

Dynamic Modeling and Analyses of Mechanical Transmission Systems on A Test Platform

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Abstract—To investigate the backlash phenomenon vibration and noise of a gear transmission system, we compared the performance of two difference types of mechanical transmission performance system: worm gears reducer (DYNABOX) and planetary gear reducer (APEX). We analyzed two kinds of models and harmonic response and evaluated the performance of DYNABOX and APEX reducer based on 3D model and simulation technology. Through the nonlinear dynamic modeling of the transmission system, we established a comparison between DYNABOX and APEX reducer. From the result of our test platform, we show that the start reduction of APEX reducer in the gears vibration at the mesh total deformation harmonic response is bigger than DYNABOX reducer at the first operation gear mesh, and then stability. Furthermore, the active vibration observed system yield in the planetary gear compared with a worm gear by means of total deformation, stress, strain, and acceleration; shows backlash. Our results revealed that APEX would be an attractive MTPS as compared to DYNABOX.

Keywords- Platform; mechanical transmission; backlash; vibration; finite element

I. INTRODUCTION

Test platforms are widely used as research tool and in the industrial engineering. The mechanical transmission performance systems (MTPSs) are being investigated for different characteristics e.g., vibration, noise and gear meshed (backlash). Better performing MTPSs are required at low cost and fair production along with suitable prolonged use.

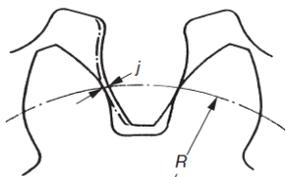


Figure 1. Backlash between two gears

In order to overcome the restrictions associated with the static test platform method, a dynamic detection test for detecting gear backlash is suggested. In this test platform, the use an AC motor actuator to rotate the gears during a test model causes the gear teeth to disengage, and reengage several times over some part of the gear teeth pairs during the performance. It is premised that an actuator variable motor speed is sensitive to gear tooth influence. Further, it may be used to determine the amount of backlash in the gear reducer system[1] and consequently, vibration, resonance, and noise[2] as well.

Backlash is one a problem encountered non-linearities in transmission systems employing gears, and refers to play between two mating gears may be represented by one pair of teeth[3]-[4], backlash is usually represented by the schematic shown in Figure 1. When used in the situation of MTPSs, backlash symbolizes two prominent features as shown in Figure 2. A mechanical hysteresis to the existence of a clearance[5]. However, to definition, the backlash defect into the model, it acts directly on the elastic meshing force. The tooth contact loss is modeled by the lack of continuity of the knowledge meshing force following the line of action. [6]

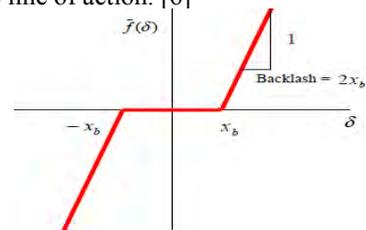


Figure 2. Clearance gear backlash non-linearity

In this paper, we will achieve on the effect of the backlash on the multi degree of freedom, non-linear dynamic gear transmission system on a test platform, incorporating vibration and noise. The relation between gears central distance error, and backlash is given and the

dynamic backlash is defined and a six degrees of freedom nonlinear dynamic gear transmission system is developed with dynamic backlash. The computer software's named as SOLIDWORKS13 and ANSYS WORKBENCH15 were used to development mechanical transmission performance a test bench to investigate. A neutral mode and dynamic responses are compared on the condition with two kinds of gear reducers.

II. MODELING AND DESIGN STRUCTURE ON A TEST PLATFORM

A. Modeling

On a test platform, the proposed model of backlash detection a system is modeled consisting of:

- (i) An AC electric servo system motor (YASKAW motor),
- (ii) An external motor load (LINGZ),
- (iii) A worm gear reducer (DYNABOX),
- (iv) A planetary gear reducer (APEX),
- (v) A Coupling to the gear reducer(PLUM),
- (vi) A Coupling to the external load(FLEXIBLE)
- (vii) A rigid body linkage.

B. Design structure

The SOILDWORKS13 software structure 3D modeling on a test platform is used to design the structural analysis. The model of (MTPS) on a test platform is easy to operate as well as easy to convert a layout. For the investigation of vibration and noise from the gear reducer and to get data at different torque speed is convenient. However, the vibration and noise of the gear comes from the backlash problem. Backlash considered is as a source of non-linearity arising from the contact interaction among the gear tooth.

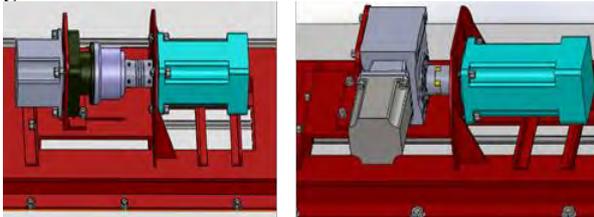


Figure 3. Liner layout and Vertical Layout

The servo-motor YASKAWA is used (Figure 3.) to excite the gear reducer that consists of the DYNABOX reducer (linear layout) and APEX reducer (vertical layout), and inverted load motor generate LINGZ. The PLUM and FLEXIBLE coupling is used both at the driver and inverse sides. A schematic linear layout and vertical layout of such a system are given in Figure 3. respectively. The performance test includes drive vibration and noise at a different torque speed. In this design, YASKAWA motor has speed of 3000 rpm; output shaft diameter is 11mm. The input shaft of the DYNABOX gear reducer is a hollow shaft with a key way. DYNABOX reducer presence on the tooth surface contacts results in a high accuracy. Additionally, on the other side of gear reducer there is a key for adjusting the distance (anti-backlash) between worm gear, and wheel. While, APEX design like a double gear anti-backlash, for accuracy, and not allow to clearance between tooth. Moreover, planetary (APEX) gear train

used a helical tooth contact to excess the area, smooth operation, and low noise.

III. STRUCTURAL DYNAMICS ANALYSIS SYSTEMS

Generally, we are cooperation with equation of multiple-degrees-of-freedom systems[7].

$$[M][\ddot{D}] + [C][\dot{D}] + [K][D] = [F] \quad (1)$$

In equation (1), [D] is the displacements vector for a problem of n degree of freedom donated by [Di], i=1,2,...,n., [F] is the external forces vector, [M] is the mass matrix, [C] is the damping matrix, and [K] is the stiffness matrix.

Equation (1) represents the governing equation of a transient structural simulation. The right hand side of the equation is the external force [F] and the first item of the left hand side of the equation, is inertia force, is damping force, and is the elastic force.

A. Modal Analysis

We analyzed a free vibration; there is no involvement of the external force [F]. So, equation (1), becomes

$$[M][\ddot{D}] + [C][\dot{D}] + [K][D] = 0 \quad (2)$$

The equation (2) is used for a problem having n degrees of freedom. Our model has six degrees of freedom; it has at most six solutions of the fundamental natural frequency (ω). In a model analysis, we are interested in the natural frequencies and the relative shapes of the vibration modes. However, the damping effect can be neglected in the equation (2), and get equation 3.

$$[M][\ddot{D}] + [K][D] = 0 \quad (3)$$

We used equation (3) in the workbench solves in a modal analysis system to find out the natural frequency for the six mode of freedom of the gear reducer.

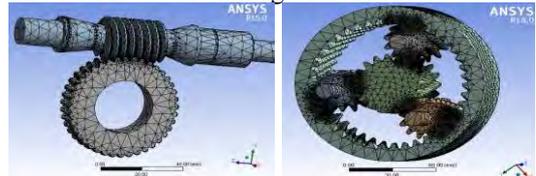


Figure 4. Worm gears reducer mesh (DYNABOX) and Planetary gear reducer mesh (APEX)

For the investigation of fundamental natural frequency, the method of solid modeling is used in ANSYS in which the different component of the model has been established. Figure 4. shows worm gears reducer (DYNABOX) and planetary gear reducer (APEX) meshing model respectively.

TABLE I. THE NATURAL FREQUENCY OF THE PLANETARY GEAR AND WORM GEAR REDUCER.

Natural frequency [Hz]	DYNABOX reducer	APEX reducer
mode 1	7450.8	7613.5
mode 2	10394	7937.7
mode 3	10418	8003
mode 4	12066	8755.6
mode 5	15563	9982.4
mode 6	20716	10422

With the use of method of solid modeling in ANSYS, we find out the six freedom natural frequencies of six resolving parts of worm gears reducer (DYNABOX) and planetary gear reducer (APEX) given in Table I. Figure 5.and Figure 6.are the graphical representation of the six

freedom natural frequencies of six resolving parts of worm gears reducer (DYNABOX) and planetary gear reducer (APEX) respectively. We use the higher natural frequencies individually of each reducer further in harmonic response model.



Figure 5. Natural frequency of the DYNABOX



Figure 6. Natural frequency of the APEX

B. Harmonic Response Analysis

The maximum torque speed of the gear reducers is 2.7 Nm. The range of operation speed of model is from zero to 3000 rpm, maybe excited by the harmonic force. Vibrations of the structure components are unavoidable, and large amplitude of the vibration will happen.

For harmonic response analysis, the external force on i th degree of freedom is in the following form equation (4), in which A_i is the amplitude of the force, i the face angle of the force, and α is the angular frequency of the external force.

$$F_i = A_i \sin(\alpha t + \theta_i) \quad (4)$$

Calculations are too much difficult due to the special form of the external forces; so finally, the steady state solution of equation (4) is in the form of equation (5).

$$D_i = B_i \sin(\alpha t + \phi_i) \quad (5)$$

In harmonic response analysis, we can find out the magnitude B_i , and the face angle, and the response of each degree of freedom.

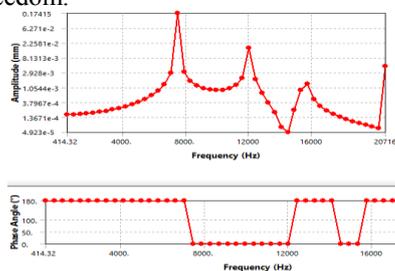


Figure 7. Frequency response total deformation DYNABOX

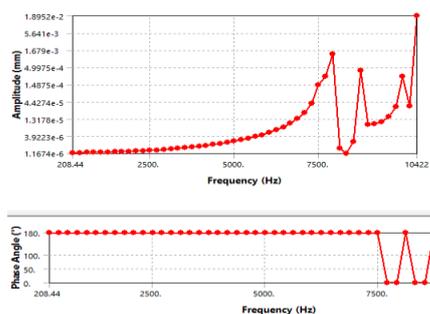


Figure 8. Frequency response total deformation of APEX

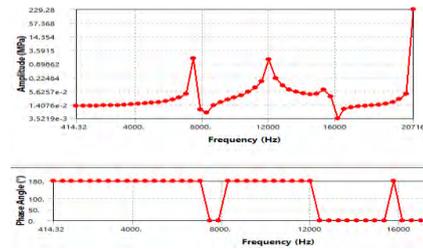


Figure 9. Frequency response stress of DYNABOX

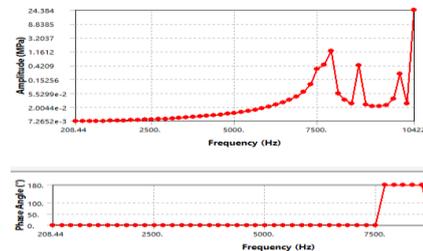


Figure 10. Frequency response stress

In the first simulation and implementation, workbench analysis is applied to compare two kinds of gear reducer on the test platform. The step size value is set for individual reducer, because that depends on the neutral mode analysis. Frequency-domain identification is used to obtain the best reducer. The simulation results in directions X in harmonic response analysis.

First amplitude and frequency for the worm gear and planetary gear reducer observed are 0.17415 mm at 7457Hz, and 2.5118e-003mm at 8754.5Hz (face angle 0) and the effect of harmonic response by total deformation of both gear reducers is shown the Figure 7.and Figure 8.respectively.

The vibration power attenuation is only achieved at frequency response stress. Similarly, the amplitude and frequency values are about 1.6101Mpa at 7457.8Hz and 1.2333Mpa at 7920.7 Hz (face angle 0) for the worm gear and planetary gear reducer. The harmonic response stress for the DYNABOX and APEX gear reducer are shown in Figure 9.and Figure 10.respectively.

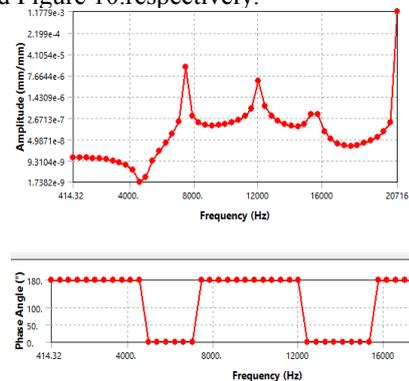


Figure 11. Frequency response strain of DYNABOX

At the dancing frequency of 7884Hz, the model is excited such that the maximum displacement is 9.343e-5 mm/mm (face angle 0) for DYNABOX gear and for the APEX gear reducer, the maximum displacement 2.333e-5mm/mm (face angle 0) is observed at dancing frequency 7800Hz. Figure 11.and Figure 12.shows the frequency

response strain of the DYNABOX gear and APEX gear reducer respectively.

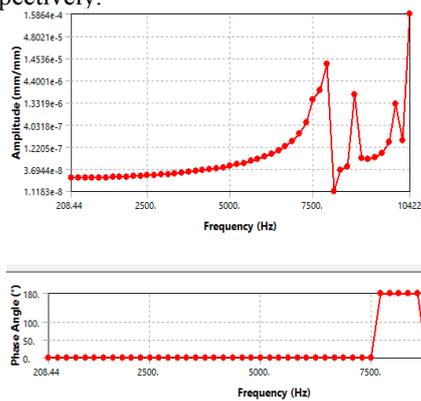


Figure 12. Frequency response strain of APEX

Finally, amplitude and frequency for the worm gear and planetary gear reducer observed are $3.8238e008\text{mm/s}^2$ at 7457.8Hz (face angle 0) and $3.1336e006\text{mm/s}^2$ at 7920.7Hz (face angle 180) and the frequency response acceleration of the DYNABOX gear and APEX gear reducer respectively are shown in the Figure 13. and Figure 14..

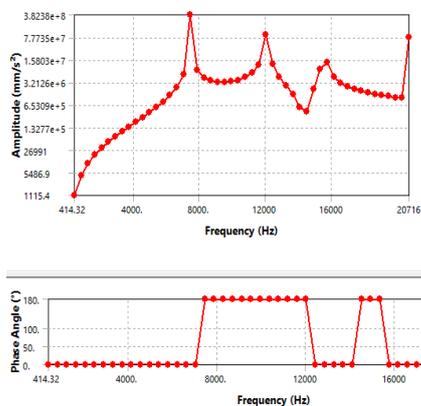


Figure 13. Frequency response acceleration of DYNABOX

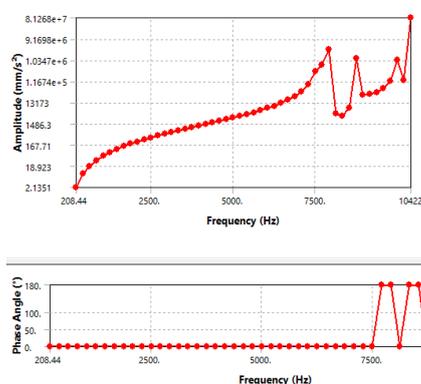


Figure 14. Frequency response acceleration of APEX

IV. CONCLUSION

Previously, the active gears reducer transverse vibration concept has been formulated on theoretical basis. In present work as a test platform, we improved the gearbox structure, design and simulation by applying the SOLIDWROKS13 and ANSYS WROKBENCH15. For a MTPS, the gearbox system is employed as a test platform

to evaluate the feasibility of a model. In order to compare two kinds of gear reducer, a mode analysis and harmonic response was adopted to assess the backlash, vibration and noise. The test platform result shows that the start reduction in the gears vibration at the mesh total deformation harmonic response is bigger in the APEX than DYNABOX reducer at the first operation gear mesh, and then stability. Furthermore, the active vibration observed system yield in the planetary gear compared with a worm gear by means of total deformation, stress, strain, and acceleration; shows backlash. Hence, from the analysis of vibration and noise at the first and continue operating gear mesh frequencies, it is proved that the planetary gear (APEX) reducer is better.

In future work, we will use the mechanical transmission performance system as a platform to assess the APEX gear and DYNABOX gear reducer on the basis of gear vibration and noise suppression, capabilities at higher operating speeds and loads.

ACKNOWLEDGMENT

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