Golf Cart Noise and Vibration Troubleshooting

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Abstract

Similar to the automotive industry, the expectations from customers for the noise and vibration performance of personal vehicles such as golf carts, ATV's, and side-by-side vehicles has continued to evolve. Not only do customers expect these types of vehicles to be more refined and to have acoustic signatures that match the overall performance capabilities of the vehicle, but marketing efforts continue to focus on product differentiators which can include the acoustic and vibration performance. Due to this increased demand for acoustic and vibration performance, additional NVH efforts are often required to meet these expectations.

This paper provides a sample of some of the efforts that have occurred to further refine and develop the noise and vibration signature for golf carts. Included are discussions regarding the current market and expectations for noise and vibration performance, along with a generalized approach for identifying the key characteristics of the acoustic signature that affect the end consumer(s). A breakdown of the overall system will facilitate a more detailed discussion regarding the performance of individual components and system and their effects on the overall acoustic performance of the vehicle.

Current Market

Currently the golf cart market is comprised of two segments based on powertrain layout: electric and gasoline powered vehicles. The electric powertrain has the benefit of being very quiet in operation, however it is costlier and more complex than a gasoline powertrain. Gasoline powered carts are cheaper and easier to maintain than their electric counterparts, but they are also significantly louder. In general, this cost vs. NVH performance is leveraged in marketing quieter gasoline powered carts, touting them as the low cost/same performance alternative to electric.

The concept of "quiet" can be measured and marketed by a variety of metrics. A-weighted decibels (dBA) are commonly used as a means of quantifying how loud a product is. Further refinement in the market sound characteristics could result in additional metrics being used to quantify the sound quality of golf carts in addition to solely relying on a measurement of overall sound level via A-weighted decibels.

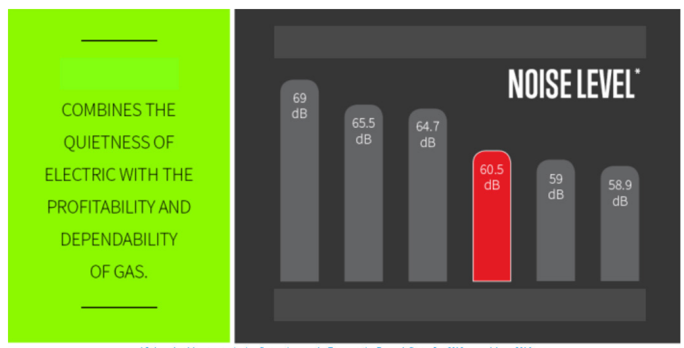

Figure 1. Marketing by noise level (dB)

NVH User Experience

The NVH performance of vehicles is highly correlated to the perceived user impression of the performance and quality of the vehicle. For golf carts, the high level of refinement available from the electric powertrain provides a case for similar NVH performance from the gasoline powered carts. Cases where low NVH impact is preferential in operation of the carts include user comfort, the ability to converse with passengers, and the effect on nearby houses or people.

Approach

The NVH troubleshooting approach in regard to golf carts is similar to that of most automotive troubleshooting studies. While the layout may differ, many of the components are fundamentally the same between golf carts and automobiles. The engine provides acoustic energy through intake and exhaust systems, and vibration through the body. Road surface input is structure-borne through the suspension and airborne to the operator position, though at typical golf cart speeds, nearly all NVH problems are powertrain related. This paper focuses on the powertrain source as it is the most typical dominant source for golf cart NVH.

Baseline Measurements - Problem Area Survey

The first task is to identify the potential areas for NVH improvement. Specific information about the vehicle signature may be gained through a formal Sound Quality Jury process, in which specific characteristics of the sound are identified by customer juries as objectionable. Another option is to address the NVH performance in terms of marketing efforts and direction. For example, if the market compares golf carts based on an overall Sound Pressure Level, then the troubleshooting focus should be on reducing that overall level. The output from these activities may direct, either explicitly or implicitly, the troubleshooting efforts toward a specific operating or load condition (e.g. starting from a stop at full-throttle on level surface). Sometimes subjectively evaluating the vehicle is necessary to make the jump from consumer or marketing feedback to objective operating conditions for testing.

Speed run-ups or sweeps are useful in examining the NVH-space of the vehicle. By analyzing a speed sweep NVH issues can be easily identified as forced responses or resonances, and help to determine "hot-spots" in engine speed or in frequency that should be further investigated. This helps to focus the troubleshooting efforts toward the specific issues. The troubleshooting process then helps to identify what specific steps are required to achieve a reduction in overall level or an improvement in sound quality.

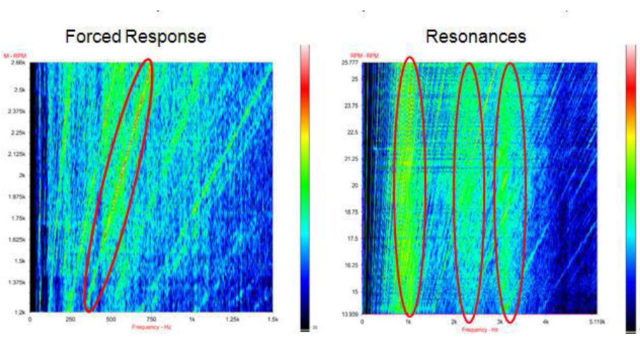

Figure 2. FFT vs RPM color maps - Forced Response vs. Resonances

Once the target operating conditions are identified, a repeatable and representative test condition must be defined. The "real-world" operating conditions of these vehicles vary in terrain, and can result in specific loading conditions that excite objectionable NVH performance. For testing, repeatability is paramount, and is best when performed indoors in a lab setting. However, for these vehicles outdoor operation is most representative. In order to achieve both repeatable and representative test conditions, some baseline data should be measured in the most representative setting in order to map the "real-world" conditions to the laboratory setting.

Source/Path Investigation

To effectively troubleshoot, the golf cart sound needs to be characterized in terms of primary contributors to the NVH problem areas. Identifying sources, paths, and receivers is instrumental in understanding the mechanism and flow of the acoustic and vibration energies.

A combination of operational and static testing can be utilized to quantify and explore the NVH problem areas. From the initial baseline measurements, the problem areas should be determined to be either airborne or structure-borne, and either caused by forced response or resonant behavior. Airborne versus structure-borne can often be determined by the frequency range of the problem areas. Frequencies below 500Hz are often structure-borne, and above 500Hz are often airborne. Forced response versus resonance is indicated as shown in figure 2.

Once these components are identified, they are controlled so as to determine the mechanism that defines the contributor. Airborne sources and paths are modified by creating resonance shifts in frequency (i.e. changing the length of an air intake snorkel) or by adding barriers or absorptive materials. Structure-borne paths are controlled by modifying their mass and/or stiffness to shift resonances and sensitivities either up or down in frequency.

Once the potential sources and paths are identified and their mechanisms defined, the next step is to establish a link between the key sources and paths and the receiver. This is typically done through an artificial excitation appropriate to the source or path type. If local control of the source or path precipitates a similar modification at the receiver position, then that source or path is identified for countermeasure development.

Countermeasure Development

As the potential sources and paths are explored and their mechanisms defined, countermeasure concepts are assembled. The modifications to key sources and paths can be used as countermeasures themselves, or guide the assembly of appropriate operationally-feasible countermeasures. Note that "operationally-feasible" does not necessitate production feasibility. As is often the case with NVH troubleshooting, the purpose is to demonstrate "proof of concept," so that the appropriate design changes or production feasible countermeasures can be developed and implemented.

The countermeasures are evaluated operationally according to the previously determined repeatable and representative test condition. Countermeasures are applied one-at-a-time and in combination in an effort to define a "best case" countermeasure package.

Typical Mechanisms

Shown below in Figure 3 is an illustration of the typical flow of acoustic and vibration energies from powertrain noise in a generalized golf cart application. The nature and connectivity of these elements is subject to variation based on the physical design and construction of a specific vehicle.

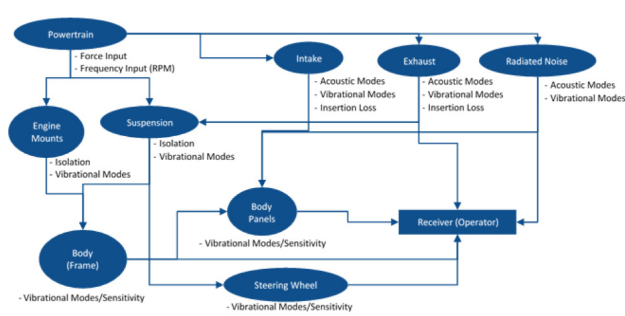

Figure 3. Typical source-path-receiver NVH Troubleshooting Flow Chart

Investigation Items

Key frequencies and/or operating speeds are identified for which improvements are desired. This identification of problem areas is used to focus the remaining activities for each of the following systems and components.

Powertrain Modes and Mounting Strategies

The powertrain has innate modal characteristics in the mounting configuration of the engine, intake, and exhaust. Based on the excitation by rotating components, primarily engine firing orders in this case, there can exist coupling between the powertrain modes and the operating speed. These components are studied both operationally and statically.

For the typical operational testing, both active and passive (relative to powertrain motion) sides of the powertrain mounts are instrumented with accelerometers, and the vehicle is operated according to the previously identified problem areas. Some mounting configurations can result in high displacements under load. In these cases, it is sometimes helpful to analyze in terms of displacement, in addition to acceleration. These data are analyzed in order to identify frequency peaks in the vibration signature that indicate resonant behavior, forced response, or frequency sensitivity, and also to determine the isolation effectiveness of the mounting components.

For static testing, excitation with a modal hammer is typically effective for identifying powertrain modes relative to the intake, exhaust, and engine mount configurations. When combined with the operational testing, this static testing can describe which peaks come from resonant behavior, and which peaks are due to operational forced response. For engine mounts, the active side of the powertrain is excited, typically impacting directly on the engine or transmission, in either three orthogonal directions (X, Y, and Z) or in one direction exciting all three directions simultaneously. For intake and exhaust systems, the modal analysis can be performed at two levels, system mounts and the system itself. The mount component is performed in a similar manner to the powertrain mounts, impacting active-side and measuring response on both sides of the mounts. Modal analysis of the intake and exhaust systems are performed by collecting responses at a set of points on the intake and exhaust system itself. For example, along the tailpipe, muffler, and header pipe.

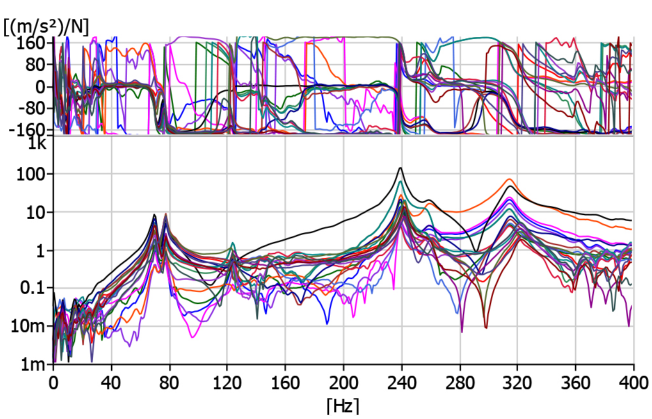

Figure 4. Example of Exhaust FRF's

Suspension Paths

The suspension in a golf cart is the path that links the body to the axles and powertrain simultaneously. As such, the bulk of the potential paths must involve the suspension at some point, making the suspension a key element of the troubleshooting process.

Where often in the automotive industry the suspension/subframe design involves multiple layers of isolation between the suspension and the body, golf carts typically mount the suspension directly to the body. In this case, the only isolation from the ground surface is provided by suspension spring and damper parts. As such, there is opportunity for refinement in the suspension at the design level, in order to improve NVH performance.

To investigate the suspension paths, a similar approach to the powertrain mounting investigation is typically employed. First, operational measurements are made with instrumentation on active and passive sides of suspension-to-body connection points. This is used to identify frequency peaks indicating either resonant behavior, forced response, or frequency sensitivity. Static testing is then performed by modal impact hammer at both passive (body-side) and active (suspension-side) measurement points, in order to identify which of the operational peaks is forced response versus resonant response or frequency sensitivity.

Intake and Exhaust

In terms of controlling the airborne powertrain noise, two primary contributors are the intake and exhaust. In the automotive industry much work has been done in regard to tuning the intake and exhaust systems to manage resonance and overall sound character.

As the market focus turns toward NVH performance, intake and exhaust refinements similar to those seen in the automotive industry can be applied to golf carts as well. The intake system of the golf cart is of particular concern due to its close proximity to the operator. Adding snorkels or tuned lengths and resonators to the intake system has been shown to improve the NVH performance at the operator position. Optimization of the exhaust system shows some promise as well, although the exhaust does not share the same proximity concerns with the intake system.

Both intake and exhaust systems can be approximated as pipes, and as such they have inherent insertion loss and resonant behaviors according to their physical dimensions. In general, given that the pipe material has some loss associated with it per unit length, longer pipes will attenuate airborne sound more than shorter pipes. However, for a given length of pipe there are resonant frequencies, at which there is little attenuation. It is important for the intake and exhaust lengths to be appropriately dimensioned in order to avoid the coincidence of resonant frequencies and engine firing orders or other problem frequencies.

Specifically, the intake and exhaust systems can be expressed as pipes with one closed termination and one open termination. The closed termination is found at the engine, where there is a large impedance change. The resonant frequencies of the pipes according to their physical dimensions is well known, and is given in equation 1, below [1], where c is the speed of sound in the pipe, and L is the length of the pipe.

$$f_n = \frac{nc}{4L}\,; \qquad n = 1, 2, 3 \ldots \tag{1}$$

At these resonant frequencies, the intake and exhaust systems can amplify the airborne powertrain noise. If the identified problem frequencies or speeds excite these resonances, the system length can be changed to move the resonance out of the frequency range of interest.

The air box and muffler are the primary sources of attenuation for intake and exhaust systems, respectively. These components typically operate on the principle of an expansion chamber, introducing a cross-sectional area change within the system, which produces a transmission loss (TL) of the sound propagation. In reality, the air box and muffler components are more complex; the methods described here are intended to direct the troubleshooting efforts.

The transmission loss of an expansion chamber can be derived from the muffler dimensions, as shown by equation 2, below [2], where S is the cross-sectional area of the pipe (subscript 1) or expansion (subscript 2), and L is the length of the expansion.

$$TL_{expansion} = 10\log_{10}\left[1 + \frac{1}{4}\left(\frac{S_1}{S_2} - \frac{S_2}{S_1}\right)^2 \sin^2(kL)\right]; \quad k = \frac{2\pi f}{c} \tag{2}$$

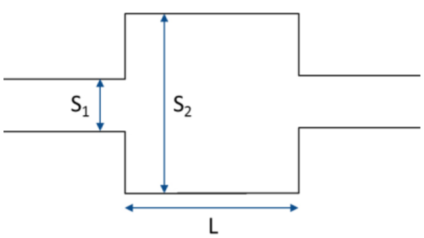

Figure 5. Expansion Chamber Dimensions

Note that this equation is formulated considering only plane wave propagation in the system, and damping is not considered. As such, the calculated attenuation will be more than seen in practice, and TL minima will be zero for the calculated, as opposed to a non-zero actual value due to damping.

A typical expansion chamber TL curve is shown below in figure 6.

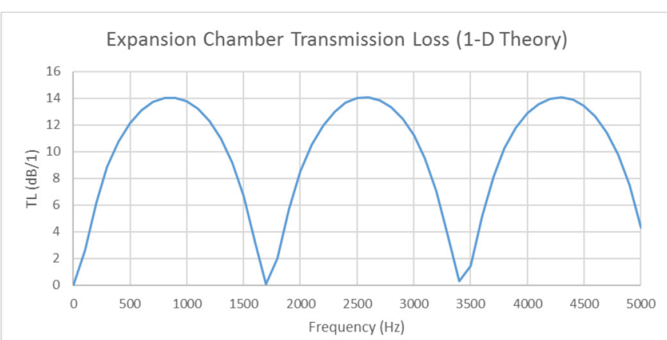

Figure 6. Typical expansion chamber transmission loss spectrum

Note that the expansion provides significant attenuation at its resonant frequencies, but negligible attenuation at other frequencies. Tuning of the expansion involves optimizing its dimensions so that the attenuation is present at frequencies of interest. Another method for optimizing the muffler performance is to include a side-branch, or quarter-wave resonator. This element can be tuned to a specific frequency, so as to fill gaps in the attenuation left by the expansion chamber.

Powertrain Radiated Noise

Another component of the airborne powertrain noise is radiated noise from the engine itself. Due to the open-air nature of the golf cart layout and the proximity of the engine to the operator position, this radiated noise can be a concern. For automotive applications, radiated powertrain noise is addressed with the use of under-hood acoustically absorptive treatments. The hood of a golf cart is also the passenger seat, typically a few inches thick and constructed of a structural panel underneath a foam cushion. Adding absorptive material and barriers to the underside of the seat panel have been shown to merit some improvement to the NVH performance of the cart, depending on the frequency range of interest.

In order to quantify the level of powertrain radiated noise that reaches the operator position, the noise reduction (NR) between the engine compartment and the operator position is measured. A microphone is placed both at the operator position and in the engine compartment. Operational measurements are made, and the difference in level between the two microphones is determined with respect to frequency. To improve the NR, absorption is typically most effective when the target frequency is above 1-2kHz, and barriers are best suited to frequencies less than 1kHz.

Figure 7, below, shows 1/3 octave spectra of two microphones, one in an engine compartment and one at a receiver position. In this case, above 1kHz there are ~20-30dB of noise reduction between the engine compartment and the receiver position, and below 500Hz there are ~10-20dB of noise reduction. Based on these data, there is some opportunity for improving the low frequency NR by adding barrier materials, however due to the open-air nature of the golf cart "cabin," flanking paths limit the overall effectiveness of airborne noise countermeasures.

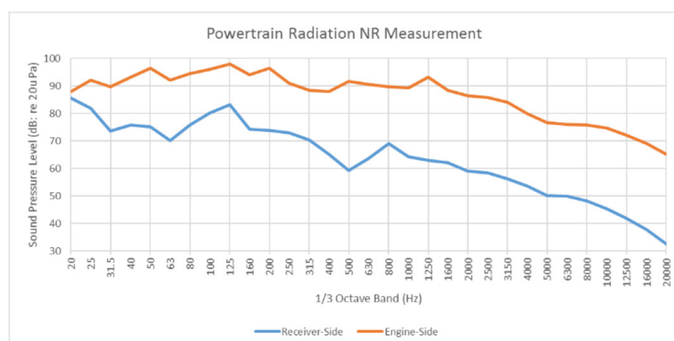

Figure 7. Example noise reduction measurement result - 1/3 octave spectra

Frame and Body Panel Sensitivities

The body shell of the golf cart is the primary barrier to airborne sound radiated from the powertrain, and also serves as the medium through which structure-borne noise from the suspension and body attachment points travels to the operator. Resonances at and sensitivities to concern frequencies enable the frame and body panels to serve as path amplifiers at those frequencies. Increasing the stiffness or adding damping treatments to sensitive path elements is an effective means of managing NVH contributions from these points.

Typical golf cart body shells are low mass/stiffness structures, and are relatively unsealed, in contrast to automotive applications where the vehicle body is well-sealed and substantial. This can present significant opportunity for increased NVH refinement. Golf cart body designs that limit panel sensitivities and act as more substantial powertrain enclosures will precipitate improved NVH performance.

In order to determine frame and body panel sensitivities, drive point (local) and acoustic sensitivity (to receiver position microphone) frequency response functions (FRFs) are measured using a modal impact hammer. The drive point FRFs illustrate local motion to the measured drive point, while the acoustic sensitivity FRFs indicate if the drive point local motion is potentially correlated to what the operator hears.

If a frequency peak at the drive point is also seen in the acoustic sensitivity FRF, then causality is established by altering the drive point response and observing the acoustic sensitivity for similar changes. Attaching mass or stiffening members to the component under study typically is sufficient means of altering the drive point response. If no change is observed in the receiver acoustic sensitivity, then the drive point in question is not significantly coupled with the receiver, and is of little concern for further troubleshooting activity.

In the case that there is a frequency peak in the drive point that is not seen at the receiver position, it is possible that the reason for which the two points are unrelated is that the vibration measured at the drive point is a component of an inefficient acoustic radiator. Acoustic radiation efficiency relates the vibration of a surface to its generated sound pressure. In general, the radiation efficiency of a component depends on the phase relationships of the mode shape. Surface points that are out of phase cancel, while points that are in phase amplify each other. In general, the ratio of in-phase to out-of-phase points indicates how well a component will radiate sound.

Results

For the case study described here, the main goal of the development activities was to reduce the overall sound pressure level as much as possible. The most objectionable operating condition was identified as full-throttle take-off from a standstill, and troubleshooting activities were conducted similar to what has been described in the sections above.

For the operational data, baseline measurements on the golf cart were conducted outdoors in a parking lot, in a grass lot, and up a grassy hill. It was shown that for all operating conditions the NVH performance was similar, so the parking lot case was chosen for ease of access. For the countermeasure evaluation portion of the testing, the vehicle was operated on a Hemi-Anechoic Chassis Dynamometer to ensure repeatability.

From initial speed sweep data, it was determined that the overall level was driven by frequencies below 400Hz, specifically engine order content. The 3rd engine firing order, and specifically the frequency range from 115-150Hz, was determined to be the highest priority for troubleshooting focus, as it provides the best opportunity for level reduction.

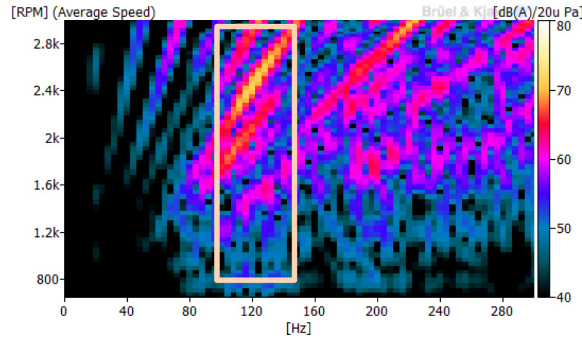

Figure 8. Baseline Sound Spectrum

Initial baseline data was conducted with transducers placed at key locations around the powertrain system to understand where potential sources and path sensitivities may lie. From this it was determined that the structure-borne powertrain noise could be passed through the powertrain mounts, exhaust vibration, intake vibration, and body panels.

A detailed troubleshooting analysis was conducted on each of these systems individually, using the methods described in the above sections, with the purpose of gaining understanding of the dominant sources and key transmission paths affecting the overall noise levels of the golf cart. As key sources and paths were identified, simple troubleshooting techniques such as the addition of mass or stiffness were utilized to assess the contributions to the customer interface points.

These investigations identified several systems and paths that were contributing to the overall sound pressure levels between 115-150Hz. The intake system, which was a large contributor, had a resonant mode around 135Hz. due to the overall system length. This resonance was coupling with a panel mode around the same frequency, causing a significant low frequency boom and high sound pressure levels. By changing the intake system length alone, these modes were decoupled, providing a 3-5dB improvement to the primary order content and 2-3dB improvement to the overall level.

Similar investigations were completed on other key systems to understand their contributions. These studies led to similar potential countermeasure opportunities. Upon completion of these studies, all of the potential countermeasures were installed one at a time and in combination, eventually building up to a “Best Case” countermeasure package at the conclusion of testing. At that point, the golf cart was then applied with the “Best Case” package and driven outdoors to prove the “real world” countermeasure impact.

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3. https://www.yamahagolfcar.com/efi/

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Definitions/Abbreviations

NVH - Noise, vibration and harshness

TL - Transmission loss

NR - Noise reduction

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