

Investigation of Characteristics of Gear Meshing Noise under Three-Axis Driving Planetary Gear Set

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ABSTRACT

Planetary gear sets are widely used in hybrid or electric vehicles. Although they have many advantages, they have been known to develop noise and vibration problems. There have been many reports dealing with the problems' characteristics for two-axis driving, but very few for three-axis driving. This paper reports three-axis driving performed on a developed driving stand, with driving torque measured by a high response torque sensor on each driving axis. Under these conditions it was found that the meshing frequency occurred at two kinds of frequency harmonics.

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INTRODUCTION

Hybrid and electric vehicles have recently been attracting increasing attention due to global warming and the drying up of fossil fuels [1]. Both vehicle types use planetary gear sets; hybrid vehicles use them as the power distribution system and electric vehicles as a high reduction system. However, planetary gear sets are complex devices and consequently their dynamic characteristics are not fully understood.

Many reports have cited the problem of planet gear displacement occurring in two-axis driving. Parker et al. noted the structured vibration modes of general compound planetary gear systems [2], used analytical and finite element models to ascertain the nonlinear dynamics of planetary gears [3], and conducted planetary gear modal vibration experiments to identify the correlation against lumped-parameter and finite element models [4]. Vex et al. reported on the dynamic simulation of eccentricity errors in planetary gears [5] and on a dynamic model for studying the effects of planet position errors in planetary gears [6]. Torque sharing of planet gears based on FEM [7] has also been reported. However, only a few reports have dealt with dynamic characteristic in three-axis driving.

In this paper, we report three-axis driving performed on a developed driving stand, with driving torque measured by a high response torque sensor on each driving axis. In particular, we focus on characteristics of gear meshing frequency components due to planet gear set driving.

We observed that the meshing frequency under three-axis driving occurred at two kinds of frequency harmonics, caused by the meshing relationship between ring and planet gear and between planet and sun gear. Comparing these results, we constructed a method of calculating their frequencies on the basis of instantaneous rotation center theory. Moreover, the meshing frequency component of the ring gear was found to be pure sound noise due to its high inertia and rotational stability, and that of the sun gear was found to be mixed sound noise with side bands due to its low inertia and rotational instability.

PLANETARY GEAR SET

A planetary gear set is shown in Fig.1. It has four basic components: ring gear, sun gear, planet gears, and a carrier that, connects the planet gears. Suffixes s, c, r, p correspond to the sun gear, carrier, ring gear, and planet gears. The rotational speed of each respective shaft is defined N_s, N_c, N_r and N_p . The input can be given through three components excluding the planet gears. A planetary gear set can provide a multi gear ratio by changing the input and output axis and deal with multi-inputs or multi-outputs at the same time. However, planetary gear sets have a complicated structure and consequently their dynamic characteristics are not fully understood.

Planetary gear sets are used for either a three-axis or a two-axis driving. Three axis driving is general driving condition of planet gear set. In three-axis driving, the sets deal with 1 input-2 outputs or 2 inputs-1 output; combining power from engine and motor to a vehicle, or distribute engine power to a vehicle and a generator. Three-axis driving can provide variable gear ratio by controlling multi inputs. Furthermore, there is a fixed rotation mode in which all

components except the planet gears rotate at the same speed in the same direction like a rigid body ($N_r = N_c = N_s$, $N_p = 0$). On the other hand, two-axis driving condition has one fixed component and it is same as ordinal gear set; it provides single gear ratio, single planet gear set can provide multi different gear ratio by changing its in and output. In two-axis driving, there are three fixed conditions. The ring gear-fix ($N_r = 0$) condition is called the planet mode, the carrier-fix ($N_c = 0$) condition is called the star mode, and the sun gear-fix ($N_s = 0$) condition is called the solar mode. Three-axis driving is considered to be a combination of these three fixed conditions.

In this paper, 2 inputs from the ring gear and carrier and 1 output to the sun gear are taken into consideration.

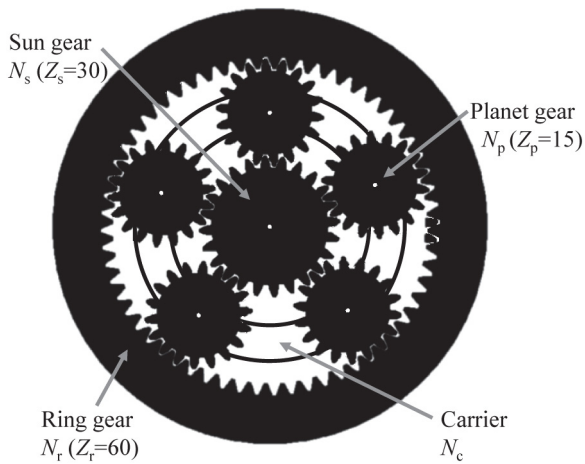


Fig.1. Planetary gear set

USAGE OF PLANETARY GEAR SET

Automatic Transmission

Planetary gear sets are widely used in stepped automatic transmissions with their advantages like multi-gear ratio, compactness, and in and out on the same axis. A single planetary gear set ordinarily provides a two-gear ratio in a stepped automatic transmission: Another ratio is used for reverse rotation. Three planetary gear sets are used in 5-speed and 6-speed transmissions. In every transmission type, planetary gear sets are used for two-axis driving. Recently, the number of automatic transmission speeds has been increasing from an efficiency aspect, in order to use the high efficiency area of internal - combustion engines, or from a friction aspect to enable down speeding. ZF has released the 9-speed transmission 9HP, which is used in Range rover [8].

Series-Parallel Hybrid System

Toyota uses planetary gear sets as the core of series-parallel hybrid system to combine engine power and motor power to achieve higher efficiency. In such system, planetary gear sets are used in a very complex way, not only for two-axis driving to transmit engine power to the car as with an ordinary automatic transmission, or to transmit braking energy to the generator to make electricity (called kinetic energy recovery brake), but also for three-axis driving functions like

transmitting combined power from the engine and motor (called power mode), or distributing engine power to the car and generator (called charge mode). A 5-planet planetary gear set is widely used in such system. Control of the planetary gear set is the key for achieving higher efficiency. At the same time, however, the noise generated by planetary gear set has emerged as a problem due to the quietness of hybrid vehicles.

Differential Gear in Formula1

A planetary gear set is used as the differential gear in Formula1 due to its compactness, light weight and high torque capability. This is a very special case and a clutch is required to lock it up as a limited slip differential (LSD). The ring gear is the input axis and the carrier and sun gear are the output axis for the left and right wheels, respectively. A 4-planet planetary gear set is widely used due to its balance of torque capability and weight. Its rigid rotation is used when the vehicle is running on the straight or if there is no difference between the right and left wheels, and it is used in three-axis driving during the cornering or when there is difference between the right and left wheels.

PLANETARY GEAR SET THEORY

Relation of Component Speeds

The speed of each component is determined by synthesis of two kinds of rotation modes, i.e., star mode and rigid rotation mode, as below.

$$N_s : N_p : N_r : N_c = 1 : -\frac{Z_s}{Z_p} : -\frac{Z_s}{Z_r} : 0 \quad (1)$$

$$N_s : N_p : N_r : N_c = 1 : 0 : 1 : 1 \quad (2)$$

In these equations, N_r is determined as a positive value and a negative value means opposite rotation direction. Eq. (1) is for star mode and Eq. (2) for rigid rotation mode. The relation of rotation speed can be calculated by solving these equations with the undecided coefficient method.

$$\begin{cases} N_p = \frac{Z_s Z_r}{Z_p (Z_s + Z_r)} (N_r - N_s) = \frac{Z_r}{Z_p} (N_r - N_c) \\ = \frac{Z_s}{Z_p} (N_c - N_s) \\ N_c = \frac{1}{Z_s + Z_r} (Z_s N_s + Z_r N_r) \end{cases} \quad (3)$$

As shown in Eq. (3), the rotation speed of planetary gear N_p is determined by the difference in rotation speeds between any two other components and the gear ratio of the planetary gear set. From this equation, the rotation condition can be divided into four states as shown in Table 1.

Table 1. Range of speed

N_c/N_r	a	$<0<$	$b<$	$\frac{Z_r}{Z_r+Z_s}$	$<c$	$<1<$	d
N_r	+	+	+	+	+	+	+
N_c	-	0	+	+	+	+	+
N_p	+	+	+	+	+	0	-
N_s	-	-	-	0	+	+	+

In Table 1, the ring gear rotation direction is defined as a positive value, i.e., described as +. When the input ratio is $N_c/N_r=0$, it is in star mode, and when it is $N_c/N_r=1$, it is in rigid rotation mode. When it is $N_c/N_r=Z_r/(Z_r+Z_s)$, it is in solar mode and no output exists although there are two inputs. In this case, its $Z_r/(Z_r+Z_s)$ value can be controlled by the gear ratio.

According to Table 1, meshing forces around the planet gear can be described as in Fig.2.

A simple lumped parameter model (Appendix 1) is used in Fig.2. The rotation center O is on the XY plane, the rotation center of the planet gears is O_p , its tangential direction is X_p , and its normal direction is Y_p . The pitch radius R is defined as in Eq. (4) below, with module m and the number of teeth Z .

$$R=mZ/2 \quad (4)$$

The meshing force F on the meshing point is defined as in Eq. (5) below, with torque T .

$$F=T/R \quad (5)$$

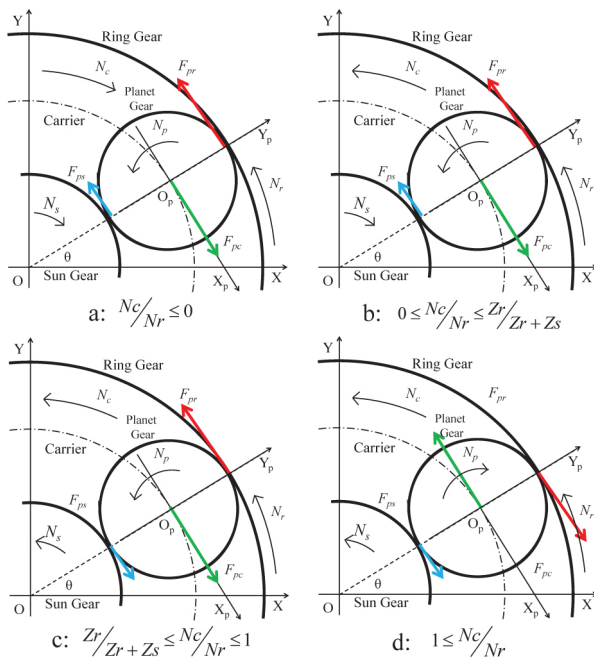


Fig.2. Model of forces

In the figure, F is nominal meshing force, i.e., the force tangential to the pitch circle. The former suffix is the element that is given the meshing force and the latter suffix is the one that gives the meshing force. The respective forces from the ring gear, sun gear and carrier to the planet gears are F_{pr} , F_{ps} , and F_{pc} . In each, transitional force should be balanced during constant driving.

Instantaneous Rotation Center

In the three-axis driving, the planet gears not only rotate but also revolve at the same time. The velocity distribution of the planet gears' rotation is shown in Fig. 3. In this paper, we define tangential speed as a positive value for the CCW (Counter Clockwise) of planet gear. In the figure, V_{sp} is the velocity at the sun gear-side and V_{rp} is that at the ring gear-side of the planet gears turning at the speed of N_p . The ratio of these two is given using Eq. (6).

$$V_{sp}:V_{rp}=-1:1 \quad (6)$$

This shows that the velocity at the two sides is the same but the direction is opposite.

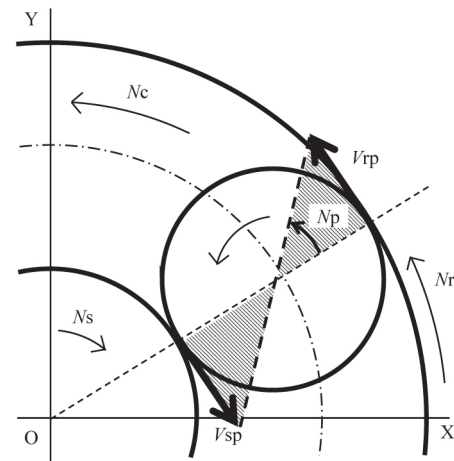


Fig.3. Velocity distribution of planet gear rotation

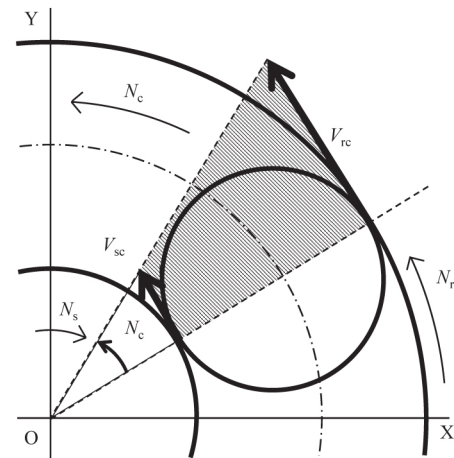


Fig.4. Velocity distribution of planet gear revolution

Furthermore, the revolution speed varies according to the distance from the rotation center O, as shown in Fig. 4. The revolution of the planet gears is equal to the rotation of the carrier.

In the figure, V_{sc} is the velocity at the sun gear-side and V_{rc} is that at the ring gear-side of the carrier turning at the speed of N_c . The ratio of these two is given using Eq. (7).

$$V_{sc}:V_{rc}=R_s:R_r=Z_s:Z_r \quad (7)$$

This shows that V_{rc} is faster than V_{sc} because of the difference in the pitch radius or the number of teeth.

When a planet gear rotates and revolves at the same time, the velocity distribution is like that shown in Fig. 5, or the combination of that shown in Figs. 3 and 4.

The ratio of V_s and V_r , respectively the sun gear-side and ring gear-side velocity of the planet gear, is given using Eq. (8).

$$\begin{cases} V_r = V_{rp} + V_{rc} = R_p \omega_p + R_r \omega_c = \frac{2\pi}{60} \{R_p N_p + R_r N_c\} \\ V_s = V_{sp} - V_{sc} = R_p \omega_p - R_s \omega_c = \frac{2\pi}{60} \{R_p N_p - R_s N_c\} \end{cases} \quad (8)$$

Instantaneous rotation center O_p' is the point at which its velocity is zero on the Y_p axis. The planet gear rotates around O_p' instantaneously.

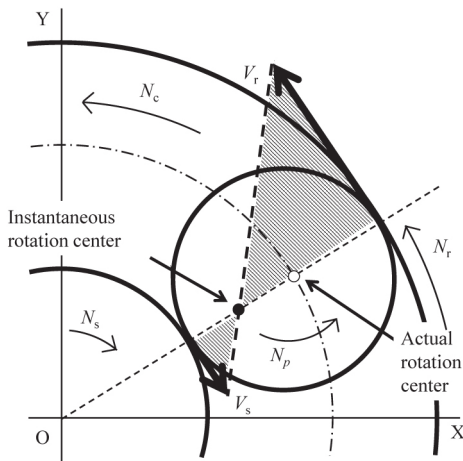


Fig.5. Velocity distribution of planet gear

Then, if the difference in tangential speeds between the planet gear and the other gears, or ring gear and sun gear, is taken into consideration,

$$\begin{aligned} V_r - \frac{2\pi}{60} R_r N_r \\ = \frac{2\pi}{60} (R_p N_p + R_r N_c - R_r N_r) = 0 \end{aligned} \quad (9)$$

$$\begin{aligned} V_s + \frac{2\pi}{60} R_s N_s \\ = \frac{2\pi}{60} (R_p N_p - R_s N_c + R_s N_s) = 0 \end{aligned} \quad (10)$$

Eq. (9) and (10) show that the tangential speed at the meshing point is the same and that proves nonslip at the meshing point.

Distance R' between O and instantaneous rotation center O_p' is defined with V_r and V_s in Eq. (11).

$$R' = \frac{V_s - V_r}{V_s + V_r} R_p + (R_p + R_s) \quad (11)$$

The ratio α , i.e., that of R' to the actual rotation center of the planet gear is calculated in Eq. (12).

$$\begin{aligned} \alpha = \frac{R'}{R_p + R_s} &= \frac{V_s - V_r}{V_s + V_r} \frac{R_p}{R_p + R_s} + 1 \\ &= \frac{-(R_r + R_s)N_c}{2R_p N_p - (R_r + R_s)N_c} \frac{R_p}{R_p + R_s} + 1 \end{aligned} \quad (12)$$

Here, R_p and R_s are the pitch radius of the planet gear and sun gear, respectively. The α values calculated with the input speed ratio of the ring gear and carrier are shown in Fig.6.

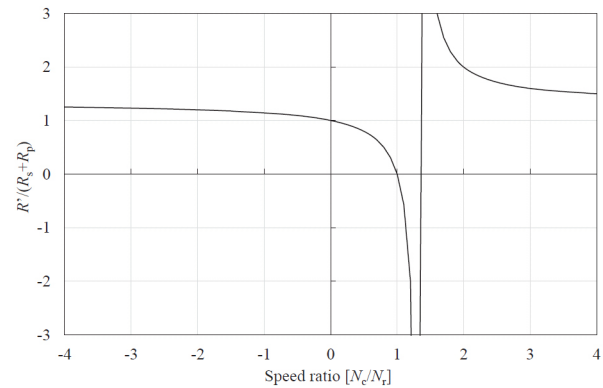


Fig.6. Instantaneous rotation center

There is an asymptote in Fig.6 and when the speed ratio is close to it, α goes to infinity as shown in Eq. (13).

$$\begin{cases} \lim_{\frac{N_c}{N_r} \rightarrow \frac{2Z_r}{Z_r + Z_s} + 0} \alpha = +\infty \\ \lim_{\frac{N_c}{N_r} \rightarrow \frac{2Z_r}{Z_r + Z_s} - 0} \alpha = -\infty \end{cases} \quad (13)$$

When the input ratio is at the exact ratio of the asymptote, the velocity distribution of planet gear is calculated as below.