3D Model of the Differential Planetary Gear and the Kinematic and Force Analysis

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Abstract: *The goal of the presented contribution was to design of the differential mechanical gear box with power shit. According to the designed force scheme, basic forces and kinematics relationships between input and output parameters were specified. Force and kinematic analyze is presented in tabular and graphic form. The other part of the contribution was to design planetary gear box which is placed in the primary branch of differential gear box and variable gear which is placed in secondary branch of differential gear box. Variable ratio allows shifting 16 speed gears.*

Key words: *power shift, planetary gear, differential mechanical gear, kinematic analyze, force analyze*

INTRODUCTION

The main purpose is to allow the gear change transmission between the engine and the driving wheels so that the motor regardless of the speed to remain high speed at which it has the greatest power. In addition, the transmission line to meet additional requirements.

Through reverse gear must allow reversing the vehicle when driving on a slope provides engine braking when the vehicle included such gear that would do the same slope are entering. While driving in a lower gear provides more flexible ride with less speed vehicle achieves greater acceleration, easier to overtake slower vehicles. All transmission gears are adjusted so that the drive shaft was disconnected from the driven shaft and set the neutral gear [1, 6, 7].

As Vlk writes [5] the transmission is used to change the transmitted torque and its long-term disruption and also to change its meaning (reverse). This is achieved by the transfer, which allows you to change gear. Speed transmissions are either gear toothed spur gears or planetary gearbox with gear wheels [2, 3, 5]. Speed gears belong to a hand with a change of gear off the clutch, the impact of each interrupt transmission of the drive torque. This is a disadvantage when traveling especially to climb for trucks.

Continuous transmission allows continuous change of torque automatically. Applications include automatic transmission with torque converter, which allows multi-plate clutch shift without interrupting torque transmission [4, 5].

MATERIAL AND METHODS

Kinematics analyse of differential planetary gear with differential on output.The body of the differential gear box is elementary planetary gear that consists of sun wheel that is connected with output shaft, satellites 2, ring gear 3 and planet currier 4 that is connected with input shaft. Differential mechanism has planetary gear with two driven elements. The first one is sun wheel and second one is ring gear. The second driven member is by means of gear set z_8, z_7, variable gear ratio iv and gear set z_6, z_5 connected with output shaft. Form the point of view of kinematics is necessary to find relation between gear ratio i_{41}* and secondary gear ratio i_{13}* (Figure 1) according to:

$$i_{41}* = f(i_{13}*) \tag{1}$$

Kinematics analyse of differential gear box was realised on the basis of relations for planetary gear with one degree of freedom. Revolutions of output shaft n_1 are dependence on revolutions of input shaft n_4 and ring gear n_3:

$$n_1 = f(n_4, n_3) \quad \text{rpm} \tag{2}$$

Output revolutions of elementary planetary gear n_1 are:

$$n_1 = \frac{n_4}{i_{41}} - \frac{n_3}{i_{31}} \quad \text{rpm} \tag{3}$$

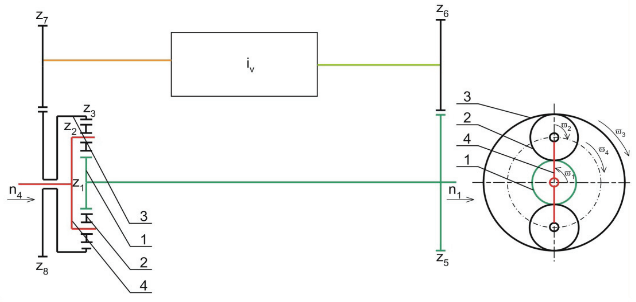

Figure 1. Kinematics scheme of differential planetary gear

Revolutions of ring gear n_3 on the basis of kinematics scheme shown in Fig.1 are:

$$n_3 = \frac{n_1}{i_{13}*} \qquad \text{rpm} \tag{4}$$

Value of secondary gear ratio $i_{13}*$ is:

$$i_{13}* = \frac{n_3}{n_1} = \frac{n_5}{n_6} \cdot i_v \cdot \frac{n_7}{n_8} \tag{5}$$

From the kinematics scheme (Figure 1) is valid:

$$n_1 = n_4$$

$$n_3 = n_7$$

$$i_{56} = \frac{n_5}{n_6}, \qquad i_v = \frac{n_6}{n_7}, \qquad i_{78} = \frac{n_7}{n_8}$$

By substitution to relation 5:

$$i_{13}* = i_{56}.i_v.i_{78} \tag{6}$$

Value of gear ratio $i_{13}*$ according to relation 6 is dependent on value of variable gear ratio i_v (gear ratio i_{65} and i_{78} are constant).

Differential gear ratio $i_{41}*$ is:

$$i_{41}* = \frac{n_4}{n_1} \tag{7}$$

By substitution of the revolutions n_3 from relation 4 to relation 3 revolutions n_1 are:

$$n_1 = \frac{n_4}{i_{41}} - \frac{n_1}{i_{31}.i_{13}*} \qquad \text{rpm} \tag{8}$$

By dividing of relation 8 by revolutions n_1 we can relation 8 to write as following:

$$1 = \frac{n_4}{n_1.i_{41}} - \frac{1}{i_{31}.i_{13}*} \qquad (9$$

The final relation for differential planetary gear $i_{41}*$ than is:

$$i_{41}* = \frac{n_4}{n_1} = \frac{i_{31}.i_{13}*+1}{i_{31}.i_{13}*} \cdot i_{41} \tag{10}$$

Value of differential gear $i_{13}*$ according to the relation 10 is specified by kinematics parameters of the elementary planetary gear (gear ratios i_{31} and i_{41}) and secondary gear ratio $i_{13}*$. Secondary gear ratio is depending on the value of variable gear ratio i_v. Output differential gear ratio $i_{41}*$ is function of variable gear ratio iv according to the following formula:

$$i_{41}* = f(i_v) \tag{11}$$

Force analyse of differential planetary gear with differential on output.

Force analysis of the planetary differential mechanism which is shown in Figure. 2 will be made on the basis of a simple planetary conditions power transfer being preferred using kinematic relationships. Circuit forces the individual members of simple planetary transfer are shown in Fgure 2 from which can be written values of the moments.

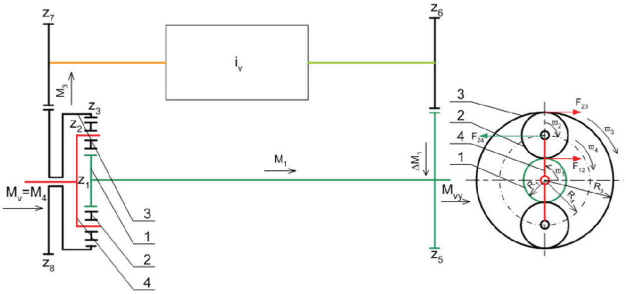

Figure 2. Power ratio differential planetary gear

As can be seen from the power differential planetary gearing ratios shown in Figure 2, the torque input $\boldsymbol{M_v}$ is the same moment $\boldsymbol{M_4}$ operating the ring gear as applying:

$M_v = M_4$ Nm (12)

Output torque will be the sum of $\boldsymbol{M_{VY}}$ moments **-** $\boldsymbol{M_1}$ and $\boldsymbol{M_1}$ can be expressed as follows:

$M_{vy} = M_1 + \Delta M_1$ Nm (13)

Moment $\boldsymbol{M_3}$ is given by:

$M_3 = F_{23}.R_3$ Nm (14)

moreover:

$M_1 = M_4.i_{41}$ Nm (15)

$M_1 = M_3.i_{31}$ Nm (16)

The moment acting on the $\boldsymbol{M_3}$ ring gear can be expressed as follows:

$M_3 = F_{23}.R_3$ Nm (17)

Where the following applies:

$M_3 = M_1.i_{13}$ Nm (18)

$M_3 = M_4.i_{43}$ Nm (19)

The power relationships described in Figure 4 is clear that the torque $\boldsymbol{M_3}$ operating the crown wheel ***3*** is transferred gears $\boldsymbol{z_7}$, $\boldsymbol{z_8}$ through transfer $\boldsymbol{i_v}$ and finally gear wheels $\boldsymbol{z_5}$, $\boldsymbol{z_6}$ (transfer $\boldsymbol{i_{56}}$) to the drive shaft planetary transmission. Then increase the time $\boldsymbol{M_1}$ on the drive shaft planetary gear can be expressed by:

$\Delta M_1 = M_3.i_{56}.i_v.i_{78}$ Nm (20)

After the introduction of secondary transmission ratio of $\boldsymbol{i_{41}}$ *******, which is given by:

$i_{41}{*} = i_{56}.i_v.i_{78}$ (21)

We get:

$\Delta M_1 = M_3.i_{41}{*}$ Nm (22)

Then $\boldsymbol{M_{VY}}$ output torque on the driven shaft planetary gear differential is the sum of the torque moment $\boldsymbol{M_1}$ a $\boldsymbol{M_1}$ growth, which can be expressed as follows:

$M_{vy} = M_1 + \Delta M_1$ Nm (23)

In further addressing power relationships differential planetary transmission is based on the effective moment of $\boldsymbol{M_1}$, which is given by:

$M_1 = M_v.i_{41}$ Nm (24)

Moment at the crown wheel M_3 can be expressed by the following relationships:

$M_3 = M_1.i_{13}$ Nm (25)

or:

$M_3 = M_v.i_{43}$ Nm (26)

Substituting for the value of the torque M_1 we get:

$\Delta M_1 = M_v.i_{43}.i_{41}*$ Nm (27)

Interdependence between the output and the input torque M_{VY} moment M_v is given by the following relationship:

$M_{vy} = M_v.i_{41} + M_v.i_{43}.i_{56}.i_v.i_{79}$ Nm (28)

From the relation (28) shows that in the case of differential planetary gear is a function of output torque M_{VY} variable gear ratio also provided that the other gear ratios are constant.

RESULTS

On the basis of kinematics scheme was created computer model of differential planetary gear box. Transmission mechanism is differential because secondary branch is parallel- connected with primary branch. Secondary branch makes it possible to shift 16 gear speed ratio according to:

$Y = X^n$ (29)

Design and arrangement of basic planetary gear in primary and secondary branch

In primary branch of differential mechanical branch is placed elementary gear (Figure 4). Driving element is planet currier, primary driven element is sun wheel and secondary driven element is ring gear.In secondary branch are in-line connected four planet gears (Figure 5). These gears created variable gear ratio i_v. Driving element each of gears in secondary branch is ring gear and driven element is planet currier. Dependence of gear ratios shown graphically in Figure 6. Depending on the value of output torque to input shown graphical solution in Figure 7.

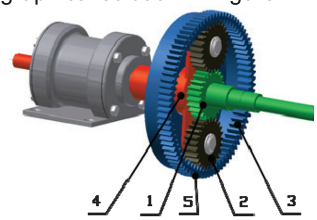

Figure 4. Planetary gear in primary branch
1- sun wheel, 2- satellite, 3- ring gear, 4-planet carrier, 5- gear wheel z_8

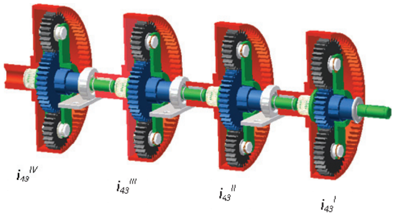

Figure 5. Planetary gears in secondary branch

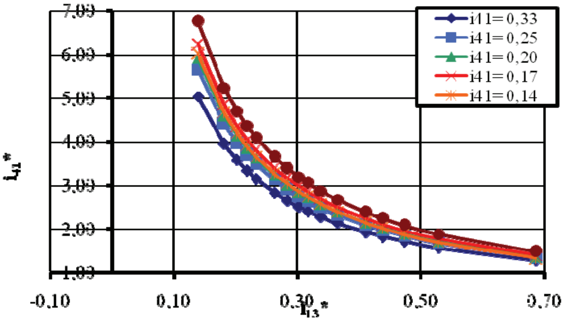

Figure 5. Dependency of primary differential gear ratio i_{41}* in value of secondary differential gear ratio i_{13}*

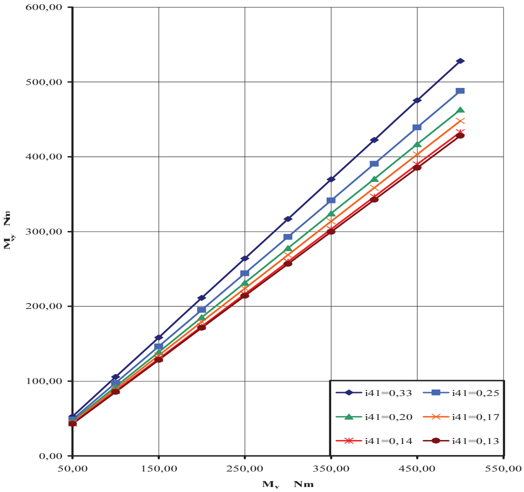

Figure 6. The dependence of the output torque value on M_{VY} input torque M_v value of gear ratio i_{31}*= 1.46

CONCLUSIONS

This paper deals with power and kinematic analysis of planetary gear differential. Analytical solutions based on the fundamental strength and kinematic relations valid for simple planetary gear. Derived force and kinematic relations express the basic functional dependence between the value of the input torque and the output torque and also dependencies between the main and secondary gear ratio. Graphical solutions to power and kinematic conditions are shown in Figures 5 and 6.The analytical solution of the differential planetary gear was prepared computer model of the transmission system and then the prototype gear serves to validate the calculated data.

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LITERATURE

[1] JABLONICKÝ, J. *Diferenciálne prevody a ich aplikácia : dizertačná práca*. [S.l. : s.n.], 2005. 190 s. Doktorandská dizertačná práca (PhD.) SPU v Nitre, 2005.

[2] JABLONICKÝ, J., ŽIKLA, A. Návrh a kinetická analýza diferenciálnej planétovej prevodovky. In 6. medzinárodná vedecká konferencia mladých 2004. Nitra :SPU, 2004.

[3] JABLONICKÝ, J., ABRAHÁM, R., ŠVEC, J. Analýza silového zaťaženia prevodového systému. 2006 In *Traktori i pogonske mašine*. ISSN 0354-9496.

[4] JABLONICKÝ, J., ABRAHÁM, R., ŠVEC, J. Analýza silového zaťaženia diferencionálneho planétového prevodu. In *Acta technologica agriculturae*. ISSN 1335-2555, 2006, roč. 9, č. 4, s. 88-91

[5] VLK, F. Zkoušení a diagnostika motorových vozidel. ISBN 80-234-6573-0, Nakladatelství. VLK, Brno 2000.

[6] ŽIKLA, A., ABRAHÁM, R. JABLONICKÝ, J. Silová analýza diferenciálneho planétového prevodu. In *III. Mezinárodní vědecká konference mladých 2001*. Praha : Česká zemědělská univerzita, 2001. ISBN 80-213-0789-7.

[7] ŽIKLA, A., DRABANT, Š., TKÁČ, Z., JABLONICKÝ, J. Differential transmission gear box with power shift. In *Automotive industry - carrier of technological progress*. Kragujevac : Faculty of mechanical engineering , 2006. ISBN 86-80581-95-X.