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Experimental Investigation on Speed and Torque Analysis of Planetary Gearing System

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ABSTRACT

Gear is rotating circular toothed element which transmits the torque and speed to another toothed element, when it meshed with each other. Gear trains are widely designed to transmit torque and angular velocity between different shafts when there is a significant space reduction within a restricted area. The needed speed ratio and the location of the shaft axes suggest the type of gear train employed. In this research paper, efforts are taken for speed and torque analysis of epicyclical or planetary gear train. The average gear ratio 5.72 is obtained for various input speed of motor. Also, input torque, output torque and holding torque is determined by experimentation by employing rope brake dynamometer. Further, the analytical and experimental torque findings were compared, which showed the error less than 15%, as a result of certain mechanical and frictional losses.

Keywords: Epicyclic Gear Train, Holding Torque, Input Torque, Output Torque, Rope Brake Dynamometer

NOMENCLATURE

Symbol	Description	Unit
η	Efficiency of motor	%
GR	Gear Ratio	-
I	Ampere meter reading	amp
V	Voltmeter reading	volts
N_1	Driving shaft speed	rpm
N_2	Driven shaft speed	rpm
R_{EH}	Mean effective Radius of holding brake drum	meter
R_{BH}	Radius of holding brake drum	meter
R_{EO}	Mean effective radius of output brake drum	meter
R_{BO}	Radius of output brake drum	meter
W_1	Spring balances reading of brake drum	kilogram
W_2	Spring balance reading of output brake drum	kilogram
T_x	Tension at tight side	kilogram
T_y	Tension at slack side	kilogram
T_B	Thickness of belt	millimeter

I. INTRODUCTION

Gear train is a mechanical arrangement which comprised of more gear wheels are meshing with one another so as to transmit power or torque between different shafts. The type of train employed is determined by the needed speed ratio and also the relative placement of the shaft axis. Depending on the arrangement of wheels; simple, reverted, compound, and epicyclical gearing system, etc. are some of the most common forms of gear trains. There may be multiple pinions that rotate around the sun gears of the planetary gear mechanism. To achieve high reduction ratio in a compact and power-dense system, epicyclical or planetary gearing systems are commonly used (Dilawer, Junaidi, and Mehdi, 2013) and (Rao, 1979). The planetary carrier revolves to carry the pinion around the other gear by connecting the central axes of the two gears. Planetary carriers transmit pinions around the sun's ring. The pinions are attached to the carrier by a bearing which meshes with the ring and sun gear at the same time. The number of planet gears changes according to the

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system's design load. Planetary gears benefit from many load routes and small packaging because they have multiple planets (Pennestri and Freudenstein, 1993). The several types of gear trains can be manually moved to achieve varying degrees of speed reduction. Conversely, epicyclical gearing systems are easily adaptable to automatic control. This degree of efficiency indicates that a significant portion of the input power (approximately 97%) is transmitted through the gearbox instead of being wasted caused by mechanical losses. Torque capability is significantly increased in an epicyclical gearing train because the load is distributed across multiple planets.

After reviewing the majority of the research publications, it is found that there has been a significant amount of research on the synthesis and optimization of epicyclical gearing system for a variety of applications. (KUMAR, ANAND, KUMAR, YADAV, and SHEKH, 2018) optimized gear design using the finite element approach and optimization technique, which reduces load failure in gears. These approaches have been modified to use the fewest number of gears possible, thereby reducing the complexity of the gearbox system. (Patel, Dubey, and Rao, 2019) created and analyzed an epicyclic gearbox for an electric drive train. The study focuses on developing and optimizing planetary gearboxes for FASE formula electric cars by considering static and wear loads for gearboxes in order to achieve the best lightweight design for their application.

(S. S. Sutar and Rawal, 2016) used both experimental and analytical methods to calculate the torque applied at holding drum in an epicyclical gearing. An error between experimental and theoretical methods ranges from 6% to 8%, which is as a result of frictional and mechanical losses that occurred during experimentation (Akhila and Reddy, 2014) and (Pennestrı and Valentini, 2003).

II. PROPOSED METHODOLOGY

In this research paper, an effort has been made to analyze the speed and torque of planetary gearing system by calculating the torque applied at input shaft and output drum and holding drum of epicyclical gearing through an experimental setup. The setup includes an epicyclical gearing system (internal type) a digital voltmeter and an ammeter to measure the input power and hence the input torque. A D.C motor of 230 V and 1500 rpm is coupled to the epicyclical gear train inside a housing whose shaft is coupled to the holding drum and further to the output drum. The output and holding torque is calculated using the rope brake dynamometer arrangement by considering the readings at the spring balances which are attached to the holding drum and output drum. The various experiments were taken for different input speeds and then the experimental values were compared with the theoretical values.

III. TECHNICAL ASPECTS OF EPICYCLIC GEAR TRAIN

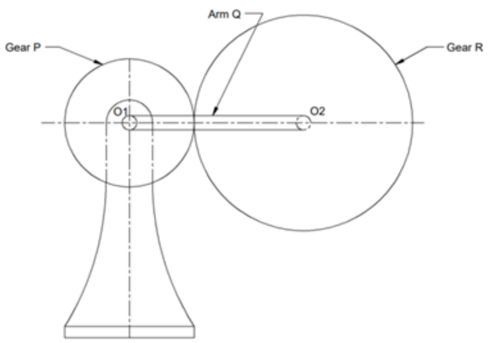

Fig 1. Epicyclical Gearing System

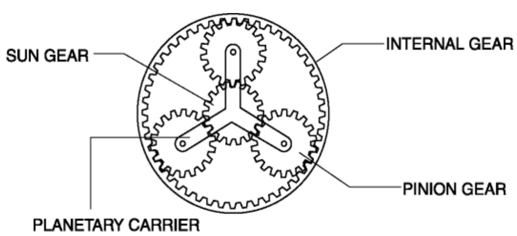

Fig 2. Internal Structure of Planetary Gearing System

A simple epicyclical gearing system shown in figure 1, which having shared center at O_1 that the gear P and arm Q may spin around. The gear R is in sync with the gear P and at the axis of the arm O_2 that it can revolve around. The gearing system or train is simple, if the arm is stationary and which rotates around the axes of gear P (i.e. around O_1). Furthermore, if arm is made stationary and turned about the gear P axis (i.e. around O_1), the wheel or gear R is compelled to revolve about the gear P. So this defines epicyclical or planetary gearing train with one or more elements that move on top of and around other element (Kumar, Athreya, Sharma, and Dinesh).

The internal structure of planetary gearing train used in the experimental setup shown in figure 2. This gear train having one sun gear that mesh with three planetary gears of the same size, which are connected together by planetary carrier.

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The sun gear serves as driver gear of the planetary gearing. The outer ring known as annular wheel having inward facing teeth which meshes with all these planetary gears (Mantriota and Pennestr`ı, 2003).

A. DIFFERENT TORQUES IN GEAR TRAIN

The three independently provided torques balance the gear train when the spinning components of an epicyclical gearing gear has no angular or rotary acceleration, as shown in Figure 3.

i. Input torque applied to the driving member (T_I)
ii. Output or resisting torque applied to driven part (T_O)
iii. Holding or fixing or braking torque applied to fixed part (T_H)

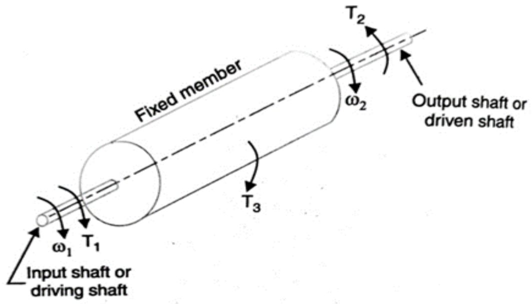

Fig 3. Torques in the gear train

The gear train must not be subjected to any net torque. In other words,

$$T_I + T_O + T_H = 0 \qquad \ldots\ldots(1)$$

$$\therefore\ F_1 . r_1 + F_2 . r_2 + F_3 . r_3 = 0 \qquad \ldots\ldots(2)$$

Here F_1, F_2, and F_3 are the equivalent forces applied externally at radii r_1, r_2, and r_3, respectively.

If angular velocity of driving member is ω_1, driven member is ω_2 and fixed member is ω_3; and the friction between meting parts are to be neglected, then total kinetic energy which dissipated by the gearing system must be zero, therefore

$$T_I . \omega_1 + T_O . \omega_2 + T_H . \omega_3 = 0 \qquad \ldots\ldots(3)$$

But, ω_3 will be zero for the fixed member.

$$T_I . \omega_1 + T_O . \omega_2 = 0 \qquad \ldots\ldots(4)$$

From the equation (1) and (4), braking torque or holding torque T_3 will be calculated as;

$$T_O = -T_I \times \frac{\omega_1}{\omega_2} \qquad \ldots\ldots(5)$$

And

$$T_H = -(T_I + T_O)$$

$$T_H = T_I \times \left(\frac{\omega_1}{\omega_2} - 1\right) = T_I \times \left(\frac{N_1}{N_2} - 1\right) \qquad \ldots\ldots(6)$$

If the input member and output drum rotate in the same rotational direction, the torque applied at input and output member will be opposite; and if the input member and output drum rotates in opposite rotational directions, the torque applied at input as well as output member will be in the same rotational direction.

B. SPEED ANALYSIS OF PLANETARY GEARING TRAIN

Table 1: Steps for speed analysis analytical calculations

Sr. No.	**Operations**	**Elemental revolution**		
		Arm Q	**Gear P**	**Gear R**

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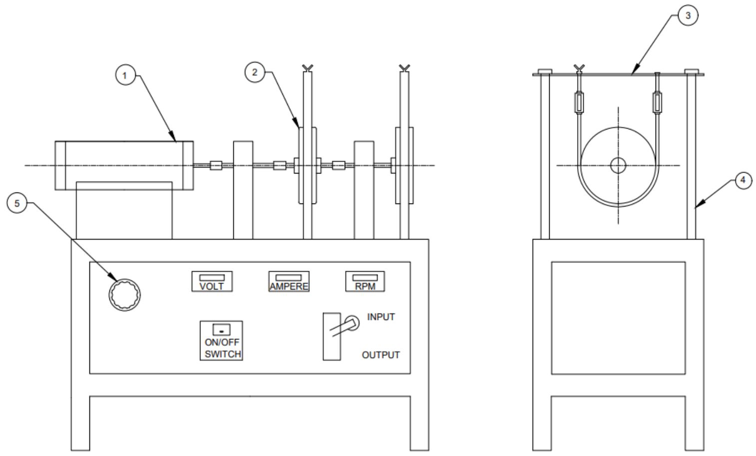
1	Fix the arm and rotate gear A by +1.	0	$+1$	$-\frac{Z_A}{Z_B}$
2	The arm motion is restricted and the wheel P rotates + x	0	$+x$	$-x \times \frac{Z_A}{Z_B}$
3	Addition + y rotation to each elements.	$+y$	$+y$	$+y$
4	Total number of rotations	$+y$	$x + y$	$y - x \times \frac{Z_A}{Z_B}$

Figure 1 shows an epicyclical or planetary gearing system. Z_A and Z_B are total number of teeth of gear P and R respectively.

Let's pretend for a while that arm is rigidly fixed. As a result, axes of gear P and R, are fixed in relation with one another. When gear P rotates in an anticlockwise direction, gear R rotates in a clockwise direction, making Z_A / Z_B revolutions. If it is to be consider anticlockwise rotation is positive and clockwise rotation is negative, so wheel P produces $+1$ rotations, wheel R provides $(-Z_A / Z_B)$ rotation. This relative motion statement is written in the first row of the table. Second, if gear P rotates at +x rotations, the R will be rotate at $(-x \times Z_A / Z_B)$ rotations. To put it another way, multiply each move (in the first line) by x. The next step is to assign +y revolutions to all members of an epicyclical gearing train and arranges them in third row of table 1. Lastly, the movements of all the gear components are added and entered on the final row. If the rotational motion requirements of any two elements are known, it is possible to estimate the unknown velocity of the third element by substituting the calculated values in the fourth row and third column.

IV. EXPERIMENTAL SETUP AND SPECIFICATIONS



1. D C Motor
2. Compact Gear Train
3. Rope Brake Dynamometer
4. Stand
5. Dimmerstat

Fig 4. Schematic Layout of Internal epicyclic type Gear Train

The planetary gearing (internal type) is used, with sun gear positioned on the input shaft. The arm is equipped with three planet gears that revolve freely on fixed pins and meshes with the central sun gear and annular wheel internal teethes. The changing RPM of input member is regulated by a DC motor and which is controlled by a dimmerstat. To monitor input power and thus input torque, a digital voltmeter and ammeter are provided. A rope brake dynamometer with spring balances is provided to measure the torque applied on holding drum and torque applied on output drum. To measure speed of the output and input shafts, a digital RPM indicator with a selector switch is provided.

SPECIFICATIONS

- Availability of electric supply: Single Phase, 230 V, 50 Hz, 4 Amp combined socket with earth connection.
- Required Floor Space: 2 m x 1 m
- Radius of Holding brake drum (R_{BH}): - 0.050 m
- Radius of Output brake drum (R_{BO}): - 0.082 m
- Thickness of Belt (T_B): - 2.5 mm = 0.0025 m

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- Efficiency of Motor (η)= 90%

V. EXPERIMENTAL PROCEDURE AND OBSERVATION

Initially it was ensured that the ON/OFF button on the control panel were turned off. Then, set the dimmerstat to its original position before turning on the main power supply. The input shaft speed were then adjusted using a dimmerstat. After adjusting the speed of input shaft load on holding brake drum was applied to stop the rotation of the input shaft by using rope brake dynamo meter and Load was recorded by spring balances. After that, the voltmeter and ampere meter readings were recorded; also the spring balance readings for the holding and output drums, as well as the rpm of the input and output shafts from the RPM indicator, using the selector switch were recorded. Then the weight or load were added on to the output brake drum to stop its rotation. After that altered the load on the holding brake drum and repeated the procedure for different loads and input speeds.

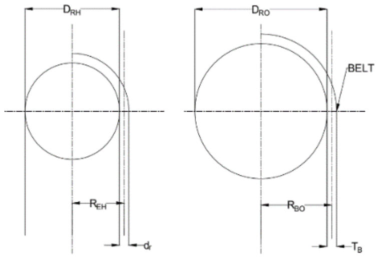

Fig 5. Sectional diagram of holding break drum and output break drum belt

The sectional diagram of holding break drum and output break drum belt for the understanding of mean effective radius while calculating torque applied at holding and output break drum shown in figure 5.

The experimental observation is shown in table 2.

Table 2. Experimental Observations

Expt. No	V (Volt)	I (Amp)	N_1 (rpm)	N_2 (rpm)	W_1 (kg)		W_2 (kg)	
					T_x	T_y	T_x	T_y
1	59.50	0.312	522.5	91.23	1.725	0.850	0.000	0.000
2	59.50	0.808	449.4	78.51	3.745	1.000	0.450	2.100
3	59.50	0.983	428.3	74.84	4.995	1.500	0.750	3.500
4	50.00	0.530	397.8	69.48	2.490	0.850	0.000	0.000
5	49.00	0.730	348.0	60.78	3.475	0.850	0.450	1.900
6	48.50	0.922	331.8	57.98	5.170	1.500	0.750	3.400
7	39.50	0.470	308.6	53.93	2.225	0.750	0.000	0.000
8	39.50	0.659	263.2	45.99	3.050	0.600	0.400	1.550
9	39.50	0.810	229.1	40.02	4.950	1.450	0.650	2.750

Following are the mathematical equations to calculate torque experimentally.

1. Gear Ratio

$$G = \frac{N_1}{N_2} \qquad \dots\dots\dots (7)$$

2. Input Torque

$$T_I = \frac{V \times I \times \eta \times 60}{2\pi N_1} \qquad \dots\dots\dots (8)$$

3. Holding Torque

$$T_H = W_1 \times g \times R_{EH} \qquad \dots\dots\dots (9)$$

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Where, $W_1 = T_x - T_y$ and $R_{EH} = \left(R_{BH} + \frac{T_B}{2}\right)$

4. Output Torque

$$T_O = W_2 \times g \times R_{EO} \qquad \ldots\ldots\ldots(10)$$

Where, $W_2 = T_x - T_y$ and $R_{EO} = \left(R_{BO} + \frac{T_B}{2}\right)$

VI. RESULTS AND DISCUSSION

The result table has been shown in Table 3

Table 3. Experimental and Theoretical Results

Expt. No.	Gear Ratio	Experimental			Theoretical			% Error in T_H (Nm)
		T_I (Nm)	T_H (Nm)	T_O (Nm)	T_I (Nm)	T_H (Nm)	T_O (Nm)	
1	5.727	0.440	0.000	0.109	0.514	0.622	0.306	14.35
2	5.724	1.380	1.348	0.327	1.545	1.873	0.920	10.70
3	5.723	1.757	2.246	0.418	1.972	2.390	1.174	10.91
4	5.725	0.825	0.000	0.204	0.963	1.166	0.573	14.35
5	5.726	1.320	1.184	0.314	1.485	1.800	0.884	11.15
6	5.723	1.845	2.164	0.412	1.946	2.359	1.159	5.20
7	5.722	0.742	0.000	0.184	0.869	1.053	0.517	14.64
8	5.723	1.232	0.939	0.302	1.428	1.731	0.850	13.77
9	5.725	1.760	1.715	0.427	2.018	2.445	1.201	12.79

The analytic and experimental comparison for torque values reveals an inaccuracy of less than 15%. There are additional mechanical and frictional losses. The efficiency of the motor, loss of friction between the rope drum and belt, and the spring stiffness employed for measuring can all have an effect on torque measurements. Figure 6 shows the experimental values of input and output speed with its respective gear ratio. It is found that, the gear ration is nearly same for all experiments.

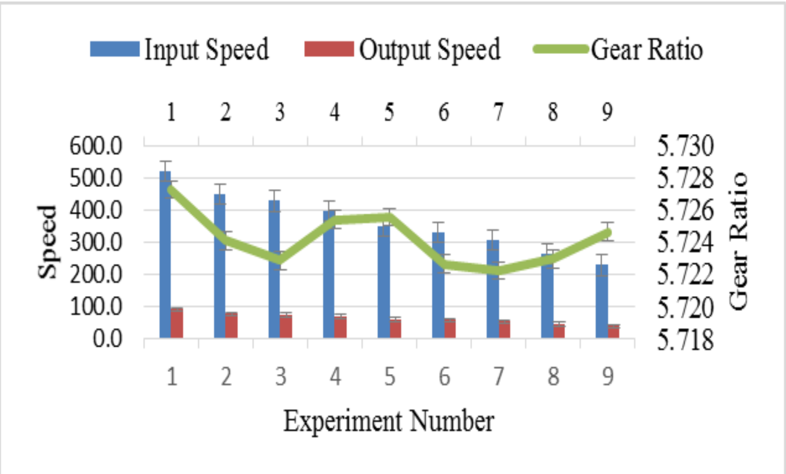

Fig 6. Speed analysis of epicyclic gear train

The experimental and theoretical torque analysis results are shown in figure 7 and figure 8 respectively. It is found that there is no major difference in the results.

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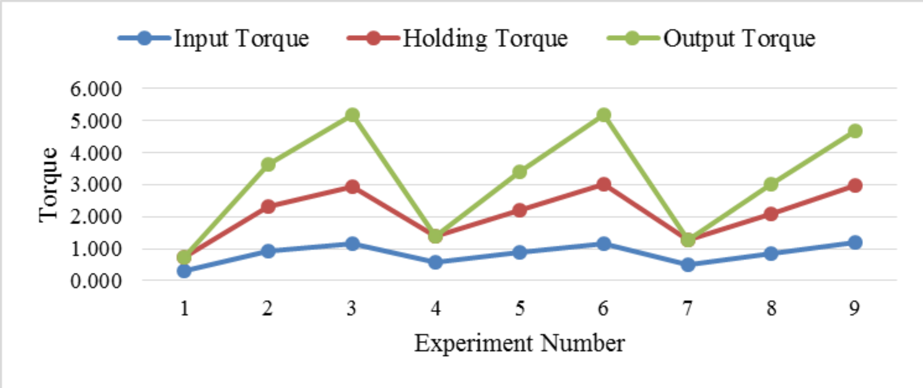

Fig 7. Experiential torque analysis

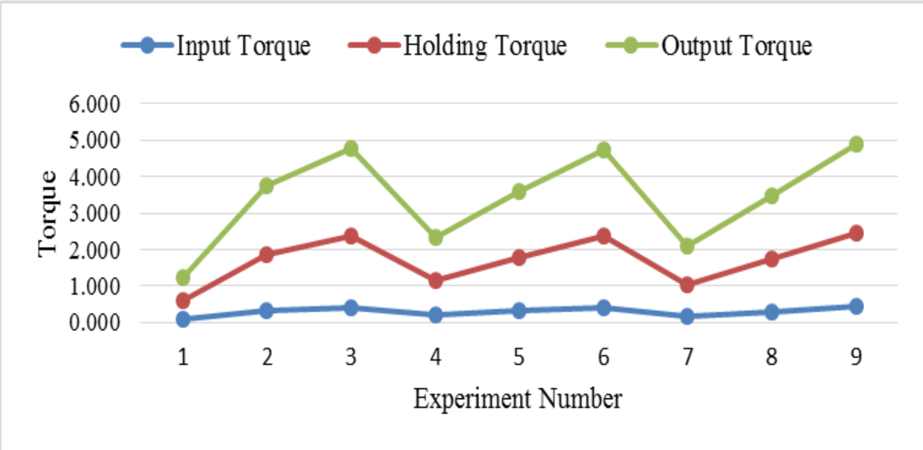

Fig 8. Analytical torque analysis

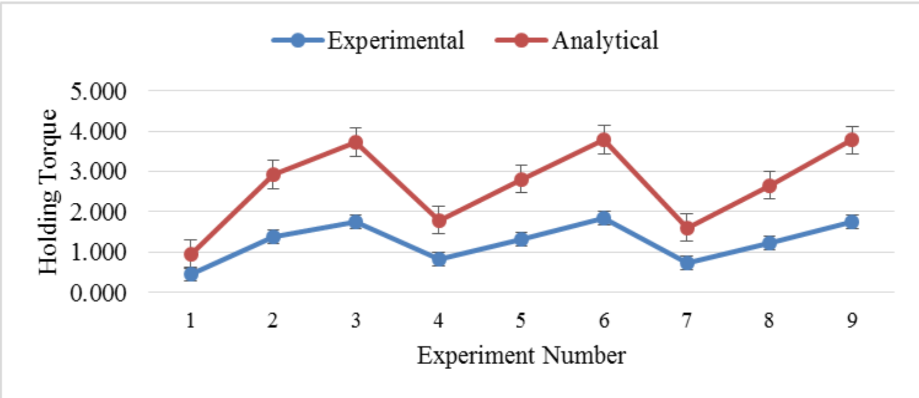

Fig 9. Analytic and experimental comparison of holding torque

The comparison between the experimental holding torque and analytical holding torque is shown graphically in figure 9. The percentage error in both the results is less than 15. The error is due to the mechanical and frictional losses in the experimental setup and also the efficiency of the epicyclic gear train was not considered 100%.

VII. CONCLUSION

In this paper, an attempt has been made to calculate the holding, input and output torque of epicyclical gear train by using the experimental and analytical method. The speed analysis has been made for the average gear ratio of 5.72. It is found that there is no major variation in the experimental and analytical torque values. There are various parameters such as frictional losses between the belt and holding brake drum and output brake drum, mechanical loss, efficiency of the motor which the error in the torque values. The percentage error between the experimental and analytical torque is less than 15%.

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