

Active Control of Gear Mesh Dynamics

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ABSTRACT

Gears are essential parts of many precision power and torque transmitting machines. However, the radiated intensive tonal noise due to the gear meshing is highly undesirable and annoying. In very severe cases, the gear vibrations can reduce the life and performance of the power transmitting components. Typical gearbox vibration and sound spectra contain several dominant narrowband tonal signals that are mixed in with a lower level broadband response signals. Hence, the control of mesh response of gearbox housing belongs to the problem of the rejection or cancellation of periodical disturbance. The frequencies of these tonal signals are related to the number of teeth and rotation speed, and highly predictable. Thus, a feedforward control system was normally adopted. In most of existed applications, an accurate reference based on the frequency information of tachometer pulse train signal is required for this kind of control system. However, in the proposed ANC system, the feedback control with internal model control structure is used. The controller is designed using the inverse modeling technique. In this paper, numerical simulation was conducted based on the recorded gear housing vibration responses and measured dynamic response of the actuation system from an actual high-density gearbox system. Simulation results show that a promising performance of the proposed control system can be achieved at the target gear mesh frequencies.

INTRODUCTION

Gears are essential parts of many precision power and torque transmitting machines. However, the transmission error resulting from tooth profile errors, misalignment and elastic deformation [1,2] will normally excite the noise and vibration in power transmission systems that tend to be annoying and reduce the life and performance of the machine. The resultant

gearbox vibration and noise are often dominated by tonal peaks that occur at the fundamental mesh frequency, which is given by the product of the shaft speed and number of gear teeth, and its harmonics. Thus, it is highly desirable to control these gear mesh excitations.

The traditional way to address this problem in the past is to apply passive noise and vibration control approaches such as tuning of the geared rotor system modal characteristics, damping treatment, and vibration isolation. Active control approach is an alternative treatment for the noise and vibration problems. However, the application of active control on the gearbox system is still limited. One of the reasons for the limited applications of active control is the complexity of the mechanisms responsible for the gear mesh excitation. Nonetheless, there are a few noteworthy contributions in the literature. One of the first published works was reported by Montague et. al. [3] in 1994. In that study, an analog phase shifter with amplifier was used to drive a piezoelectric actuator mounted onto one shaft in the gearbox. Up to 70% reduction in gear mesh vibrations was achieved in their experimental test. However, in that work, only the fundamental mesh frequency was controlled. Furthermore, the proposed phase shifter and amplifier had to be manually adjusted for each running condition, which is not very convenient for most applications.

In 1999, Rebbechi et.al. [4] proposed an active control approach that used a pair of magnetostrictive actuators mounted at one of the support bearing locations to isolate the shaft vibrations from the housing. The experimental results showed 20~28 dB reductions at the fundamental mesh. However, the higher harmonics levels were only reduced by 2~10 dB. Later, Chen and Brennan [5] developed an active vibration control scheme that uses 3 magnetostrictive actuators mounted directly onto one of the gears. With this

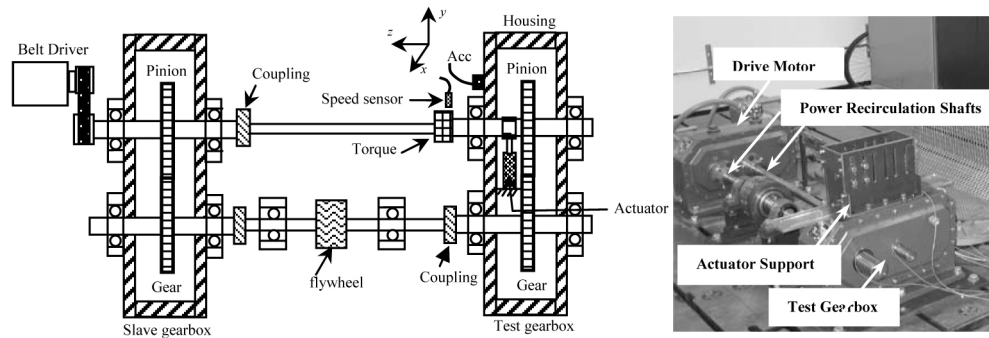


Figure 1. Sketch and picture of the testing power circulating gearbox system

design, the controller can produce circumferential forces that suppress steady-state torsional vibrations. Their experimental results demonstrated a reduction in gear angular vibrations of about 7 dB at the fundamental mesh frequency of 250 Hz.

In 2003, Guan et. al. [6] proposed a control system using several inertial actuators on both gear bodies with a direct hybrid adaptive controller that simultaneously tackles gear pair torsional and translational vibrations. The numerical studies of their work showed the effectiveness of the proposed controller in achieving substantial reduction in gear mesh vibrations. In that design, the use of complex slip rings are required to transfer power signals to the rotating actuators. Furthermore, other major structural modifications are required, which may not be entirely feasible in practice. In 2004, Guan et.al. [7] examined several different actuation concepts from the viewpoint of the required control forces and amplifier power. In that work, it was concluded that active shaft transverse vibration control is one of the most optimum setups. Later, the active shaft transverse vibration control system was tested by Guan et. al. [8-9] on a small single-stage power-absorbing gearbox. In those studies, both the delay-x LMS algorithm and the filtered-x LMS (FXLMS) algorithm were applied in the controller design. Results showed more than 20 dB of reduction for the first two harmonics. Later, the developed active shaft transverse vibration control setup was further applied to a closed-loop, re-circulating power, gearbox system that operates at a much higher power density [10].

In most of the reviewed research work mentioned above, a feedforward control with FxLMS algorithm is adopted. In this paper, to minimize the cost of the active control system by using only error sensor, a feedback control system with internal model control (IMC) arrangement is used.

CONTROL SYSTEM DESCRIPTION

TESTING GEARBOX SYSTEM

Figure 1 shows the cross-sectional view illustrating the inner details of the closed-loop and power re-circulating gearbox setup. Right side of the figure is the picture of the

experimental gearbox system. The system consists of a belt driver, a torque transducer, a speed sensor, a clutch, a flywheel, one slave gearbox and one test gearbox. The right side gearbox is the test gearbox that contains a spur gear pair with a 1:1 tooth ratio. Each gear has identically 35 teeth, 25.4 mm (one-inch) face width and 152.4 mm (6-inch) pitch diameter. The pressure angle is about 20 degrees. The line-of-action of gear meshing force is along the pressure angle of the spur gear pair. To separate the responses due to test gear meshing from the responses caused by slave gear meshing, the gears of slave gearbox have different teeth number which is 46 from test gear set. To simulate the high power density without the need for a high power driver, the gears can be preloaded by adding torque to one of the shafts coupling the two gearboxes. Of course, this will also create a matching static torque in the other shaft. The flywheel and couplings are used to isolate vibrations in the slave gearbox from the test gearbox.

One tri-axial accelerometer is used as error sensor in the system (labeled as Acc) for monitoring the housing vibration level and attached to the gearbox housing just above the driven shaft bearing location facing the slave gearbox. The x and y axes of the accelerometer are perpendicular and parallel to the gear mesh line-of-action, respectively. The angle between the y direction and vertical is adjusted to 20 degrees, which is identical to the pressure angle. The y-axis signal from the accelerometer, denoted as Accy, is used as the error signal for the controller. The actuation method includes a piezoelectric stack actuator connected to the driven shaft via a needle bearing. The actuator is driven by a high voltage amplifier that receives command signals from the controller. Note that the actuation direction is parallel to the gear mesh line-of-action, such that the active control force is better able to disrupt mesh force generation and vibration transmissibility. Also, in the mounting of the actuator, a proper preload level is set to avoid problems associated with bearing clearances as well as gear eccentricity. Furthermore, to compensate for possible run-out and excessive shaft deflection that can generate harmful lateral forces to the actuator, a thin stinger rod that is stiff in the longitudinal direction and relatively flexible in the transverse direction is