length and width of chamber was approximately four and three times of vehicle width respectively. Two microphones were placed at half-meter distance from the engine. Using data acquisition system and data analysis software LMS-Pimento, noise levels are measured during vehicles gradual acceleration. Accelerometer mounted on the engine was used to measure the engine acceleration levels. Quantification of noise levels with and without rubber dampers on the cylinder head was done systematically. 4 dampers are removed each time and the engine was run at various speeds from 0 to 6000 RPM. Hence the radiated noise was measured with 16, 12, 8, 4 and 0 (no dampers) dampers. Fig. 3 shows the noise (dB) measured at various engine speeds. It is noticed that noise radiated by the engine fins is lowest when all the dampers are present on the cylinder head throughout the engine speed range. As the number of rubber dampers are reduced, noise level increases. Fins radiate an average of 3–4 dB higher noise level when no dampers are used. To make the effect of rubber dampers more clear, the radiated noise during gradual acceleration with all and no rubber dampers is shown in Fig. 4. Clearly, rubber dampers affect the noise level and an average of 4 dB excess noise is radiated when no dampers are present. Therefore, root cause of the excess noise needs further investigation. From Fig. 4 it not clear about the range of frequencies in which the radiated noise is dominant. The noise spectrum from the engine is shown in the form of Campbell diagram in 1/3 octave band (Fig. 5), which provides the overview of the frequency of the noise signals as function of engine speed. Noise signals are compared with dampers (Fig. 5a) and without dampers (Fig. 5b) between the fins. From Fig. 5(a), it can be noticed that strong sound pressure levels are located in the range of 1000–1600 Hz at higher engine rpm (range 5000–6000 rpm) and these noise signals intensified further without rubber dampers (Fig. 5b) to low engine rpm. Further, the noise levels increased to higher frequencies range (2000–4000 Hz) in the engine without rubber dampers. Hence, the overall effect of rubber dampers is that it helps in reducing engine high frequency noise signals at higher engine speeds.

The next step is to identify the root cause of the problem that contributes to the increase in radiated noise without rubber dampers. Engineers have followed essentially two major approaches to

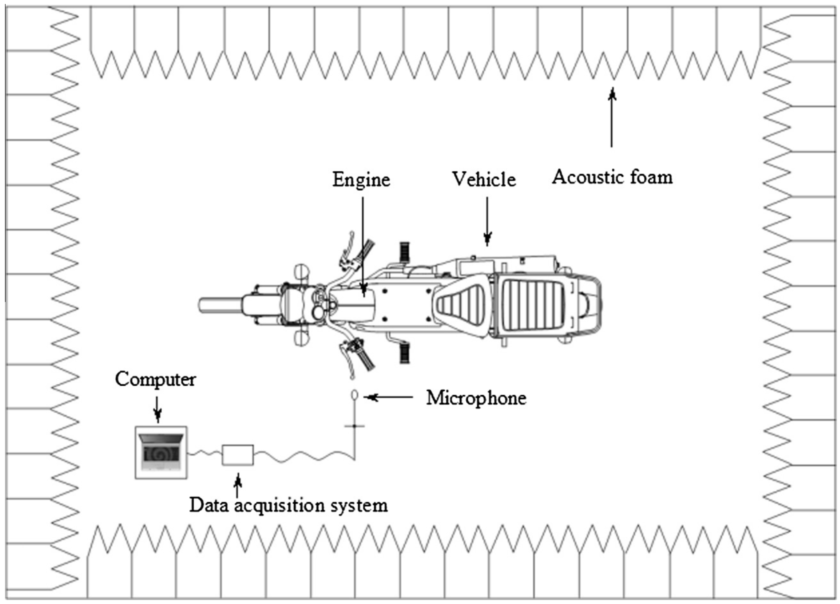

Fig. 2. Schematic diagram showing vehicle in the semi-anechoic chamber for noise measurements.

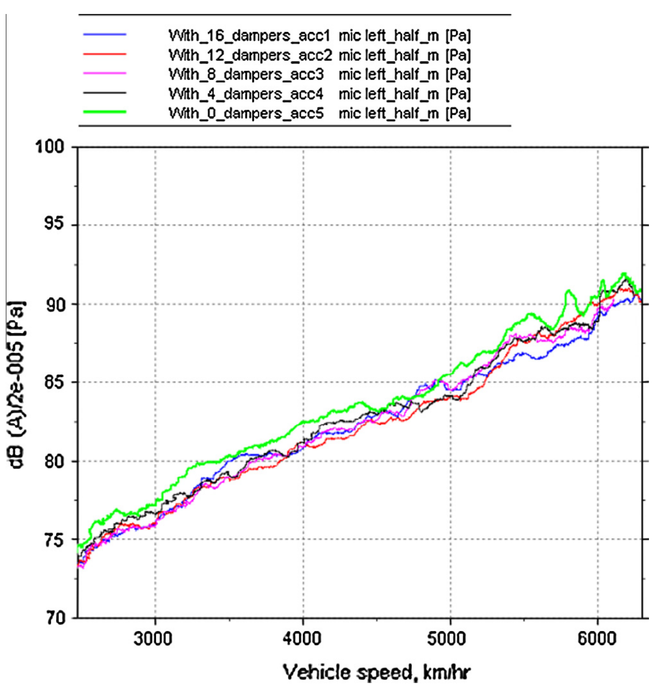

Fig. 3. Noise levels variations with different dampers on the cylinder head during gradual acceleration.

solve structural noise problems [9–11], namely: (i) find ways of reducing the sound power by passive means such as re-design or modification of the structure and (ii) use active control approaches. Although active control solutions have a strong potential in the future, especially for low-frequency noise, passive control still has to be considered because it addresses the problem at the source generally with a low cost and is especially successful for high-frequency noise. Furthermore, according to previous observations [12–14], reduction in ability of the structure to respond to excitation is the only possible way of obtaining any significant reduction at source. It is observed that fins get excited when rubber dampers are removed. Hence, the following section looks into the structural modifications of engine cylinder head and its effect on the radiated noise.

2.2. Finite element analysis: Modal and harmonic response analysis

Modal analysis is used in the structural mechanics to determine the natural mode shapes and frequencies of a structure during free vibration. Modal analysis gives direct insight into the root cause of the vibration problems. Most often the desired modes are the lowest frequencies because they can be the most prominent modes at which the object will vibrate, dominating all the higher frequency modes. Modes are inherent properties of a structure, and are determined by the material properties (mass, damping, and stiffness), and boundary conditions of the structure. If the material properties, structural design or the boundary conditions of a structure change, its modes will change. In modal analysis, damping and external force are neglected.

A finite element model of the cylinder head was prepared. All the modal analysis in ANSYS software was run without rubber dampers. Modeling rubber would involve non-linear analysis [15], which requires huge computational resources and time. Furthermore, rubber dampers does not effect the natural frequencies significantly as it is evident by the relation, $f_d = \sqrt{1 - \xi^2} f_n$ where f_d is the damped natural frequency, f_n is the undamped natural frequency and ζ is the damping ratio. For example, with 0.1 damping ratio, the damped natural frequency is only 1% less than the undamped frequency. The value of damping ratio of rubber is 0.05 [16], which will not alter the natural frequencies of the cylinder head significantly. Base of cylinder head was fixed for all degrees of freedom in modal analysis. Properties of aluminum are used. The value Young's modulus of elasticity equal to 73,000 MPa and density equal to 2790 kg/m^3 was used. Modal analysis was followed by harmonic response analysis using mode superposition method for each design modification. In harmonic analysis, mode superposition method with excitation force of 3 times of gravity in the plane of engine vibration was applied to the model. This acceleration level was obtained from the experimental measurement. Damping ratio of 2% was used. Reduction in amplitude of vibration of fins was investigated for each design modification.

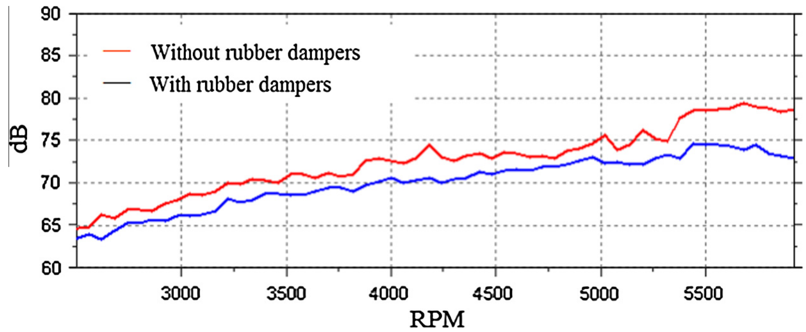

Fig. 4. Comparison of noise radiated from the engines with and without rubber dampers on the cylinder head at 3150 Hz.

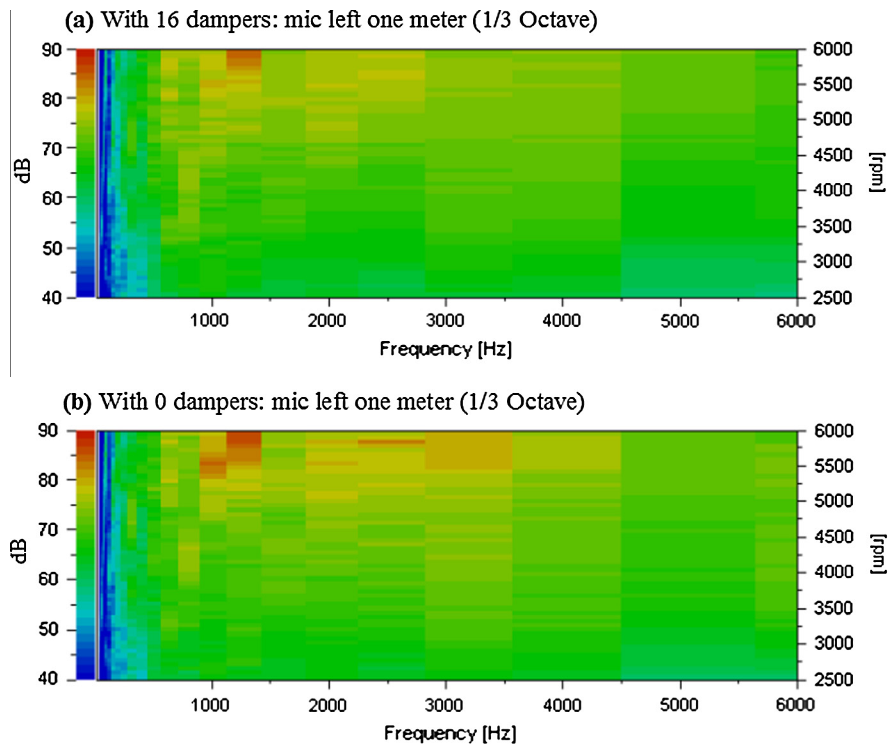

Fig. 5. Campbell diagram showing noise levels with (a) rubber dampers and (b) without rubber dampers.

3. Results and discussion

3.1. Modal analysis

Fig. 6 (left panel) shows the various design modifications investigated and figures on right panel shows the corresponding 5th mode of vibrations of the cylinder head obtained from the modal analysis. The subsequent design modifications were based on the results of the previous designs. Values of the first & fifth natural frequencies and their percentage increase from the base design (design 1) are mentioned on the left and on the right side of the figure. First 15 natural frequencies are extracted for each design (see Table 1). The first natural frequency of design 1 is 1470 Hz. Since the first mode frequency is too low, the first mode frequency of the subsequent design is targeted to increase it above 4000 Hz. Human ear is less sensitive to frequencies above 4000 Hz. From the vibration pattern at this frequency it is seen that the fins located at the center exhibits the local modes of bending motion. These local modes of vibration could radiate higher noise levels at resonance. One interesting observation is that all the 15 modes of vibration of this design exhibit local modes. In design 2, ribs are now connected to the top of the fins where local modes were seen earlier. The first natural frequency has now increased to 2969 Hz, which is about 100% improvement from the previous design. Fins in this design have twisting modes about the center of the cylinder head. In design 3, additional ribs were added to the fins to avoid local twisting modes of vibration. The first mode natural frequency increased to 3482 Hz, an improvement of 513 Hz from design 2 and an improvement of 2009 Hz ($\sim$136%) from design 1. Interestingly, the modes of vibration remain the same as that of design 2. However, the number of fins participating in the vibration have reduced.

It is to be mentioned here that prototypes of design 3 were made and tested on the vehicle in the semi-anechoic chamber. Though the noise radiated by this design of the cylinder head without rubber dampers was acceptable, however, due to manufacturing constraint in mass production, design 4 was conceived, studied and tested for NVH and thermal performance. In design 4, ribs are now connected from the bottom of the fins to the top of the fins. The first mode natural frequency is 4134 Hz, which is above the

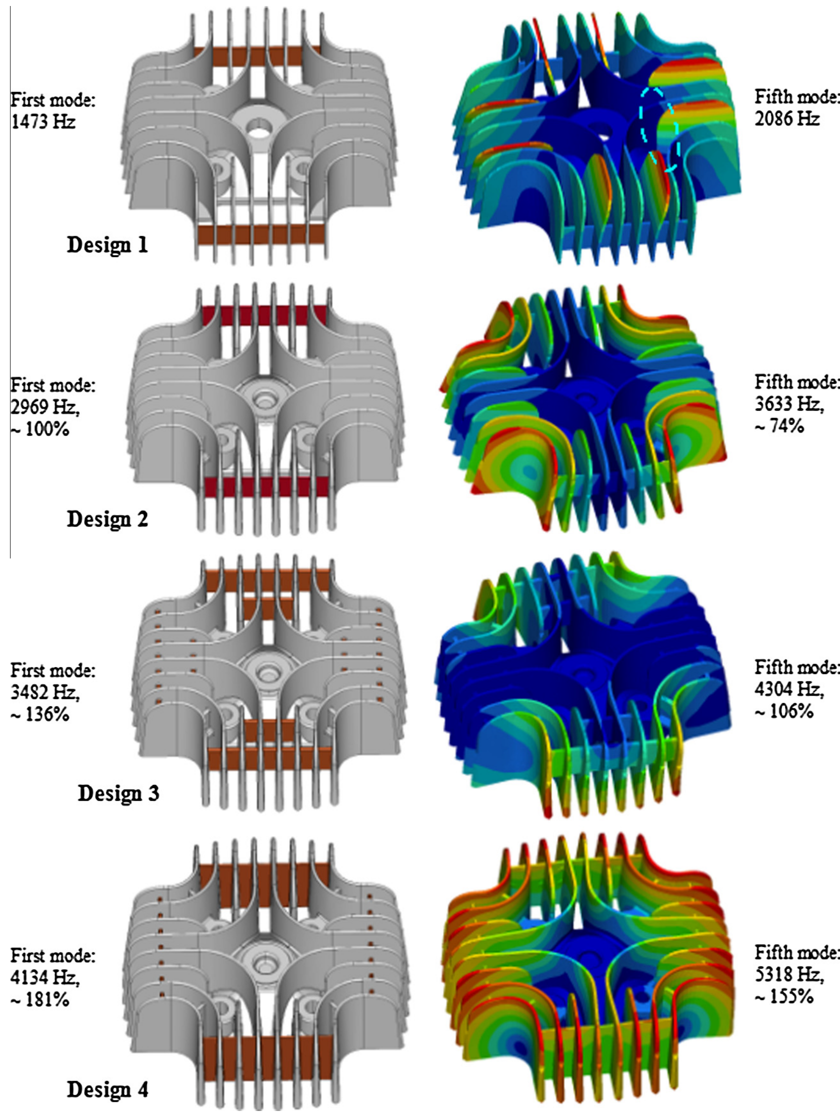

Fig. 6. Various design modifications (left panel) and corresponding mode shapes of fifth natural frequency from the modal analysis. Natural frequencies of first and fifth modes are mentioned on the left and right hand side of the figure. The dot-marked fin was used to study the frequency response due to external excitation.

Table 1
Natural frequencies of the cylinder head design 1–4.

Mode no.	Design 1 (Hz)	Design 2 (Hz)	Design 3 (Hz)	Design 4 (Hz)
1	1470.3	2968.9	3481.6	4134.0
2	1652.2	3123.9	3973.1	4203.6
3	1654.0	3128.3	4008.5	4297.7
4	1992.0	3130.7	4012.2	4302.7
5	2086.1	3632.6	4304.2	5317.8
6	2348.4	3962.4	4313.4	5437.3
7	2355.8	3969.9	4547.2	5726.1
8	2425.6	4107.2	4723.2	5742.2
9	2568.1	4158.5	5003.6	6127.4
10	2571.7	4160.9	5048.6	6134.5
11	2579.1	4240.8	5052.5	6477.8
12	2588.8	4555.4	5260.1	7267.6
13	2591.5	5624.2	6655.3	9413.7
14	2800.7	5632.9	6659.4	9790.1
15	2809.8	5721.8	6675.4	9796.5

targeted frequency of the first normal mode of design 1. All the first 12 mode shapes of vibration consist of bending and twisting components in which all the fins participate in the vibration. Only at the 13th natural frequency of 9414 Hz in design 4, the local fin mode vibrations were observed. It is to be noticed that the gain in the first natural frequency from design 1 to 4 is higher than the gain in higher modes. For example, a 181% gain in the first natural frequency is obtained from design 1 to 4 where as only 155% is obtained in 5th natural frequency.

All the 15 natural frequencies obtained from the modal analysis of design 1 to 4 are listed in Table 1. A clear trend in the increase of natural frequency from design 1 to 4 can be seen across the higher modes. It is to be mentioned here that the substantial increase in the natural frequencies is not at the expense of increasing the thickness of the fins but by proper addition of ribs between the fins. This demonstrates that the effects of ribs are like extra mass to the lower order vibrations but an increase in stiffness to the higher modes of vibration [17].

3.2. Harmonic response analysis

Fig. 7 shows the frequency response plot demonstrating the response (displacement, δ) amplitude of a particular fin (marked in Fig. 6) over the frequency range. A node on top and another at bottom part of the fin was selected to extract the amplitudes of vibration over the entire frequency range of the applied force. The same nodes were selected in the study of other design modifications. Fig. 7a shows the response amplitude of the top part of the fin for design 1 to 4. Following observations are made: (a) the frequencies at which first resonance occurred increased from design 1 to 4. These are 1652 Hz, 3131 Hz, 4012 Hz and 4298 Hz for design 1 to 4 respectively, and (b) significant decrease in the vibration amplitude at resonance from design 1 to design 4. For example, in design 1, $\delta = 1.069 \times 10^{-2}$ mm at 1652 Hz, and in design 4, $\delta = 2.881 \times 10^{-3}$ mm at 4298 Hz; a reduction of 73% in amplitude is achieved. Our target was to change the design of the cylinder head to increase its natural frequency beyond 4000 Hz with lower amplitude of vibration, which is achieved in design 4. The above discussions demonstrate that higher harmonics, which contain information on the stiffness of the design, yield relatively low fin responses. It is to be mentioned here that the first resonant frequency obtained from the simulations is about 100 Hz higher than that observed in experiments. One reason being the FEA model in which only the cylinder head was modeled with its based fixed. As a result, the FEA model adds additional stiffness due to the boundary condition. However, in experiment the whole engine vibrates. Nevertheless, since we are interested in understanding the local modes of fin vibration and its frequencies, the model is acceptable.

Fig. 7b shows the frequency response plot of the bottom part of the same fin as mentioned above. Few interesting observations are made from the plot: (a) δ is an order of magnitude smaller than that observed on the top of the fin for all designs in the input frequency range. This is due to the fact that bottom part of the fin is thicker than the top of the fin, (b) the resonance frequency is same as that observed for the top part of the fin, (c) the sum of δ at top and bottom part of the fin for design 4 is about two orders of magnitude less than design 1 across the frequency range. For example, for design 1, $\delta_{top+bot} = 1.082 \times 10^{-2}$ mm and for design 4 it is $\delta_{top+bot} = 1.598 \times 10^{-4}$ mm. The physical interpretation of this is that a significant part of vibration energy (kinetic and potential energy) of the system in design 1 is absorbed by fins alone due to local modes of vibration (see Fig. 6) whereas in design 4, all fins contribute to the absorption of energy due to global mode of vibration and thus, resulting in low amplitudes of vibration. The next stage of investigation was prototypes development of design 4, experimental noise measurements and comparison with design 1 of the cylinder head.

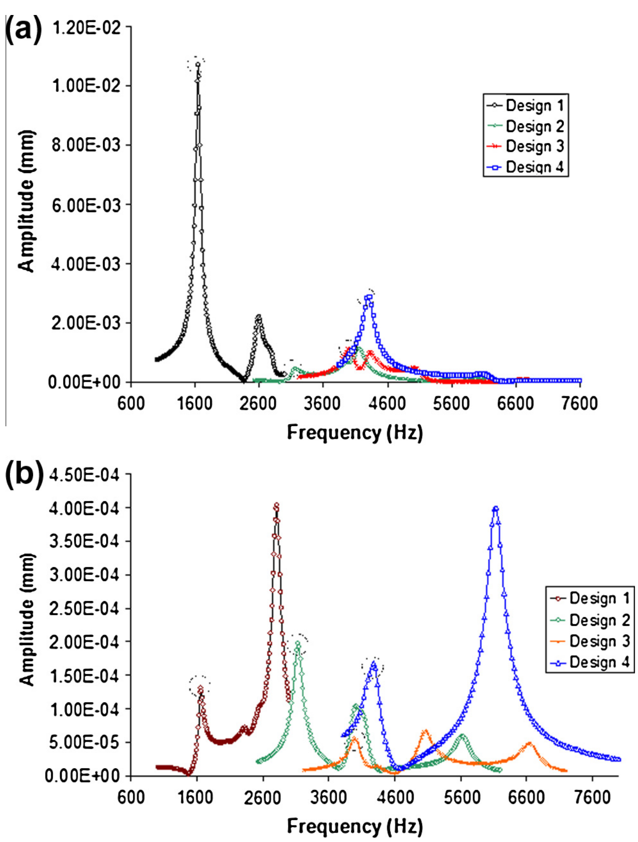

Fig. 7. Frequency response plot of the (a) top and (b) bottom part of the fin across the input frequency range. The marked circle shows the pick amplitude of the first resonant frequency for all four designs.

3.3. Experimental noise measurement and comparison

The prototypes of the cylinder head (design 4) were made and radiated sound was measured without rubber dampers in semi-anechoic chamber using the same procedure as described earlier. The total time taken from the simulation stage (all designs) till the prototype development was approximately eight months. The noise radiated from the existing design (design 1) of cylinder head was also measured. Fig. 8 shows the sound pressure levels radiated from the cylinder head of design 1 (with rubber dampers) and design 4 (without rubber dampers) measured during the gradual acceleration. It can be noticed that the radiated noise levels across the engine speed are of same order and magnitude in both the designs. Fig. 9 shows the noise levels in Campbell diagram radiated from the design 1 and design 4 of the cylinder head. A careful comparison reveals that both the designs radiate same noise levels across the frequency range and engine speed. It signifies that NVH performance of design 4 without rubber dampers is same as that of design 1 with rubber dampers. In the following section the effect of rubber dampers on the thermal behavior is investigated using computational fluid mechanics (CFD) simulations.

4. Computational fluid dynamics analysis

4.1. The CFD model

Rubber dampers affect the free flow of ambient air around the engine surfaces and this in turn can increase the engine temperature. Data on the effect of these dampers on engine thermal performance is rare or not available in the scientific literature. The effect of dampers on the engine temperatures using CFD simulation is investigated. A conjugate heat transfer model was developed to study both the fluid and solid domain. The computational domain is shown in Fig. 10a. The engine assembly with head–block with rubber dampers between the fins is placed in a wind tunnel for the external aerodynamics study. The domain size of the wind tunnel is as follows. Inlet: 2× engine length from the front of the engine; outlet: 4× engine length from the rear of the engine; sides: 3× engine width from engine sides; ceiling: 3× engine height above engine. In this way effect of side walls are minimized [18]. Outlet is 2× longer than inlet as wake regions generated behind the engine needs to be captured to avoid any convergence problem during the computation. The polyhedral cell volume of the engine with rubber dampers are shown in Fig. 10b. The base mesh size for head and block was 5 mm with minimum size was kept at 10% of the base size after the grid independence study. A fine mesh is generated near the fluid and sold interfaces to keep the wall $y+$ within 5. The differential equations governing the flow, turbulence and heat transfer under the assumptions of steady, incompressible flow [19–21] were solved using finite volume method. All these equations are solved using Star CCM+™ finite volume commercial code. Implicit second order upwind scheme was used for solving the

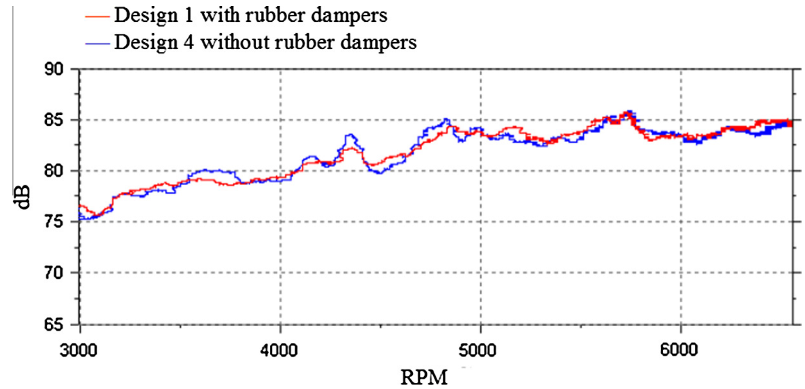

Fig. 8. Noise level during gradual acceleration for design 1 (with dampers) and design 4 (without dampers).

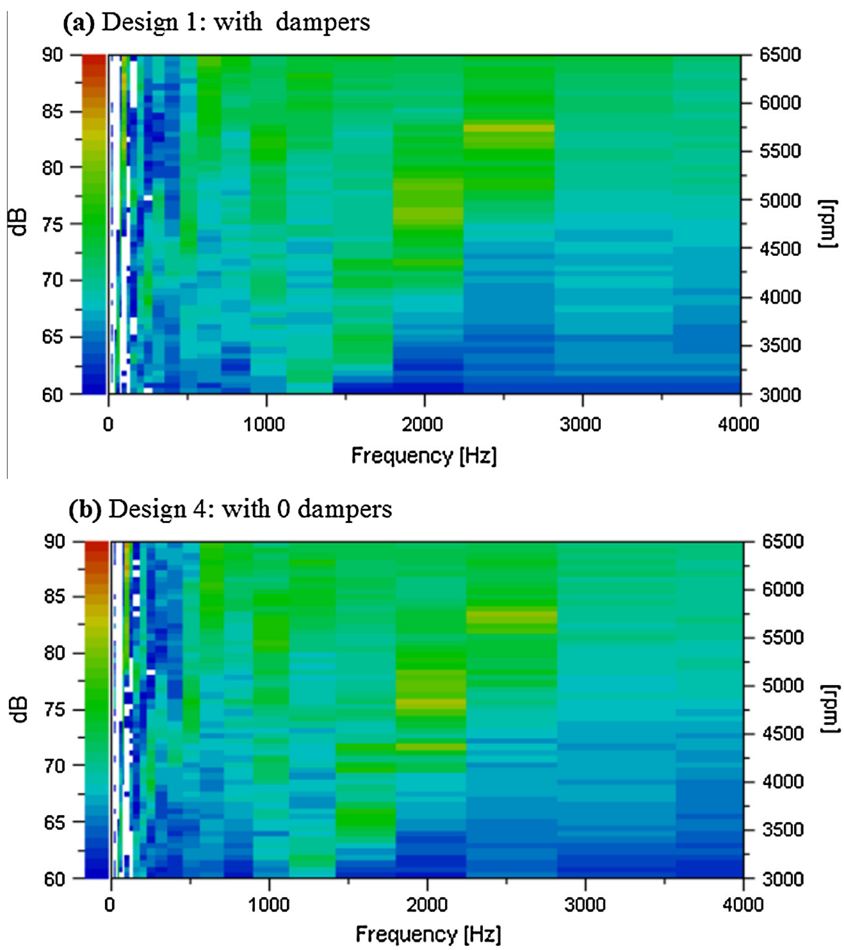

Fig. 9. Campbell diagram showing noise levels with (a) rubber dampers on base design and (b) without rubber dampers on design 4.

above equations. The pressure correction approach using the SIMPLE algorithm was used to solve the variables. The convergence criterion was fixed such that the residual values are lower than 10^{-6}. Two sets of simulations were carried out: one with rubbers dampers between the fins and other without rubber dampers. In the first case, properties of rubbers were specified.

4.2. Boundary conditions

Velocity of vehicle was specified as 40 km/hr at the inlet whereas static pressure (ambient value) was specified at the outlet of the wind tunnel. Constant heat flux boundary condition was applied in the engine liner and combustion chamber. This heat flux was taken as 50% of the maximum engine BHP as thermal load. This heat input again divided into 70% and 30%. The 70% heat flux was applied at combustion chamber by dividing with the surface area of combustion chamber. The remaining 30% was specified on 70% of the area of the liner. The bottom 30% liner is considered as adiabatic boundary wall. Convection boundary conditions for the inlet and exhaust ports were calculated from the correlations available in the literature. The heat transfer coefficient and bulk temperature on inlet ports were specified as 150 W/m^2 K and 50 °C respectively, and exhaust port as 250 W/m^2 K and 650 °C. The contact resistance between the fins and dampers was

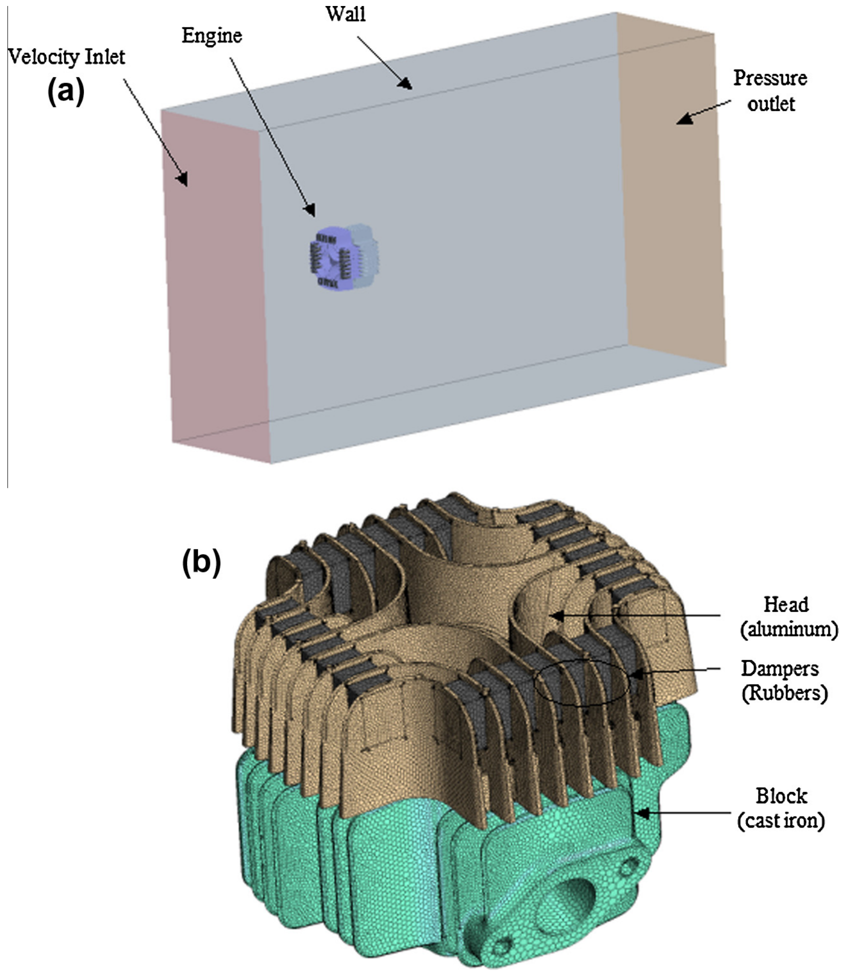

Fig. 10. CFD model of the engine. (a) Engine place in the wind tunnel, (b) volume mesh of the engine with rubber dampers.

neglected. For validation of these boundary conditions, a full vehicle CFD model was build though the purpose of this exercise was different and not to study the effect of rubber dampers. Experiments were performed and temperatures were measured at the specified locations on the engine. Though details of the validation exercise are not presented here, the results from the CFD model were within 10% of the experimental results.

4.3. Results and discussion

Table 2 shows the volume average cell relative velocity (CRV), Nusselt number and temperatures of the engine with and without rubber dampers. CRV is the relative velocity between the engine surfaces and the boundary. It is noticed that CRV is 44% higher on the cylinder head without rubber dampers compared to the case with rubber dampers. Cylinder block shows similar results. When dampers are present on the cylinder head, downstream velocity also gets affected and this results in lower CRV. A higher CRV signifies higher heat transfer from the engine surfaces and hence higher Nusselt number (Nu). Nu is a non-dimensional number and defined as the ratio of convective to conductive heat transfer across the boundary. Table 2 shows that when no dampers are used on the cylinder head, an increase of about 13% and 27% in the Nu of cylinder head and block respectively will be obtained. In other words, heat transfer on engines decreases with the use of rubber dampers. This is undesirable. The overall effect of dampers is the increase in temperatures of head and block. High engine temperature may lead to the failure of engine components [22]. Without dampers, head and block temperatures are 93.49 °C and 115.52 °C respectively, which increase to 103.23 °C and 127.48 °C when dampers are used. That is, about 10% increase in the engine temperature. Clearly, use of lengthy fins to enhance the heat transfer becomes counterproductive when rubber dampers are used. It is to be noted that the number of dampers simulated is more than that is present on the engine (see Fig. 1). Simulations were also carried out with same numbers dampers as seen in the Fig. 1 and the results fall in between the two cases shown in Table 2.

Now the local variation of temperature and cell relative velocity near the region where dampers are present is discussed. Fig. 11a shows the variation of non-dimensional temperature T^*

Table 2
Comparison of velocity, Nusselt number and temperature of the engine with and without dampers.

	With damper	Without damper	% Difference
Cell relative velocity (m/s)			
Head	5.61	8.08	44.00
Block	3.34	4.76	42.50
Nusselt number			
Head	23.54	26.69	13.38
Block	19.46	24.74	27.13
Temperature (°c)			
Head	103.23	93.47	−9.45
Block	127.48	115.52	−9.38

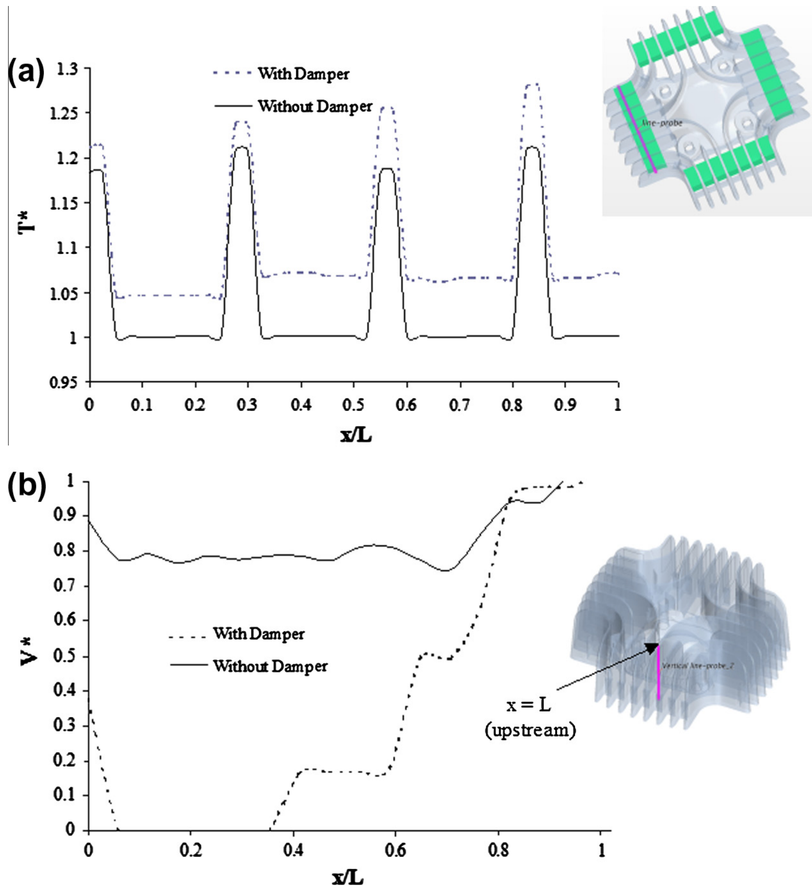

Fig. 11. (a) Non-dimensional temperature variation across the fins and rubber dampers along the line (see inset), (b) non-dimensional cell relative velocity variation along the fin in vertical direction.

(normalized by free stream air temperature) along a line passing through the dampers. Temperature data is extracted from the model along the line (see inset) that passes through the centre of each damper. Due to symmetry in the structure, temperature variation is shown from one end of line to the middle of it, which includes fins with three and half number of dampers. The wave like temperature variation is due to the presence of metal and rubber dampers or air. In case of without dampers, the figure shows air temperature. Temperature rises due to fins and decreases due to rubber dampers or air. It is to be noted that temperature is consistently higher throughout the length when dampers are present. The rise in the damper temperature is due to the conduction from the fin surfaces. Fig. 11b shows the non-dimensional cell relative velocity V^* (normalized with the free stream velocity) variation along the direction of airflow shown by a line (see inset) passing over the fin surfaces. It can be seen from the figure that dampers affects the velocity magnitude near the fins surfaces. Velocity reduced to zero when dampers are present and it is significantly less on downstream side. This is due to the wake formation behind the dampers. The overall effect of these local variations in temperature and velocity is the increase in temperature of engine head and block as shown in Table 2.

5. Costs and environmental impact

Design 4 was implemented for the mass production of the vehicle. The major benefits of designing engine without rubber dampers were realized after implementation. The mass difference between the existing cylinder head (design 1, Fig. 6) and new cylinder head (design 4) was about 30 gram. Thickness of the fins were remained unchanged. Following are the few advantages: (i) rubber damper manufacturing process is eliminated completely; rubber production is harmful to the environment (ii) long term benefits; over a period of time typically after six months of use, rubber damper properties deteriorates and it becomes brittle due to high temperature of the fins. The initial grip between the dampers and fins reduces and finally these dampers fall off the engine. The noise radiated from the engine increase again. Hence, putting rubber dampers on engines does not provide long-term benefits, (iii) logistics and inventory reduction; logistics of dampers involves the integration of information, transportation, inventory, warehousing, material handling, and packaging, (iv) man power saving; since rubber dampers are not an integral part of the engine, additional workmen are needed to hammer down the rubbers between the fins (v) part count reduction; rubber dampers are additional parts that needs to put on the engine before integrating on the vehicle and hence increases the number of part count of the engine. In the present case 17 parts (16 dampers, 1 engine) reduced to 1 part, (vi) improves engine cooling; rubber dampers restricts the free flow of air around the engines and hence increases the overall engine temperature as discussed in Section 3. Clearly, an engine without rubber dampers has many advantages. In other words, an engine with rubber dampers affects the company's bottom line, which is generally overlooked.

6. Conclusions

This research work provides a systematic methodology to investigate the effect of rubber dampers on engines NVH and thermal performance using numerical simulations and experimental measurements. Experiments conducted in the semi-anechoic chamber on the engine with and without rubber dampers reveal that frequency band of the radiated noise increases at higher engine speeds without dampers. An increase in 3–4 dB noise levels was observed on the engine without dampers. Finite element analyses were carried out on the base design to understand the fin modes of vibration. It was observed that local fin modes at lower frequencies were the dominant modes of vibration. Further design concepts were investigated. It was found that with the increase in natural frequency of vibration, amplitude of vibration also reduces. Prototypes of the final design of the cylinder head was made and tested in the semi-anechoic chamber. The noise radiated by the new design without dampers were comparable with that of base design with dampers. Hence, it is demonstrated that an engine without rubber dampers can be designed with the same NVH performance. Further investigation on the effect of rubber dampers on engine thermal behaviour using CFD suggests that rubber dampers effects the free flow air around the engine, which in turn increases the overall engine temperature. It comes out from this research that use of rubber dampers affects industry bottom line with hidden costs and processes that makes the product non-eco friendly. This research may serve as a useful guideline to automotive industry for engine structure design without rubber dampers. Further research should be carried out to quantify the effect of aero-acoustic noise production due to fluid (air)-structure (dampers) interactions at various engine speeds.

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