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Experimental structure-borne energy flow contribution analysis for vibro-acoustic source ranking

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ABSTRACT

Noise legislations and the increasing customer demands determine the NVH (Noise Vibration and Harshness) development of modern commercial vehicles. In order to meet the stringent legislative requirements for the vehicle noise emission, exact knowledge of all vehicle noise sources and their acoustic behavior is required. The development of new analysis tools to investigate the vibroacoustic behaviour within vehicle development process is of essential importance to achieve better products in combination with time and cost reduction. This paper discusses the application of an experimental multi-point and multi-DOF energy flow analysis (EFA) methodology for the NVH assessment and ranking of vibro-acoustic sources. The methodology consists of two steps in which the system mobility matrix are obtained by direct measurements whereas the dynamic loads under operational conditions are obtained through an inverse procedure. A finite difference technique is used to distinguish between degrees of freedom. A practical example is given for a gearbox, for which the noise paths from the gears through the gear-supporting shafts and bearings into the gearbox housing has been experimentally assessed in terms of energy flows. This paper also discusses the accuracy and limits of the methodology.

1. Introduction

The characterization of vibration and noise sources is a challenging task required for the purpose of effective noise vibration and harshness (NVH) optimisation. The development of new analysis tools to investigate the vibroacoustic behaviour within the vehicle development process is of essential importance to achieve better products in combination with development time and cost reduction. There are several state of the art methodologies for the proper identification and assessment of both noise and vibration sources. Methodologies like lead covering (also referred to as shielding technique), surface vibration technique, acoustic intensity technique, transfer path analysis (TPA), acoustic holography and beamforming are experimental methodologies commonly applied in automotive industrial applications [1–6].

This paper discusses a methodology that applies energy flow analysis (EFA) for the NVH assessment and ranking of different system sources. An energy flow contribution analysis (EFCA) considers the different sources in terms of exchanged power with the system.

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The earliest study associated to Energy Flow Analysis (EFA) has been published by Lyon and Maidanik in 1961 [7]. They worked on vibration propagation in simple systems. This study later formed the basis for the SEA (Statistical Energy Analysis) technique. A good review of SEA method can be found in [8]. In 1980 Goyder and White proposed a broad theoretical study of structural energy flow transmission in flexible structures [9–11]. Their definition of energy flow (in their paper addressed as power flow) corresponds to the effective power that can be derived in harmonic state from the real part of the complex power. In their work, Goyder and White mainly investigated power transmission of waves in beams, plates and interaction between rigid substructures connected to flexible receiver by spring elements. They demonstrated that vibrational power is a parameter allowing to quantify and to compare, in an efficient way, the main effects associated to structural vibrations. In literature, energy and power problems were also further investigated and extended. Petersson and Plunt in 1982 defined a matrix formulation for the problem of structure-borne sound power transmission between multi-point coupled structures and investigated the importance of the estimation of the effective mobility [12,13]. To be cited is also the work of Mondot and Peterson, [14], that introduced two new quantities to fully characterise a source substructure and the transmitted energy flow: the source descriptor (representing the source ability to deliver power) and the coupling function (representing the portion of energy flow transmitted to the receiver). Miller et al. (1989, 1990) Applied EFA on structural networks identifying the components which contain the largest amount of vibrational energy, studying the energy flow at the junctions and its control modifying junction reflection and transmission properties [15,16]. Fulford and Gibbs studied the relationship between point and transfer mobilities for the analysis of structure-borne sound power and source characterization in multi-point connected structures [17–19]. In 2001, Moorhouse introduced three new concepts to characterise sources of structure-born sound [20]. He introduced the concept of mirror power, which is the power delivered by a vibration source when connected to a passive receiver structure that is a mirror image of itself. A second quantity, the characteristic power was defined as the dot product of the blocked force and free velocity vectors and this was shown to be four times the mirror power. In addition, Moorhouse also provided expressions for the maximum available power from a source. Bonhoff and Petersson investigated the importance and significance of Fourier coefficients for the characterization of vibrational sources [21]. On average their cross-order terms are found to be insignificant and can be neglected with good approximation. Therefore, few years later a method that does not consider cross-order terms was proposed by Mathiowetz and Bonhoff [22]. A broader review of EFA and its applications can be found in [23,24].

In this paper a gearbox-like structure is studied. A good review of the overall gearbox dynamic modeling can be found in [25,26]. The power transmitted through the different shaft bearings into the gearbox structure is experimentally investigated. In literature, similar problems were already examined. Typically, bearings have been modeled as ideal boundary conditions or scalar spring dampers. More complex models consider a stiffness matrix of dimension 6 to describe motion or force transmissibility across rolling elements [27]. This matrix formulation was also incorporated into an overall numerical calculation procedure to describe the related energy flow problem [28]. This numerical methodology requires a prior knowledge of system matrices (shafts and gearbox mass, stiffness and damping matrices) and bearings are modeled as massless joints characterised by stiffness and damping properties.

Here, a methodology to experimentally measure the power transmitted through the bearings is proposed. The main issue is related with bearing moment estimation and moment frequency response function (FRF) determination. For moment excitation a number of options exist, for example, twin shaker arrangements [29], blocks [30–32], magnetostrictive exciters [33,34] and synchronized hammers [35]. Other methods such instead aim to avoid direct moment excitation altogether [36–38]. The moment acting on bearing is here estimated using a finite difference technique [39].

Once excitation and velocities are determined, energy flow analysis is applied to the target structure and the power contributions of the different bearings (here considered as the energy sources of the gearbox structure) are assessed and ranked. The aim of this study is to characterise and rank the different sources of the system using power quantities, giving more insight on the vibration generation and transmission mechanisms.

2. Theoretical basics

2.1. General formulation of a dynamic system

Considering a mechanical system, the equation describing its dynamics due to external harmonic excitations is

$$\boldsymbol{M}\ddot{\boldsymbol{q}}(t) + \boldsymbol{D}\dot{\boldsymbol{q}}(t) + \boldsymbol{K}\boldsymbol{q}(t) = \boldsymbol{f}e^{i\omega t} \tag{1}$$

where $\boldsymbol{M}, \boldsymbol{D}$ and $\boldsymbol{K}$ are respectively mass, damping and stiffness matrices of the system, $\boldsymbol{q}$ is the vector of local translational and rotational displacements of all points of the considered body and $\dot{\boldsymbol{q}}$ and $\ddot{\boldsymbol{q}}$ are respectively its first (velocities) and second time derivative (accelerations). $\boldsymbol{f}$ is the vector of external forces or moments directly applied on the considered system. Moreover, i is the imaginary unit and ω is the angular frequency of the harmonic signal considered. For harmonic motion, assume a harmonic solution of the form

$$\boldsymbol{q}(t) = \boldsymbol{u}(\omega)e^{i\omega t} \tag{2}$$

where $\boldsymbol{u}$ is a complex displacement vector. Taking the first and second derivative of Eq. (2) and introducing into Eq. (1), it is obtained