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High Gear Ratio Epicyclic Drives Analysis

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[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

Epicyclic gear stages provide high load capacity and compactness to gear drives. There is a wide variety of different combinations of planetary gear arrangements [1, 2]. For simple epicyclic planetary stages when the ring gear is stationary, the practical gear ratio range varies from 3:1 to 9:1. For similar epicyclic planetary stages with compound planet gears, the practical gear ratio range varies from 8:1 to 30:1.

This paper presents analysis and design of epicyclic gear arrangements that provide extremely high gear ratios. Using differential-planetary gear arrangements it is possible to achieve gear ratios of several hundred to one in one-stage drive with common planet gears and several thousand to one in one-stage drive with compound planet gears. A special two-stage planetary arrangement may utilize a gear ratio of over one hundred thousand to one.

This paper shows an analysis of such uncommon gear drive arrangements, defines their major parameters, limitations, and gear ratio maximization approaches. It also demonstrates numerical examples, existing designs, and potential applications.

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Introduction

Epicyclic gear stages provide high load capacity and compactness to gear drives. There is a wide variety of different combinations of planetary gear arrangements [1, 2]. For simple epicyclic planetary stages when the ring gear is stationary, the practical gear ratio range varies from 3:1 to 9:1. For similar epicyclic planetary stages with compound planet gears, the practical gear ratio range varies from 8:1 to 30:1.

This paper presents analysis and design of epicyclic gear arrangements that provide extremely high gear ratios. Using differential-planetary gear arrangements it is possible to achieve gear ratios of several hundred to one in one-stage drive with common planet gears and several thousand to one in one-stage drive with compound planet gears. A special two-stage planetary arrangement may utilize a gear ratio of over one hundred thousand to one.

This paper shows: an analysis of such uncommon gear drive arrangements, defines their major parameters, limitations, and gear ratio maximization approaches. It also demonstrates numerical examples, existing designs, and potential applications.

One-stage arrangements

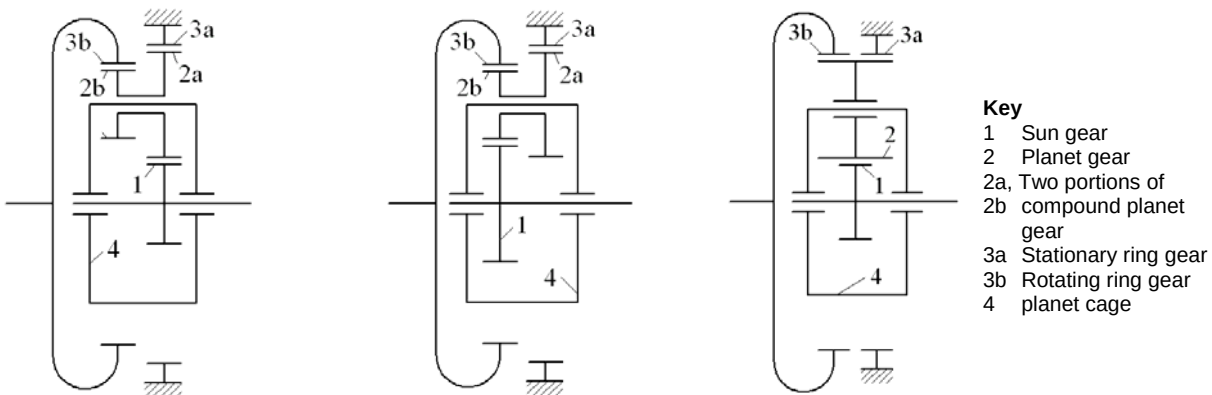
There are one-stage differential-planetary arrangements that provide much higher gear ratios. In these arrangements the output shaft is connected to the second rotating ring gear instead of the carrier, like in the epicyclic planetary stages. In this case a carrier does not transmit torque and it is called a cage because it is used just to support planet gears.

Figures 1a and 1b presents differential planetary arrangements with compound planet gears. In the arrangement in Figure 1a the sun gear is engaged with a portion of the planet gear that is in mesh with the stationary ring gear. In this case the gear ratio is:

$$u = \frac{1 + \frac{z_{3a}}{z_1}}{1 - \frac{z_{2b}z_{3a}}{z_{2a}z_{3b}}} \quad (1)$$

where

- u gear ratio;
 z_1 sun gear number of teeth;
 z_{2a} number of teeth the planet gear engaged with the sun gear and stationary ring gear;



a. Compound planet gears b. Compound planet gears c. Common planet gears

Figure 1. Differential-planetary arrangements

- z_{2b} number of teeth the planet gear engaged with the rotating ring gear;
- z_{3a} stationary ring gear number of teeth;
- z_{3b} rotating ring gear number of teeth.

In the arrangement in Figure 1b, the sun gear is engaged with a portion of the planet gear that is in mesh with the rotating ring gear. In this case the gear ratio is:

$$u = \frac{1 + \frac{z_{3a}z_{2b}}{z_1z_{2a}}}{1 - \frac{z_{3a}z_{2b}}{z_{3b}z_{2a}}} \quad (2)$$

If a gear ratio is negative, the input and output shaft rotation directions are opposite.

All gear meshes in differential planetary arrangements have the same center distance. This condition allows for definition of relations between the operating modules, m_w , or diametral pitches, DP_w . For the arrangement in Figure 1a they are:

$$m_{w12a}(z_1 + z_{2a}) = m_{w2a3a}(z_{3a} - z_{2a}) = m_{w2b3b}(z_{3b} - z_{2b}) \quad (3)$$

or

$$\frac{z_1 + z_{2a}}{DP_{w12a}} = \frac{z_{3a} - z_{2a}}{DP_{w2a3a}} = \frac{z_{3b} - z_{2b}}{DP_{w2b3b}} \quad (4)$$

The relation between operating pressure angles in the gear meshes $z_1 - z_{2a}$ and $z_{2a} - z_{3a}$ is defined by equation 5:

$$\frac{\cos \alpha_{w2a-3a}}{\cos \alpha_{w1-2a}} = \frac{z_1 + z_{2a}}{z_{3a} - z_{2a}} \quad (5)$$

where

- α_{w1-2a} operating pressure angle in a mesh of the sun gear and the planet gear engaged with the stationary ring gear;
- α_{w2a-3a} operating pressure angle in the planet/stationary ring gear mesh.

Similar to the arrangement in Figure 1b:

$$m_{w12b}(z_1 + z_{2b}) = m_{w2b3b}(z_{3b} - z_{2b}) = m_{w2a3a}(z_{3a} - z_{2a}) \quad (6)$$

or

$$\frac{z_1 + z_{2b}}{DP_{w12b}} = \frac{z_{3b} - z_{2b}}{DP_{w2b3b}} = \frac{z_{3a} - z_{2a}}{DP_{w2a3a}} \quad (7)$$

The relation between operating pressure angles in the gear meshes $z_1 - z_{2b}$ and $z_{2b} - z_{3b}$ is defined by equation

$$\frac{\cos \alpha_{w2b-3b}}{\cos \alpha_{w1-2b}} = \frac{z_1 + z_{2b}}{z_{3a} - z_{2b}} \quad (8)$$

where

- α_{w1-2b} operating pressure angle in a mesh of the sun gear and the planet gear engaged with the rotating ring gear;
- α_{w2b-3b} operating pressure angle in the planet/rotating ring gear mesh.

In differential planetary arrangements with compound planet gears, operating pressure angles in the planet/stationary ring gear mesh and in the planet the planet/rotating ring gear mesh can be selected independently. This allows for balancing specific sliding velocities in these meshes to maximize gear efficiency, which could be 80–90%, depending on the gear ratio [2]. The maximum gear ratio in such