International Journal of Automotive Technology, Vol. 23, No. 4, pp. 1153–1161 (2022)
DOI 10.1007/s12239–022–0101–3

pISSN 1229–9138/ eISSN 1976–3832

AUTOMOTIVE ELECTRIC POWER STEERING SYSTEMS NVH PERFORMANCE INVESTIGATIONS

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(Received 16 November 2020; Revised 9 December 2021; Accepted 20 December 2021)

ABSTRACT–This paper deals with noise and vibration problems in electric power steering systems used in modern automotive platforms. Various types of commonly used power steering systems are presented. The noise and vibration performance of new EPS designs can have a significant impact on users. Computational FEM methods used to investigate the dynamic properties mechanical systems were implemented to situationally test the NVH performance of EPS systems. Computational FEM model building procedure is presented with detailed description of crucial EPS system elements and contacts. Conventional EPS vibro-acoustic measurement techniques such as modal testing and Field Data Replication are presented alongside newer techniques such as acoustic camera measurements.

KEY WORDS : NVH, Electric power steering, EPS, FEM, Acoustic camera, Modal analysis

NOMENCLATURE

NVH : noise, vibration, and harshness
EPS : electric power steering
HPS : hydraulic power steering
CEPS : column-assist EPS
BEPS : level brush motor column-assist EPS
REPS : rack-assist EPS
SPEPS : single pinion assist EPS
OEM : original equipment manufacturer
FEM : finite element method
BCs : boundary conditions
CRB : center rack bushing
FDR : field data replication

1. INTRODUCTION

Automotive industry is known to be one of the most demanding branches when it comes to NVH. Designers are trying to temp new customers with breath taking car body and interior design, but an important factor that can strongly influence the purchase decision is also the acoustic and vibrational performance. New materials, constructions and shapes are introduced all over the car in order to improve the NVH. Development of new components for power trains, chassis, interior Electric Power Steering (EPS) can introduce new potential noise and vibration problems that might lead to low sale volume/rate or generate additional service and operational costs due to warranty repairs and claims. In order to prevent potential or existing NVH problems originating at EPS both numerical and experimental tests are conducted. Chosen aspects of computational and experimental approaches, both well established and newer techniques, are presented in next sections of this article.

Steering systems implemented in passenger cars are under constantly undergoing modifications and development. Numerical methods are used in the process of design and troubleshooting of new products. Numerous literature positions describe computational approach for power steering noise and vibration simulations. Badawy *et al*. (1999) describe the use of reduced order modelling for EPS systems. Patrick (2002) presents a EPS numerical model including mechanical and electric parts of the system. Besson *et al*. (2009) describe the problem of power steering systems dynamic behavior simulation, proposing a FE simulation model with linear and nonlinear elements, which allow to simulate system transit response to excitation signals. Kim and Kim (2008) describe the problem of power steering systems vibration at idling and numerical modeling of this phenomenon. Freund *et al*. (2017) studies the EPS vibrations with use of a continuous system model.

There are two main types of power steering systems used in the automotive industry: hydraulic power steering (HPS) and electric power steering (EPS). HPS uses high pressure fluids to assist driver steering. An engine-driven power steering pump creates system pressure. Then pressurized fluid is routed into a steering gear cylinder that turns the wheels of the vehicle. HPS is currently seen as decreasing

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Figure 1. Single pinion assist EPS.

Figure 2. Rack-assist EPS (REPS).

technology and is set to be fully replaced by EPS technology in the near future. As for disadvantages, one of the key ones is that the pump must run constantly which wastes engine power, hence increases the fuel consumption and, finally, the CO_2 emissions.

With EPS, an electric motor replaces the hydraulic pump. The electric motor is either attached to the steering rack (Rack EPS, Single/Dual Pinion EPS) or to the steering column (Column EPS). The electronic control unit controls the steering dynamics. EPS is often a preferred system since it results in better fuel economy and lower emissions. EPS systems most commonly are a combination of electric motors, actuators, sensors and controllers that provide reliable steering control in all driving conditions. Dyer (1997) shows a comparison of various power steering systems and the influence on vehicle fuel economy. Consequently, new EPS systems reduce fuel consumption by up to 6 percent and CO_2 emissions by up to 8 g/km (Nexteer, 2020). It also saves more space in the engine compartment for the car manufacturer and eliminates the leakages seen in the HPS systems. Moreover, EPS systems protect from Loss of Assist (Nexteer, 2020) in case the engine turns down while driving the vehicle.

Dual motor electric power steering systems applications are considered for commercial vehicles. Li *et al*. (2019) shows the advantages of dual motor EPS systems such as improved control stability, reduction of vehicle weight influence on steering performance.

Column-assist EPS (CEPS) integrates system electronics (motor, controller and sensor) and the assist mechanism with the steering column. There are also entry level brush motor column-assist EPSs (BEPS) available, specifically tailored for developing markets.

Rack-assist EPS (REPS) presented in Figure 2 integrates the required electric assist mechanism with the steering rack where they are contained under the hood in the engine compartment. Rack Assist EPS is designed for heavier vehicles to handle higher front-axle loads and optimize packaging space. These systems are capable of steering mid-sized vehicles, all the way up to full size pick-ups/ trucks. System electronics are integrated with the steering rack housing and are designed to withstand the high temperatures and environmental exposure typical in the engine compartment. In addition to saving valuable space in engine compartments, REPS' unitized motor-controllers are easily fitted to different vehicle platforms. They reduce assembly costs and increase performance integrity compared to systems employing remote electronic controls.

Pinion-assist EPS expands the application range and flexibility. Single Pinion Assist EPS (SPEPS) integrates the electric assist mechanism into the primary steering gear pinion shaft. Single Pinion EPS has a design range comparable to Column-Assist EPS, but offers OEMs greater packaging flexibility, reduces OEM crossbeam strength/stiffness requirements and reduces torsional compliance of driving torque through the column and intermediate shaft assemblies. Dual Pinion EPS allows for the primary pinion to be optimized for vehicle dynamics and performance and a secondary pinion to be optimized for assist. It also provides additional design flexibility to locate the motor, controller and assist mechanism opposite the driver side of the vehicle hood.

2. NUMERICAL TESTING

As presented in the introduction section, the electric power steering is a multicomponent mechanical system. For its proper functioning and expected dynamic, vibration and noise performance the relatively high precision of geometry must be provided in interacting components surfaces. Additionally, while the dynamic behavior is one of the crucial element, multiple mechanical properties e.g. stiffness, friction coefficient or damping, are required to be included in the model. When taking under consideration system like EPS, one can face a challenge to develop proper model and the reasons are mentioned above.

To develop model which complexity is suited to the investigated phenomenon, great understanding of the system plays the underlying role.

In purpose to investigate NVH performance of the mechanical components, the simulation type must be dynamic. Due to the character of investigated phenomenon and its relatively long observation time, implicit dynamics analysis is recommended. This will allow to run longer simulations where time trace signals from experimental test can be used as load and the computational time will be relatively acceptable. However, one should expect convergence problems in such complex system. Whenever that is possible, without negative influence to representation of real behavior, simplified spring, damper, beam elements etc. should be used in the model. This can be done only if the detailed behavior of specific subassembly is not the goal of the investigation. However, if needed the detailed model of particular subassembly can be developed as a separate simulation e.g. steel composite gear models for vibration analysis (Kim *et. al.* 2019). The results can be substituted by proper boundary conditions or substituting elements to the global model. If supported by the used software, sub modeling or super element approach can be used.

Modal analysis of the full system would not be the proper simulation type. System and its interaction, contacts, supports characterizes with nonlinearities which play significant role in the system behavior. The main assumption of modal analysis is system linearity and reciprocity. Assessment of the EPS system with that approach, would give valuable knowledge. However, preliminary assessment of main structural components, with use numerical modal analysis, should be the first step of model preparation. This will provide the general overview about the dynamics of the components and can be used to define the allowed geometry simplification of EPS assembly elements. In case of the EPS, the main critical assembly components are usually: housing, rack, pinion, worm wheel and wheel, or any other major component of the system. Having the knowledge about the dynamic characteristics of those particular components, decisions about using proper constrains, contacts, substituting elements etc. can be made.

In the presented example, numerical modal analysis of the housing indicated that main structural modes are of relatively high frequency. Moreover, the housing itself, when assembled in the car or on the testing rig, is clamped by bolts to the frame of higher rigidity than the housing itself. That makes the housing structure relatively stiff element of the system.

Additionally, internal structural components are supported in the housing through elastic supports (bushings), components with clearance (bearings) or damping elements (vibration isolators). When comparing the relative stiffness of all mentioned elements, one can assess that stiffness of housing will not play the crucial role on the NVH investigation and model developments should be started from internal structural components only, which through positioning elements (bushings, bearings, vibration isolators etc.) will be rigid supported – representing the stiff housing. Elimination of housing significantly reduces the model size what directly corresponds to the computational time needed to run the simulations. Housings of great stiffness can be included in the model as one of final elements if needed, after the behavior of internal components is properly represented.

In case of internal structural components, the preliminary assessment of dynamic characteristics will also give the information if the selected mesh size is proper. As presented in Figures 3 and 4 the general finite element mesh size does not correspond to the gear mesh contact area. Only locally, where contact of rack bushing assembly components and rack-pinion will be present, were represented with a finer mesh size adjusted to the tooth geometry. However, as

Figure 3. Rack FEM mesh.

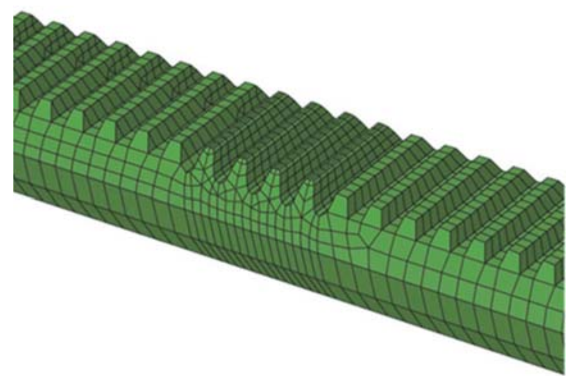

Figure 4. Rack gear contact – finer mesh area.

Figure 5. Rack structural mode shape.

visible in Figure 5 the general, coarse mesh size was appropriate for representation of structural mode shapes – even those complex ones. Proper optimization of the mesh size of every particular component allows to minimize the total model size and reduce the computational resources and time needed to conduct simulations.

When an experienced investigator is running the simulation, not all components need to be verified by numerical modal analysis. If the structural modes are expected to be out of frequency bands of interest for the other dynamic simulations, mesh must be adjusted to represent general stiffness of the components. Only contact areas can be modeled precisely to properly represent local boundary conditions e.g. stiffness and gear meshing. Example of this approach is the worm wheel presented in Figure 6.

With that approach, one can observe that preliminary assessment of dynamic behavior allows to significantly reduce the size of the model. The numerical modal analysis results should be compared with the experimental modal analysis carried out for components as a preliminary validation of the model.

Such procedure should be run for each main structural

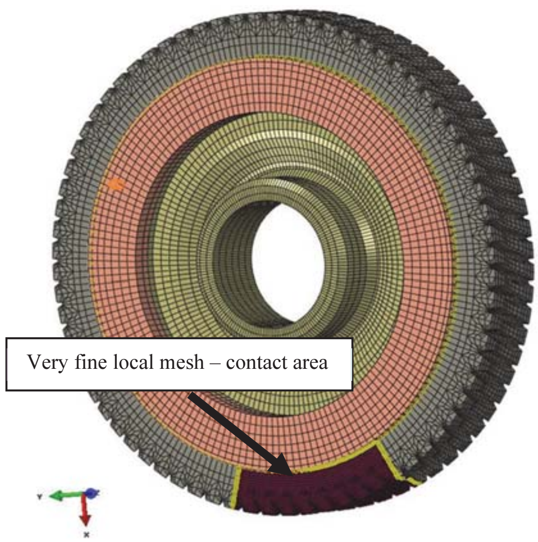

Figure 6. Worm wheel mesh definition.

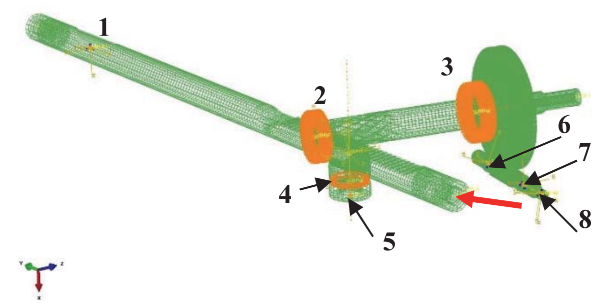

Figure 7. Model assembly and boundary conditions.

component of the assembly which modal modes may contribute to resonances and deteriorate the system NVH performance.

The same force time signal was implemented in experimental testing with an electrodynamic shaker (subsection 3.2).

2.1. Interactions Definition, BCs

When all structural components are modelled and assembled, proper definition of boundary conditions and interactions need to be defined. As the simulation consist of several structural components which mostly interacts in the model with contact, any simplification of BC's which can be introduced will improve the model performance (convergence, calculation time) significantly. In the EPS structure considered in this paper, boundary conditions were defined as follows (Figure 7) where some of theme where realized with of the connectors:

1 – assembly of bushing and two O-rings, 2, 3 – fixed support of the bearing outer ring, 4 – spring and O-ring supporting CRB, 5 – one side fixed slot connector, 6 and 7 – dampers/ vibration isolators supporting the worm, 8 – electric motor (inertia, electromagnetic resistance). The stiffness of the component 1, which is subassembly of the bushing and O-rings, was identified in separate simulation where the assembly was loaded by normalized load in all 6DOF. With that approach the complex pre-stress simulation and nonlinear interaction between those components was substituted by the bushing connector corresponding to the stiffness identified in the separate simulation. Thanks to that, the main simulation (dynamics) was less time consuming and with greater convergence.

The red arrow in Figure 7 represents place where external load was applied to the structure. Load was applied as a force signal in the time domain. Force value and character of its variability was acquired during track testing on chosen road surface types, vehicle speeds and vehicle platforms.

2.2. Simulation

Simulation was conducted in two time steps. First the preload of springs, bushing and vibration isolating components. Second step consisted of application of operational load – force time trace. Preliminary results indicated that the noise and vibration problems are triggered by the impact load observed as a dynamic collision between CRB-RACK-PINION pair.

Identification of the causing phenomenon (Figures 8 and 9) is relatively straightforward. The main challenge in the definition of simulation FEM dynamic model, where the NVH is the main target of the results simulation, is the proper representation of impact between the two components namely investigations of contact stiffness. When assessing the global dynamic behavior of the components (e.g. deflection shape) the local contact stiffness

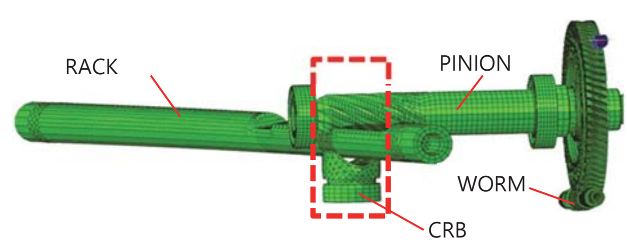

Figure 8. Contact separation.

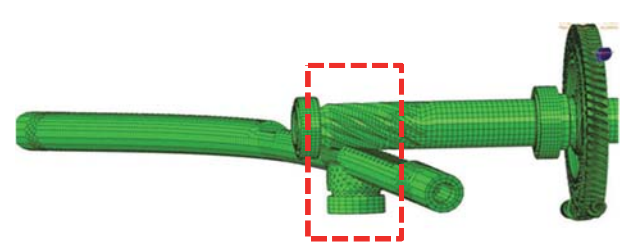

Figure 9. Impact between components.

does not play the main role. However, when the vibration and noise generation is analyzed, the local impact between components is the main analyzed phenomenon.

The default "hard contact" interaction in most cases introduce non-proportionally high dynamic response of the components. Moreover, in the production EPS assembly the surfaces of rack – pinion contact is covered with grease which is introducing strong damping. Due to the fact, that real contact stiffness, as well as damping coefficient, were not defined, several simulations covering multiple combinations of contact stiffness and contact damping definition were performed. Exemplary results of acceleration at the CRB component, in the CRB main axis - rack perpendicular direction, are presented in Figure 10 where;

test_config_1 – stiff worm longitudinal support, linear general contact stiffness

test_config_2 – nonlinear elastic longitudinal support of the worm (worm dumpers), linear general contact stiffness

test_config_3 – worm dumpers, nonlinear general contact stiffness

test_config_4 – worm dumpers, linear general contact stiffness, 0.1 viscous dumping on CRB spring

test config_5 – worm dumpers, linear general contact stiffness, 0.2 viscous dumping on CRB spring

This presents, how the model is sensitive for mentioned parameters and how it can change the response range. The most reliable and resulting in expected model behavior/sensitivity, is contact definition of nonlinear contact stiffness and bilinear contact viscous damping model. In Figure 11 magnified acceleration course of single CRB impact is presented.

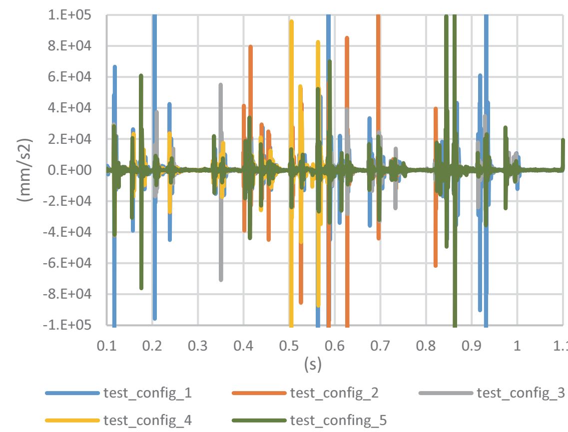

Figure 10. Acceleration response measured on CRB in the direction of its translation – simulations results.

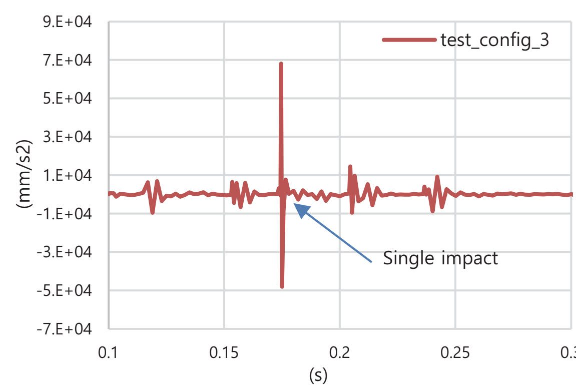

Figure 11. Acceleration response measured on CRB in the direction of its translation - simulation results.

3. EXPERIMENTAL TESTING

Noise and vibration performance of electric power steering systems can be investigated with use of several measurement techniques and approaches. Standard sound pressure level tests provide some information about the noise levels at certain operation conditions but in many cases acoustic tests are run according to specific – car platform and manufacturer requirements. EPS noise and vibration problems can be investigated with a root cause analysis approach by implementation of following techniques: Modal analysis; Operational vibrations and noise - test rig; Acoustic camera measurements - test rig; Field testing –test track. Field testing is time and resource intensive and can mostly indicate that the problem of noise or vibration exists at the point of advanced product design rather than during early project.

3.1. Modal Analysis

Experimental modal analysis is used to determine the dynamic characteristics of choses EPS components and assemblies. Depending on the construction of the EPS modal analysis can be carried for racks, pinions, housings, power packs etc. Natural frequencies and mode shapes of mentioned components can indicate noise and vibration causing elements, assemblies or areas and provide information about measures that can lead to vibration and noise reduction (Norton and Karczub, 2003; Crocker, 2007). Depending on the element size and frequency range of interest's modal analysis can be performed with use of an impact hammer or shaker.

First three mode shapes of the EPS rack are presented in Figure 12. The rack was removed from housing and suspended flexibly during measurements. The natural frequencies of the rack are significantly above the standard excitation frequency range which is assumed, depending on manufactures to be less than 50 Hz. Nevertheless, mode shapes and frequencies of the rack and other main EPS components can be used in FEM model validation. Computational modal analysis results of those elements can be compared with experimental data to simplify the FEM

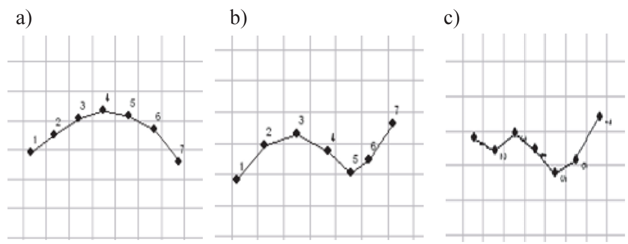

Figure 12. EPS – Rack mode shapes at a) 260(Hz), b) 720(Hz), c) 830(Hz).

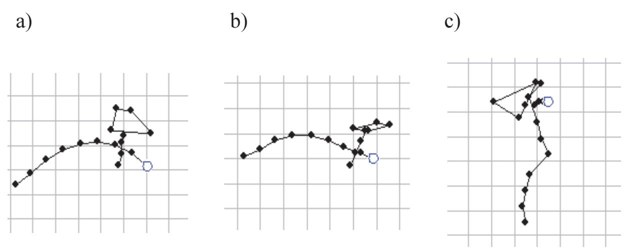

Figure 13. EPS – Housing mode shapes at a) 320(Hz), b) 730(Hz), c) 790(Hz).

geometry, modify the mesh size and validate the chosen element models.

Figure 13 presents first three mode shapes of the EPS housing. Once again the natural frequencies are positioned out of the excitation range. Similarly to the rack measurements, this data can be used to optimize the FEM system dynamics simulation. Those experiments validates numerical model (rack) and proves that modeling of the housing is not necessary (see section 2) while it stiffness is much greater than the internal components stiffness.

3.2. Testing on the Rig

Testing of electric power steering systems can be carried in anechoic or semi-anechoic chambers. The tested specimen is usually mounted to a heavy test bench placed inside the anechoic chamber via a custom fixture which resembled in car mounting position and insures proper tie rod angle position in relation to the excitation force vector at the shaker.

Excitation signals depend on the test procedure, in some test cases random or sine signals are used. In a specific car platform tests, the input force signals are acquired at various points of car prototype during test track runs and then replicated at the test rig by means of FDR (Field Data Replication). Excitation force generated on the electrodynamic shaker is applied to the tested specimen via a stinger attached to the tie rod joint which resembles the EPS real operating conditions and allow to better noise and vibration troubleshoot. System response is registered with accelerometers and sound pressure transducers. The

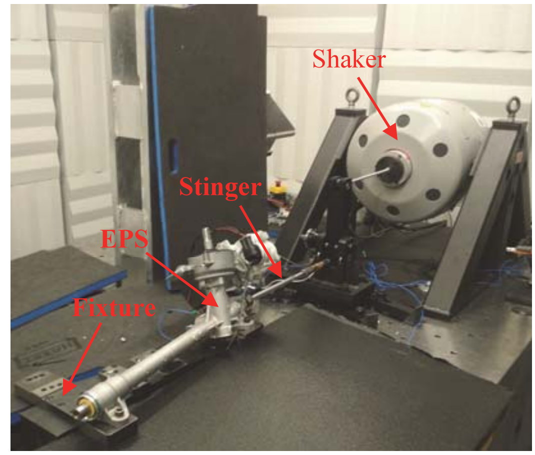

Figure 14. EPS in the semi-anechoic chamber.

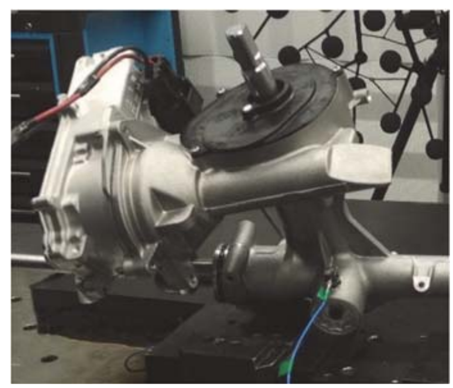

Figure 15. Figure 1 Acceleration measurement points.

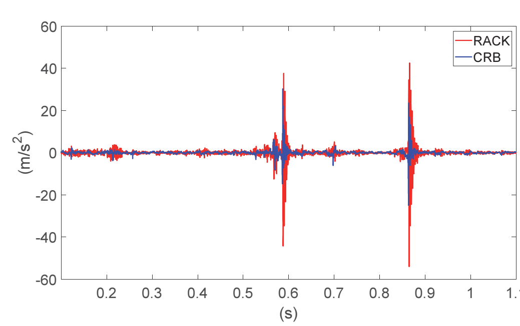

Figure 16. Exemplary acceleration signals acquired at two different points at the EPS - test track excitation.

acceleration measurement points are selected based on engineering and product experience.

Acceleration signals presented in Figure 16 can be used to indicate the vibration origin and noise causes. Detailed analysis of system input and output signals can directly connect the test track occurrence such as braking, turning etc. to vibration problems and its place.

3.3. Acoustic Camera Testing

Acoustic camera testing is a relatively new noise source localization method in comparison to sound intensity or acoustic holography. The acoustic camera is a multi-microphone matrix coupled via digital signal processing algorithms with a video camera. The sound pressure level map measured by microphones is overlaid on observed object photo or video in case of non-stationary noise. Additional information on acoustic cameras, computing algorithms such as beamforming etc. is presented in work of Hald and Christensen (2004), Erić (2011) and Michel (2006).

Similarly, to mentioned above vibration measurements, acoustic photos and videos can be taken during excitation with external signals. Data analysis algorithms allow to investigate chosen frequency ranges and create acoustic photos and videos at specified frequency range of interest. Usually photos are generated in frequency ranges where the sound pressure level exhibits higher values or for NVH problem specific range. FDR testing at rigs allows to pinpoint the noise source with the acoustic camera, otherwise impossible during test drive.

Acoustic photos acquired during camera measurements can help to pinpoint the cause of noise, especially when the operation or natural frequencies of EPS components are known. Acoustic camera measurement results can be compared with other tests and simulation results to find the noise root cause. The noise localization at 1400 (Hz) coded with color scale presented in Figure 19 covers a wider area than in case of other analyzed frequency ranges. In this case the acoustic signal was caused by impacts presented in Figure 16. The recorded noise was not stationary thus the acoustic photo cannot fully display the actual emission. In such case another approach is possible, instead of acoustic photos, an acoustic video can be rendered with noise source localization for chosen frequency ranges. This method provides detailed information about the change of noise emitting areas during excitation.

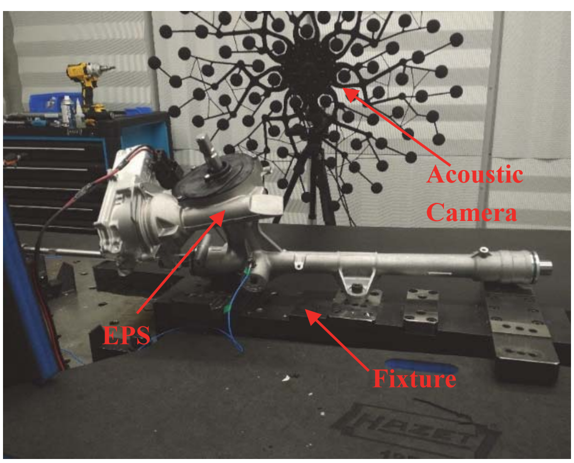

Figure 17. EPS on fixture, acoustic camera in the background.

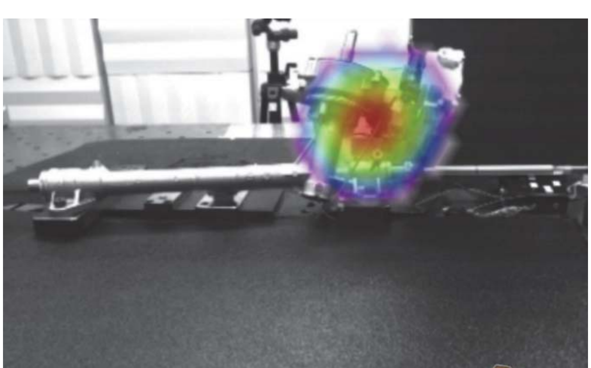

Figure 18. Acoustic photo – Noise localization at frequency of 1000 (Hz).

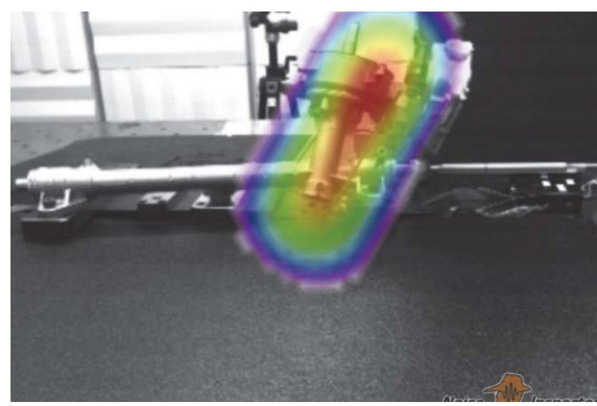

Figure 19. Acoustic photo – Noise localization at frequency of 1400 (Hz).

Figure 20. Acoustic photo – Noise localization at 8200 (Hz).

4. CONCLUSION

It is well known that vehicle noise and vibration performance has great impact on sales rate as well as on the number of warranty repairs and customer complains. The steering wheel of a passenger car is one of the most important link between the driver and the vehicle. This link incorporates a power steering system which performance can significantly affect the driving experience. Noise and vibration originating at the EPS can be transferred directly to the driver via the steering wheel or cause problems in other areas of the vehicle due to structural transfer.

Experience in designing and manufacturing power steering systems is obviously an important factor that helps with preventing NVH problems sourcing from EPS. Structural interactions between the EPS and vehicle chassis or other components is hard to predict especially in some cases e.g. different vehicle platform.

Experimental testing is a great tool to identify and solve NVH problems however vast resources are needed to conduct experimental investigations on prototypes in different environments such as test tracks or laboratories. Well known experimental methods such as dynamic structural testing are used to define the vibration properties of power steering systems and solve NVH problems. New testing techniques such as acoustic camera measurements can be used to localize the noise source much faster than conventional methods. Acoustic cameras can be extremely useful for nonstationary noise phenomena. An acoustic video can indicate the origin of noise problem and the potential transfer path indicating crucial interfaces and objects for further FEM modeling.

Computational methods are used alongside experimental investigations to improve EPS noise and vibration performance. Finite element method dynamic EPS models are created to investigate the dynamic behavior of the steering system. Internal interactions of the EPS are considered in analysis as well as the interactions between other car sub systems. Multi parametric models incorporate diverse material properties, material and geometrical nonlinearities and various contact types between components. Models are validated through experimental research, which can be more cost effective in comparison to large scale full system prototype experiments. Modal analysis was used to validate the models of major EPS components and rule out potential resonance conditions. Test rig FDR vibration testing provides great benchmark for implicit dynamics simulation result comparison and final model validation. The acceleration level of impact obtained in numerical simulation is comparable qualitatively and quantitatively with experimental test results.

Validated models or their crucial parts can then be placed in new virtual environments that represent new vehicle platforms in order to investigate the NVH performance of the new system.

ACKNOWLEDGEMENT–Research presented above was cofounded by the European Union, European Regional Development Fund - Operational Programme Smart Growth 2014-2020. Project implemented as part of the National Center for Research and Development: Priority Axis I: Supporting R&D carried out by enterprises, Measure 1.2: Sectoral R&D Programmes. Project no.: POIR.01.02.00-00.0270 / 16-00, “New generation of Single Pinion Electric Power Steering system ready for future EU market needs with Automatically Commanded Steering Function and enhanced mechanical performance”. Project value: 19,645,215.24 PLN, Co-financing amount: 8,203,215.94 PLN.

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