

Static Torque Sharing Distribution of Planetary Gear Sets Generated by Manufacturing Error Distribution

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ABSTRACT

Creating an appropriate design for gear strength in a planetary gear set requires that manufacturing errors be grasped statistically and that the distribution of torque to the planet gears due to manufacturing errors be projected. This article describes the derivation of an expression of relation between manufacturing errors and the torque applied to planet gears in a common n-planet planetary gear set. As a result, a formula expressing the distribution of torque to the planet gears could be obtained. Tests were also conducted to verify the relational expression, and its validity was confirmed. In order to project the distribution of torque, it was necessary to estimate the stiffness of the planetary gear set in the relational expression. Those stiffness values were calculated by numerical analysis using a 3D finite element model (FEM) to which the plan values and material property values were entered. The accuracy of the estimated stiffness values was verified by testing, and it was confirmed that the stiffness values could be estimated with a high degree of accuracy. Given the above, the distribution of torque to planet gears due to manufacturing error in a common n-planet planetary gear set could be estimated with a high degree of accuracy using only estimated data obtained by desktop methods.

1. Introduction

Because the loads input to planetary gear sets are distributed between multiple planet gears before being output, the use of these gear sets can reduce the size and weight of the automatic transmission. Ideally, input torque should be distributed equally among the planet gears and transmitted. However, in actual practice planetary gears may be unable to effectively demonstrate their advantages because manufacturing errors would result in an uneven distribution of torque to each gear.

Because manufacturing errors differ depending on the product, the ratio of torque distributed to each planet gear in a planetary gear set also differs depending on the product. Given this, when the entire set of manufactured products is considered statistically, the collection of the torque of a planet gear, which shares a given arbitrary torque input to a planetary gear set with other planet gears, shows a distribution in response to the distribution of manufacturing errors. It is therefore necessary to take manufacturing error into consideration when designing the strength of the gears used in planetary gear sets.

Conventionally, in strength design for gears used in planetary gear sets, the effect of production errors has

been considered based on experiential evidence of the safety factor, and the gears have tended to be manufactured with excessive quality. If it was possible to predict the distribution of torque to the planet gears from the distribution of production errors, the safety factor could be optimized, and it would be possible to design gears that possessed the necessary and sufficient strength.

Dimensional error in machining is known to generally display a normal distribution, and the dimensions of the gears are managed using blueprint values based on this fact. Because of this, the distribution of torque to the planet gears in relation to the distribution of production errors could be known if the relationship between any arbitrary production error that affects the distribution of torque between the planet gears and the torque on each of the gears was known theoretically. In addition, if the theoretical relationship between production errors and the torque on each planet gear was known, it would be possible to predict the distribution of torque to the planet gears without conducting strength analyses for a number of arbitrarily-selected combinations of production errors.

An equation expressing the relationship between an arbitrary production error and the torque on the planet gears has been derived for 4-planet planetary gear sets⁽¹⁾,

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but not for normal n -planet planetary gear sets. In addition, the validity of the relational expression has not been verified.

Because the torque capacity of planetary gear sets can be increased by increasing the number of planet gears, planetary gear sets using five or more gears are now in use in vehicles in order to achieve increased torque capacity. Given this, an equation expressing the relationship between any arbitrary production error and the torque on the planet gears is necessary for planetary gear sets containing five planet gears or more.

This paper therefore discusses the derivation of a relational expression between any arbitrary production error and the torque on the planet gears for a normal n -planet planetary gear set. The validity of this equation was verified in tests.

2. Relationship between Eccentric Error and Load

Hayashi et al.⁽¹⁾ and Hidaka et al.^{(2),(3)} formulated linear models of planetary gear sets with multiple degrees of freedom in the transverse plane, and conducted analyses of the relationship between various types of error and the torque distribution between the planet gears by defining the non-uniformity of the torque distribution between the gears resulting from the various types of error as differences in the timing of the commencement of meshing between the gears produced by the errors, and converting the different production errors to transmission errors between the gears. Following this methodology, in the project discussed here the author formulated a linear model of a planetary gear set in the transverse plane, assuming an proper design in which the distribution of the tooth-contact pattern for each gear in the set was close to the centers of the teeth. Converting production errors into equivalent transmission errors and focusing on the self-centering action in the planetary gear set resulted in the derivation of an expression of relation between production errors and the torque on each planetary gear in a planetary gear set containing n normal planet gears. When production errors were converted to equivalent transmission errors, the production errors were sufficiently small in relation to the distance between the axes of the gears so that the change in the line of action due to the production errors could be ignored. The planetary gear set studied in the project was assumed to display sufficient clearance in the radial direction, and its self-centering action was assumed to function perfectly. Because the production errors were converted to equivalent transmission errors using the same scale of measurement, they are treated below as a single category of production error. Eccentric errors in the positioning of the holes for the pins that support the planet gears are known to be the errors that have the greatest effect on the distribution of torque between the gears⁽⁴⁾, and production errors are

therefore treated here as eccentric errors in the positioning of the pin holes.

2.1. Eccentric Errors and Equivalent Transmission Errors

For a planetary gear set with n planet gears, the equivalent transmission error for the meshing between the j th planet gear and the sun gear, ε_{SPj} , and the equivalent transmission error for the meshing between the j th planet gear and the internal gear, ε_{IPj} , both produced by an eccentric error in the positioning of the hole for the support pin for the j th planet gear, can be expressed as follows:

$$\varepsilon_{SPj} = -\vec{d}_{pj} \cdot \vec{i}_{SPj} \quad (1)$$

$$\varepsilon_{IPj} = -\vec{d}_{pj} \cdot \vec{i}_{IPj} \quad (2)$$

where

$\vec{d}_{pj}$: Eccentric error in positioning of hole for support pin for j th planet gear

$\vec{i}_{SPj}$: Unit vector in line of action of sun gear and j th planet gear (from sun gear towards planet gear = positive)

$\vec{i}_{IPj}$: Unit vector in direction of line of action of internal gear and j th planet gear (from internal gear towards planet gear = positive)

The symbol $\cdot$ in the equations represents the inner product of the vector.

The equivalent transmission errors ε_{SPj} and ε_{IPj} take on positive values when the eccentric error of the pin hole position is located in the direction of separation of the gear teeth. The sum of the equivalent transmission errors for the meshing of the j th planet gear, the sun gear, and the internal gear is expressed as ε_{pj} .

$$\varepsilon_{pj} = -a_p \vec{d}_{pj} \cdot \vec{i}_{pj} \quad (3)$$

Here,

$$\vec{a}_{pj} = \vec{i}_{SPj} + \vec{i}_{IPj} \quad (4)$$

$$\vec{a}_p = |\vec{a}_{SP1}| = |\vec{a}_{SP2}| = \cdots = |\vec{a}_{SPn}| \quad (5)$$

$$\vec{i}_{pj} = \vec{a}_{pj} / a_p \quad (6)$$

To take an example, when the position of one of the pin holes deviates from the correct position as shown in Fig. 1, the planet gear supported by that pin (Planet 1) will commence meshing earlier than the other planet gears. The degree of this difference in meshing times, expressed as a distance on the line of action, is the equivalent transmission error.

2.2. Translational Motion and Equivalent Transmission Error

When torque increases in a planetary gear set in which there is an eccentric error in a pin hole position, the

positions of the sun gear, the planet carriers, and the internal gear will settle into a position in which the gear mesh forces are in equilibrium, as long as sufficient clearance exists in the radial direction of the supports.

There are countless combinations of positions of the sun gear, planet carriers, and internal gear in which the gear mesh forces are in equilibrium, but for the sake of convenience, in this paper the sun gear and the internal gear will be considered to be concentric, and only the position of the centers of the planet carriers will differ against these. Because the torque on each planet gear is identical even in differing combinations of positions, generality is not lost despite this approach.

For example, for the pin hole displacement in Fig. 1, the position of equilibrium for the planet carrier is in the direction of the resultant force of the gear mesh forces on the planet gear Planet 1. As Fig. 2 shows, from the state shown in Fig. 1, the planet carriers move in the direction of the resultant force of the gear mesh forces on Planet 1, to the position where Planet 1 and the opposite gear Planet 3 simultaneously transmit torque. Because Planet 1 and Planet 3 transmit an identical volume of torque in this position, the forces on the planet carriers are in equilibrium. With regard to Planet 2 and Planet 4, the distance between the gear teeth of the sun gear and the planet gears will either increase or decrease, and the distance between the gear teeth of the planet gears and the internal gear will consequently either decrease or increase; these effects cancel each other, and there is therefore no change in the fact that these planet gears are delayed against Planet 1 in commencing to mesh, despite the movement of the planet carriers described above.

The equivalent transmission error for the meshing of the j th planet gear and the sun gear, c_{SPj} , and the equivalent transmission error for the meshing of the j th planet gear and the internal gear, c_{IPj} , generated by the eccentricity of the planet carriers, can be expressed as follows:

$$c_{SPj} = -\vec{c} \cdot \vec{i}_{SPj} \quad (7)$$

$$c_{IPj} = -\vec{c} \cdot \vec{i}_{IPj} \quad (8)$$

where

$\vec{c}$: Position of center of planet carrier in relation to centers of sun gear and internal gear in state of equilibrium of forces

The sum of these errors is determined and termed c_{pj} .

$$c_{pj} = -a_p \vec{c} \cdot \vec{i}_{pj} \quad (9)$$

The sufficient clearance in the radial direction assumed in the planetary gear set is given when RCL_c , the measured clearance of the planetary carrier in the radial direction on the condition that the sun gear and the internal gear are concentric, satisfies the following

equation:

$$RCL_c \geq |\vec{c}| \quad (10)$$

2.3. Equations of Gear Mesh Force

A linear spring model of the type shown in Fig. 3 will be considered as a stiffness model of a planetary gear set. Here, the meshing stiffness of the sun gear and the planet gears, the stiffness of the bearings supporting the planet gears, the stiffness of the pins, and the stiffness of the planet carriers are considered linear springs with fixed spring constants; k_{sp} is the synthetic spring constant

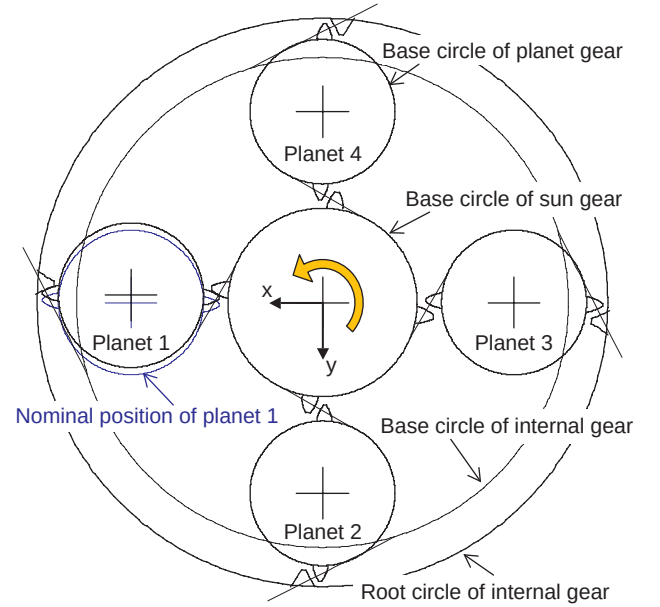


Fig. 1 Pin hole displacement

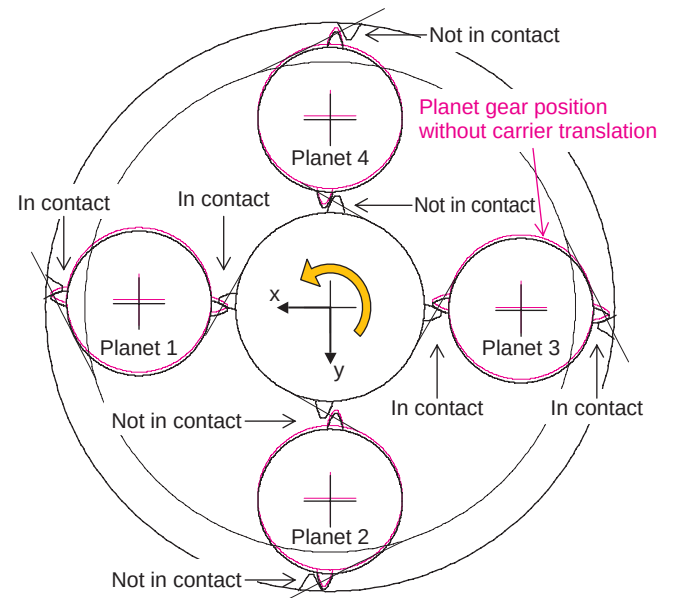


Fig. 2 Carrier translation

formed by synthesizing these terms. k_{IP} is the synthetic spring constant formed by synthesizing the meshing stiffness of the internal gear and the planet gears, the stiffness of the bearings supporting the planet gears, the stiffness of the pins, the stiffness of the planet carriers, and the annular stiffness of the internal gear, considered as linear springs with fixed spring constants.

The degree of progression of the contact points between the j th planet gear and the sun gear on the line of action in the period between the commencement of contact between the j th planet gear and both the sun gear and the internal gear and the completion of deformation of each part of the j th planet gear under the resulting loads is termed x_{SPj} . The degree of progression of points of contact between the j th planet gear and the internal gear on the line of action is termed x_{IPj} . In addition, because of the equilibrium in the direction of rotation of the planet gears, the magnitude of the gear mesh force between the planet gear and the sun gear and the magnitude of the gear mesh force between the planet gear and the internal gear are equivalent, and the magnitude of these forces is termed f_{pj} . The magnitude of the gear mesh force on the j th planet gear, f_{pj} , in relation to meshing with the sun gear and meshing with the internal gear, can be expressed as follows:

$$f_{pj} = k_{SP}x_{SPj} \quad (11)$$

$$f_{pj} = k_{IP}x_{IPj} \quad (12)$$

In addition, the sum of x_{SPj} and x_{IPj} is found and substituted as x_{pj} .

$$x_{pj} = x_{SPj} + x_{IPj} \quad (13)$$

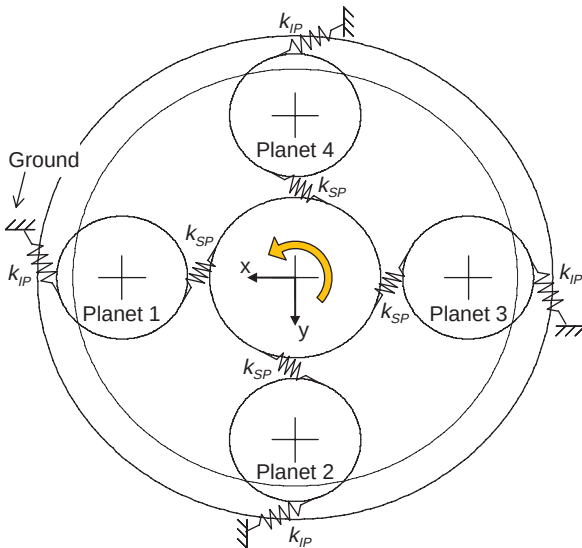


Fig. 3 Stiffness model of planetary gear sets

Given Eqs. (11), (12), and (13), the magnitude of the gear mesh force on the j th planet gear, termed f_{pj} , can be expressed as follows:

$$f_{pj} = K_P x_{pj} \quad (14)$$

Here,

$$K_P = k_{SP}k_{IP}/(k_{SP} + k_{IP}) \quad (15)$$

Because the difference in the degree of progression of the contact points between the planet gears is the equivalent transmission error, values from x_{pj} , excepting ε_{pj} and c_{pj} , are equivalent for each planet gear. Terming this value $\hat{x}_p$,

$$x_{pj} = \hat{x}_p + \varepsilon_{pj} + c_{pj} \quad (16)$$

From Eq. (16), f_{pj} can be expressed as follows:

$$f_{pj} = K_P(\hat{x}_p + \varepsilon_{pj} + c_{pj}) \quad (17)$$

The following equations can be obtained from Eq. (17):

$$\vec{f}_{SPj} = K_P(\hat{x}_p + \varepsilon_{pj} + c_{pj})\vec{l}_{SPj} \quad (18)$$

$$\vec{f}_{IPj} = K_P(\hat{x}_p + \varepsilon_{pj} + c_{pj})\vec{l}_{IPj} \quad (19)$$

Here,

$\vec{f}_{SPj}$: Gear mesh force between j th planet gear and sun gear

$\vec{f}_{IPj}$: Gear mesh force between j th planet gear and internal gear

2.4. Equation of Equilibrium of Forces

The following equation can be obtained from the state of equilibrium between the forces on the planet carriers and Eqs. (18) and (19):

$$\sum_{j=1}^n K_P(\hat{x}_p + \varepsilon_{pj} + c_{pj})a_p\vec{l}_{pj} = \vec{0} \quad (20)$$

Because the sum of the torque transmitted between the planet gears is the torque on the sun gear, T_s , if the radius of the base circle of the sun gear is termed r_{BS} , then

$$\sum_{j=1}^n K_P(\hat{x}_p + \varepsilon_{pj} + c_{pj})r_{BS} = T_s \quad (21)$$

2.5. Relationship between Equivalent Transmission Error and Torque Distribution

A relational expression between equivalent transmission error and torque distribution was derived from the equilibrium between the forces.

Even if, from symmetry,

$$\vec{i}_{p1} = \begin{pmatrix} 1 \\ 0 \end{pmatrix} \quad (22)$$

generality is not lost. At this time,

$$\vec{i}_{pj} = \begin{pmatrix} \cos(2\pi(j-1)/n) \\ \sin(2\pi(j-1)/n) \end{pmatrix} \quad (23)$$

The following equation is obtained from Eqs. (9), (20), and (23):

$$\vec{c} = \begin{pmatrix} 2u/(na_p) \\ 2v/(na_p) \end{pmatrix} \quad (24)$$

Here,

$$u = \sum_{h=1}^n \varepsilon_{ph} \cos(2\pi(h-1)/n) \quad (25)$$

$$v = \sum_{h=1}^n \varepsilon_{ph} \sin(2\pi(h-1)/n) \quad (26)$$

Further, the following equation, expressing the relationship between equivalent transmission error and the gear mesh force between each planet gear, can be obtained from Eqs. (9), (17), (21), and (24):

$$f_{pj} = T_s/(nr_{BS}) + K_p E_{pj} \quad (27)$$

Here,

$$E_{pj} = \varepsilon_{pj} - \left(U_{pj} + V_{pj} + \sum_{h=1}^n \varepsilon_{ph} \right) / n \quad (28)$$

$$U_{pj} = 2u \cos(2\pi(j-1)/n) \quad (29)$$

$$V_{pj} = 2v \sin(2\pi(j-1)/n) \quad (30)$$

E_{pj} , obtained from Eq. (28), will be termed the projected error for the j th planet gear.

Having termed the radius of the base circle of the planet gear r_{BP} , from Eq. (27), the torque transmitted by the j th planet gear, T_{pj} , is

$$T_{pj} = (T_s r_{BP}) / (nr_{BS}) + K_p r_{BP} E_{pj} \quad (31)$$

The first term on the right-hand side of this equation is the value of torque when the load is equally distributed, and the second term on the right-hand side is the deviation from the torque value when the load is equally distributed. The deviation is proportional to the projected error of the j th planet gear, E_{pj} .

If the positional tolerance of the pin hole is termed ϕb , the following equation for the distribution of torque between the planet gears can be obtained from Eq. (31):

$$f(T_{pj}) = (1/\sqrt{2\pi}\sigma) \exp(-(T_{pj} - m)^2 / (2\sigma^2)) \quad (32)$$

Here,

$$m = (T_s r_{BP}) / (nr_{BS}) \quad (33)$$

$$\sigma = a_p K_p r_{BP} \sigma_{PH} \sqrt{(n-3)/n} \quad (34)$$

$$\sigma_{PH} = b/6 \quad (35)$$

3. Verification in Actual Planetary Gear Set

This chapter will discuss the results of tests conducted to verify the validity of the relational expression between the eccentric error of the pin hole position and the torque on the planet gears. Following this, it will discuss the results of tests conducted to verify the accuracy of estimation of the stiffness value K_p , which is necessary for prediction of the distribution of torque between the planet gears resulting from production errors.

The tests used a 4-planet planetary gear set, and the verification process focused on the relational expression in the case of four planet gears. The process of verification using the actual gear set was divided into two stages. First, a planet carrier with vanishingly small pin hole displacement was used to verify the method of measurement of the torque on the planet gears. Next, planet carriers in which pin hole positions had been intentionally displaced were used to verify the relational expression. Following this, the test results were used to verify the accuracy of estimation of the stiffness value, K_p .

In the case of a 4-planet planetary gear set, the relational expression between pin hole displacement and the torque on the j th planet gear, T_{pj} , is as follows:

$$T_{pj} = (T_s r_{BP}) / (4r_{BS}) + K_p r_{BP} E_{pj} \quad (36)$$

Here,

$$E_{pj} = (\varepsilon_{p1} - \varepsilon_{p2} + \varepsilon_{p3} - \varepsilon_{p4}) / 4 \quad (j = 1, 3) \quad (37)$$

$$E_{pj} = -(\varepsilon_{p1} - \varepsilon_{p2} + \varepsilon_{p3} - \varepsilon_{p4}) / 4 \quad (j = 2, 4) \quad (38)$$

From Eqs. (37) and (38), it was predicted that the torque on opposite pairs of planet gears in the test would be equivalent.

3.1. Test Rig

Figure 4 shows the layout of the test rig. The test rig contained two planet gears. Two internal gears were fixed to the case, connecting the planet carriers, and two sun gears were connected by a shaft. The planet gear in the left of the diagram was the subject of the measurements. Table 1 shows the specifications of this gear. Direct

Table 1 Planetary gear set specifications

Term		Sign	Value
Number of teeth	Sun gear	z_s	30
	Planet gear	z_p	23
	Internal gear	z_i	78
Normal module		M	1.3
Normal pressure angle		α	20 deg
Helix angle		β	20 deg
Center distance		a	37.1 mm

measurement of the torque on each planet gear represented a challenge. Strain gauges were therefore attached to the outer circumference of the internal gear, and the strain on the circular section of the internal gear when the planet gear passed under the section to which the strain gauges were attached was measured. Torque was estimated on this basis. Figure 5 shows the internal gear with a strain gauge attached. Two strain gauges were attached in the axial direction at intervals of approximately 90° in the circumferential direction, for a total of eight strain gauges.

3.2. Test Conditions

The tests to verify the method of measurement of the torque on the planet gears were conducted using a planet carrier with vanishingly small pin hole displacement. The tests to verify the validity of the relational expression were conducted using planet carriers in which pin hole positions

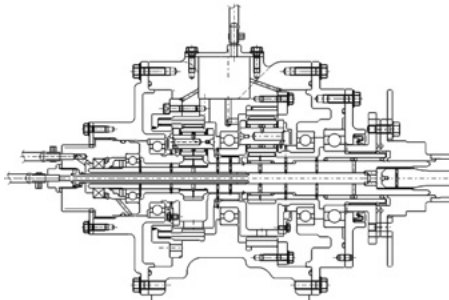


Fig. 4 Layout of test rig



Fig. 5 Position of strain gauge

had been intentionally displaced. Table 2 shows the deviation from the correct pin hole position in the four planet carriers, A to D. Table 3 shows the projected error of each planet gear in the planet carriers. Tests were conducted on each of the planet carriers at input torques of 118, 157, 196, and 235 Nm.

3.3. Results of Verification Test of Torque Measurement Method

Figure 6 shows the results of strain measurements at the different levels of input torque in the planet carrier with vanishingly small pin hole displacement. Because the speed of rotation was fixed, the horizontal axis that represents time is virtually corresponding to the angles of rotation of the planet carriers. The vertical axis shows strain on the internal gear. Strain was generated cyclically in response to the rotation of the planet carriers. Tensile strain was generated when the planet gears passed directly underneath the strain gauge, and compressional strain was generated at the median position between the two adjacent planet gears. The peak of tensile strain in particular can be seen as displaying a correlation with the torque on the planet gears. The four peaks shown in Fig. 6 correspond to each of the four planet gears in the planetary gear set. The values of the four peaks are virtually identical, and this indicates that the torque is evenly divided between the planet gears.

The peak values for tensile strain were isolated from measurement results for each input torque, and the arithmetic mean for each input torque was determined. The arithmetic means were plotted against the torque on the planet gears. Considering the load as having been distributed equally to each planet gear, the values of torque on the planet gears were estimated from the torque sharing ratio, the input torque on the sun gear, and gear ratio. The results are shown in Fig. 7. The horizontal axis shows the value of torque on the planet gear, while the vertical axis shows the arithmetic mean of the peak values of strain on the internal gear.

The strain on the outer circumference of the circular section of the internal gear is proportional to the torque

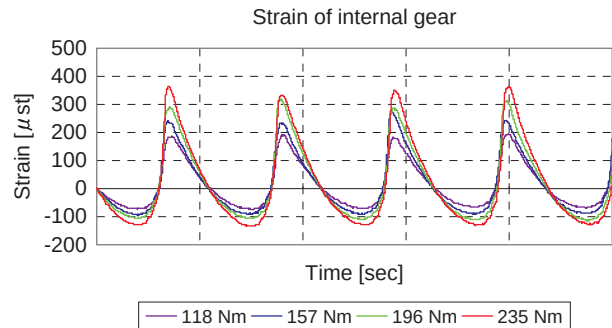


Fig. 6 Strain of internal gear for carrier with vanishingly small pin hole displacement

Table 2 Pin hole displacement of test carriers

Carrier	Pin hole displacement							
	μm							
	Planet 1		Planet 2		Planet 3		Planet 4	
	dx	dy	dx	dy	dx	dy	dx	dy
A	-1.5	40.2	0.5	-9.2	1.7	-3.3	-2.5	-5.4
B	-25.9	-2.0	-30.5	-8.6	-24.6	-16.5	-17.9	91.1
C	-11.6	2.5	-23.6	89.9	89.5	-20.9	1.0	-108.8
D	-20.4	3.0	-19.8	3.1	-19.1	54.4	-20.1	4.2

Table 3 Projected error of test carriers

Carrier	Projected error			
	μm			
	Planet 1	Planet 2	Planet 3	Planet 4
	E_{P1}	E_{P2}	E_{P3}	E_{P4}
A	-21.8	21.8	-21.8	21.8
B	-3.3	3.3	-3.3	3.3
C	7.7	-7.7	7.7	-7.7
D	24.0	-24.0	24.0	-24.0

on the planet gear. Therefore, the ratio of strain for each planet gear can be considered to be the ratio of torque carried by each planet gear. In the verification test for the relational expression, the ratio of torque carried by each planet gear to the total torque applied was determined from the ratio of strain, and the torque carried by each planet gear was estimated from the determined ratio of torque, the input torque to the sun gear, and gear ratio.

3.4. Results of Measurements of Torque on Planet Gears at Different Pin Hole Displacements

Figure 8 shows the results of measurements of strain at differing input torques for test piece A. The peak value of tensile strain was focused on, and it was seen to differ

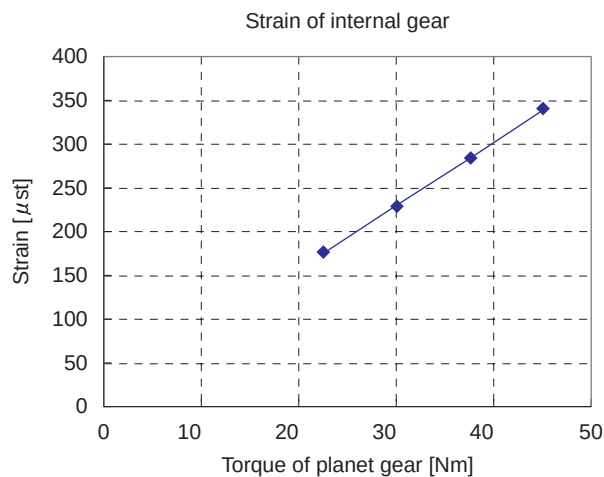


Fig. 7 Peak value of strain of internal gear with planet gear torque for carrier with vanishingly small pin hole displacement

depending on the planet gear. This result can be interpreted as reflecting the unevenness in the distribution of torque between the planet gears. The level of strain on pairs of opposing planet gears was virtually identical. This can be interpreted as resulting from the fact that the torque on opposing planet gears is identical, and accords with the result predicted from Eqs. (37) and (38).

Tests were conducted for test pieces A to D, and the peak values of tensile strain corresponding to each planet gear were isolated from the measurement results. The arithmetic means of the peak values for each input torque were determined, and the torque carried by each planet gear was estimated from these mean values. The torque estimated for each planet gear from strain was plotted against the projected error calculated from the degree of displacement of the pin hole positions in each test piece. Figure 9 shows the results for one of the four planet gears. The horizontal axis shows the projected error, and the vertical axis shows the torque on the planet gear. The

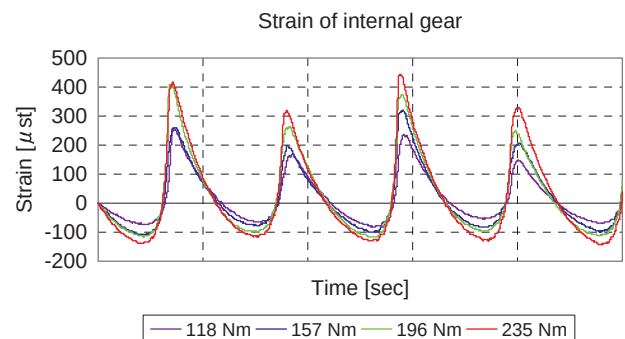


Fig. 8 Strain of internal gear for test carrier A

lines on the graph show the results of a least mean square approximation of test data using linear functions. The different colors show the different input torques.

As Fig. 9 shows, the value of the torque on the planet gear is proportional to the projected error. In addition, the y-intercept of the fitted lines showing the approximations are virtually identical to the torque value when the input torque is distributed equally between the planet gears. These results demonstrate that the proportional relationship expressed in Eq. (36) holds true in an actual planetary gear set.

3.5. Verification of Accuracy of Prediction of Stiffness Value K_p

Test results demonstrated that the proportional relationship expressed in Eq. (36) held true, but the accurate prediction of the distribution of torque between the planet gears would necessitate the accurate estimation of the stiffness value K_p . The value of K_p can be determined from a numerical analysis conducted using 3D FEM. The accuracy of estimation of the stiffness value K_p was verified for the planetary gear set used in the tests.

A 3D FEM model of the planetary gear set was formulated, and a numerical analysis was conducted. The only data employed was blueprint data such as shape and layout data, and physical values for the materials, such as Young's modulus. Using the stiffness value K_p obtained from calculation results, the torque on the planet gears was calculated for test pieces A to D. Figure 10 shows the results of these calculations with test results. The horizontal axis shows torque on the sun gear, i.e., input torque, and the vertical axis shows torque on the planet gears. The colored circles in the figure show test results, while the blank squares show calculation results. The differences in color represent the different planet carriers used in the tests.

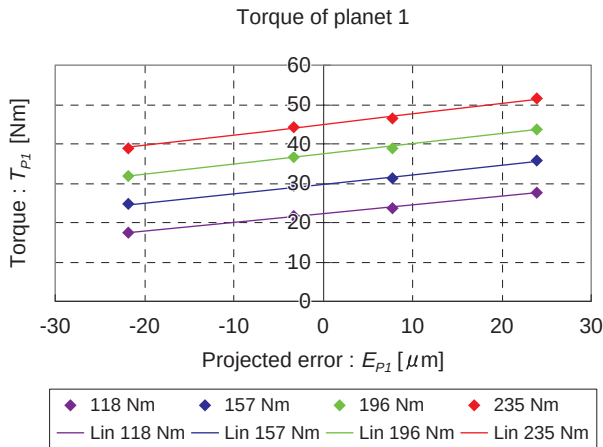


Fig. 9 Variation of torque of planet gear No.1 T_{P1} with projected error E_{P1} for different input torque

A comparison of test and calculation results reveals a good match between the two. This demonstrates that the stiffness value K_p can be accurately predicted by means of a 3D FEM analysis using only computer data.

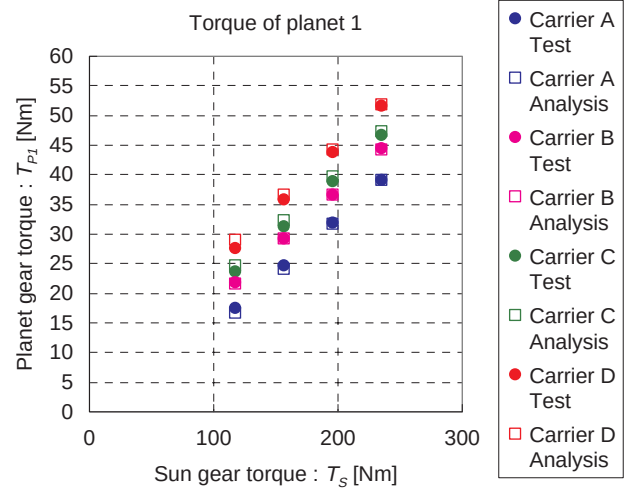


Fig. 10 Variation of torque of planet gear No.1 T_{P1} with sun gear torque T_s for different carrier

5. Conclusion

The project discussed in this paper focused on equivalent transmission error for planetary gear sets, with the following results:

- (1) A relational expression between eccentric error of the pin hole positions and the torque on the planet gears in a planetary gear set containing n normal planet gears was derived.
- (2) This relational expression made it possible to derive an equation expressing the distribution of torque to the planet gears in a planetary gear set due to production errors in relation to the pin hole positions.
- (3) The relational expression held true in an actual planetary gear set.
- (4) It was possible to accurately predict the stiffness value, K_p , necessary to the prediction of the distribution of torque to planet gears due to production errors, using a 3D FEM numerical analysis.

Based on these results, it is now possible to accurately predict the distribution of torque to the planet gears in a planetary gear set containing normal n planet gears using only computer data.

References

- (1) Hayashi, T., Takahashi, T., Ishikawa, J.: Load Balancing Theory on a Planetary Gearing, Transactions of the Japan

- Society of Mechanical Engineers, Vol. 36, No. 288, p. 1394-1404 (1970)
- (2) Hidaka, T., Terauchi, Y., Dohi, K.: On the Relation between the Run-Out Errors and the Motion of the Center of Sun Gear in a Stoekicht Planetary Gear, Transactions of the Japan Society of Mechanical Engineers, Vol. 44, No. 385, p. 3221-3230 (1978)
- (3) Hidaka, T., Sugimoto, N., Ishida, T.: Effects of Errors of Elements on Load Distribution in Planetary Gears with Various Load Equalizing Mechanisms, Transactions of the Japan Society of Mechanical Engineers. C, Vol. 52, No. 480, p. 2200-2206 (1986)
- (4) Shirokoshi, N., Hidaka, T., Kasei, S.: Studies on Influences of Geometrical Errors to Final Performances in Small Backlash Planetary Gear: Modeling and Simulations on Load Distribution, Transmission Error, and Sun Gear Radial Displacement, Transactions of the Japan Society of Mechanical Engineers. C, Vol. 65, No. 633, p. 2014-2021 (1999)

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