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Kinematic Scheme and Design of Automatic Planetary Gear Boxes Based on a New Module

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Abstract

The author has carried out analysis of the existing and projected planetary gear boxes and regarded the prospects of their application and development. New planetary systems called an all-purpose multi-line differential gear train and covered by RF patent №2384773 were developed which possess the possibility to unite as more differential devices as possible in their mechanical diagram. A brassboard module of planetary system was fabricated and based thereon the mechanical diagrams of planetary gear boxes were developed. A mechanical diagram of eight-diameter stepped planetary gear box is presented and calculation of gear transmission ratio with the operation concept is stated. The design of a new eight-diameter stepped planetary gear box was produced.

Keywords: planetary system, all-purpose all-round differential gear train, mechanical diagram, friction lock-up clutch, friction brake

1 Introduction

Application of planetary type gear boxes in automatic transmission lines of land vehicles is stipulated by the possibility of getting a light-sized design, compact in componentry and easily fitting in with the space limited by the body dimensions and providing affordability, acceleration capacity, comfort ability and safety of movement.

Performance standards of the existing and projected planetary gear boxes (PGB) is defined by the following criteria: the number of planetary gear sets, key links, degrees of freedom, control elements, gears and transmissions, balance of transmission units within the required range, shafts ply rating, indices of dimensions and mass, application of frame motion, complexity of design and fabrication, cost parameters. After the analysis of these criteria it is possible to state that producing of PGB most completely satisfying all the above requirements is feasible to implement together with a module – a planetary system being two-staged and formed by minimal quantity of planetary gear sets at minimum numbers of the key links.

Applying a well-known principle of building multiple-reduction mechanical gearboxes due to addition of the divider and dual high transmission to main gearbox is also possible at building planetary type gearboxes if to use two-step planetary system with 4-5 transmissions in the function of the basic transmission (module). With a view of reducing the specific amount of metal per structure and dimensions it is implemented with the minimal key-links number [1, 2].

2 Planetary gear box module

In a fundamental research laboratory «Derivative geared and hydra-mechanical variators» at Kazan Federal University (KFU) new planetary systems [1] have been developed that were called an all-purpose multi-line differential gear train possessing the possibility to unite as more differential devices as possible in their mechanical diagram at minimum numbers of the key links and were covered by RF patent No. 2384773, MPK F16H 3/44 "Self-actuated staged planetary gear box"/Voloshko V.V. , Salakhov I.I. Published March, 20.03.2010, bul. №8.). The patent is aimed at performance build-up of wheeled transport vehicles transmission due to accomplishment of the technical result that lies in: - increasing the range of gear numbers ratio; - providing the balance of transmission units within the required range; - bettering the traction control and dynamic characteristics and fuel consumption thanks to the increase in gear numbers ratio from 4 up to 14; - simplification of gear box control system design and power available per worker; - enhancement of the gear box efficiency; - enlarging the fleet of motor vehicles where a new gear box is possible to install [2]. A brassboard module of planetary system based on all-purpose multi-line differential gear train is presented in figure 1.

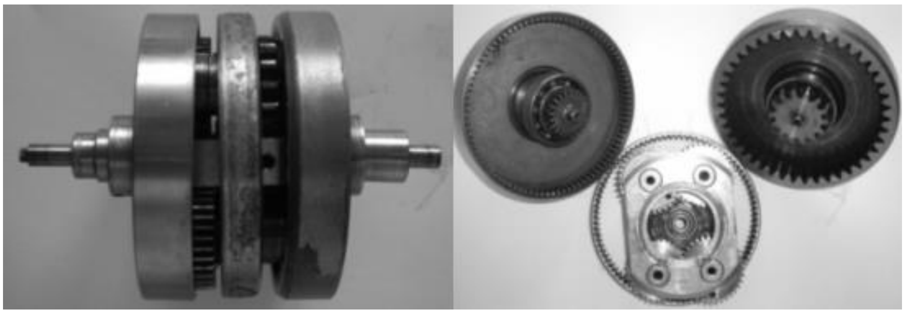

Figure 1: The design of the planetary system of universal multi-threaded differential mechanism

3 Planetary system based on all-purpose multi-line differential gear train

At present various models of mechanical diagrams for four-, eight-, and twelve-staged planetary gear boxes and methods of their kinetic calculation and force computation are presented based on the given module.

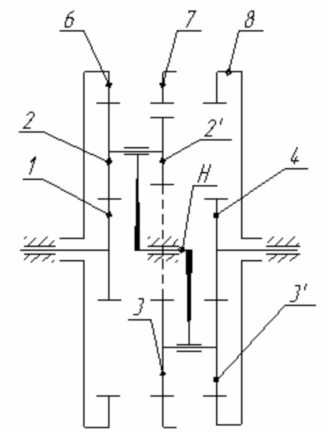

Figure 2: Kinematic diagram of the eight-speed planetary gearbox

A design variation providing for possible series engagement of gear box input element with sun central pinion of the first planetary set and cage with the help of friction lock-up clutch and simultaneous gearing of a key element we get eight passes of gearing: neutral, - reducing transmissions (1st – 5th gears); - direct gear; - reducing transmission (7th -8th gears)); - reverse gear [3].
The planetary gear box (figure 2) implementing eight passes of direct gearing and

one reverse gear comprises a case 10, input link 9, output link 8, five friction brakes 11, 12, 13, 14, 18 and all-purpose multi-line differential gear train (AMDGT). Input links of AMDGT are sun central pinion 1 and cage H (5) that can be coupled by means of friction lock-up clutches 16, 17 with gear box input element 9, sun central pinion 1 is coupled by means of friction brake 18 with gear-box casing.

At friction lock-up clutch 17 being on and series engagement of one of the brakes transmission gear ratio between input shaft 1 and 8 output links is determined by the equation [3]:

$$i_{1-8} = i_{1-H}^{q} \cdot i_{H-8}^{q}, \tag{1}$$

where i_{1-H}^{q} – transmission gear ratio between input shaft 1 and cage when link q is a bit held what is determined from the equation:

$$i_{1-H}^{q} = 1 - i_{1-q}^{q}, \tag{2}$$

i_{H-8}^{q} – transmission gear ratio between the cage and output shaft 8 when link q is a bit held is equal to

$$i_{H-8}^{q} = \frac{1}{1 - i_{8-q}}. \tag{3}$$

At friction lock-up clutch 16 ($i_{1-H}^{q} = 1$) being on and series engagement of one of the brakes transmission gear ratio between input H (5) and output 8 links is determined from the equation:

$$i_{H-8} = i_{H-8}^{q} \tag{4}$$

When friction lock-up clutch 17 is engaged and the cage H (5) is locked by friction brake 12 this mechanism is transformed into ordinary inline reducing gear train and the transmission gear ratio between input link 1 and 8 output links is determined by the equation:

$$i_{1-8} = i_{1-2} \cdot i_{2'-3} \cdot i_{3'-8} \tag{5}$$

At simultaneous engagement of friction lock-up clutches 16, 17, link 1 and link H (5) of all-purpose multi-line differential gear train get the same rotating speed, the device is blocked and it begins working as a rigid shaft realizing a direct gear.

The given planetary gear box is a three-staged one since simultaneous engagement of two control elements is required to get any certain gear [3].

The planetary gear box works as follows: after the motor vehicle has commenced moving its acceleration up to the required speed is implemented by the sequential gearshift at gear changing from the first up to the eighth gear. When friction control elements being out the self-actuated staged planetary gear box is in neutral position.

The first gear being on friction lock-up clutch 17 is engaged coupling input link 9 with sun central gear 1 and friction brake 12 coupling the cage H (5) with the self-actuated gear box housing. This being done, the cage H (5) angular rate (rotary speed) is equal to zero and the mechanism of all-purpose multi-line differential gear train works as an ordinary inline reducing gear train where central gear 1 acts in the function of an input link.

At changing to the second speed forward friction lock-up clutch 17 is shut down and friction lock-up clutch 16 is engaged, friction brake 12 is out and friction brake 13 is on. This being done, angular rate of the crown wheel 7 comes to zero.

At changing to the third speed forward friction lock-up clutch 16 is out and friction lock-up clutch 17 is on, friction brake 13 is turned off and friction brake 11 turns on. Upon that, angular rate of the crown wheel 6 comes to zero.

At changing to the fourth speed forward friction lock-up clutch 17 remains to be on, friction brake 11 shuts off and friction brake 14 turns on. At this time angular rate (rotary speed) of the sun central gear 4 becomes equal to zero.

When changed to the fifth speed forward friction lock-up clutch 17 turns off and friction lock-up clutch 16 is engaged, friction brake 14 shuts down and friction brake 18 is on. At this time angular rate (rotary speed) of the sun central gear 1 becomes equal to zero.

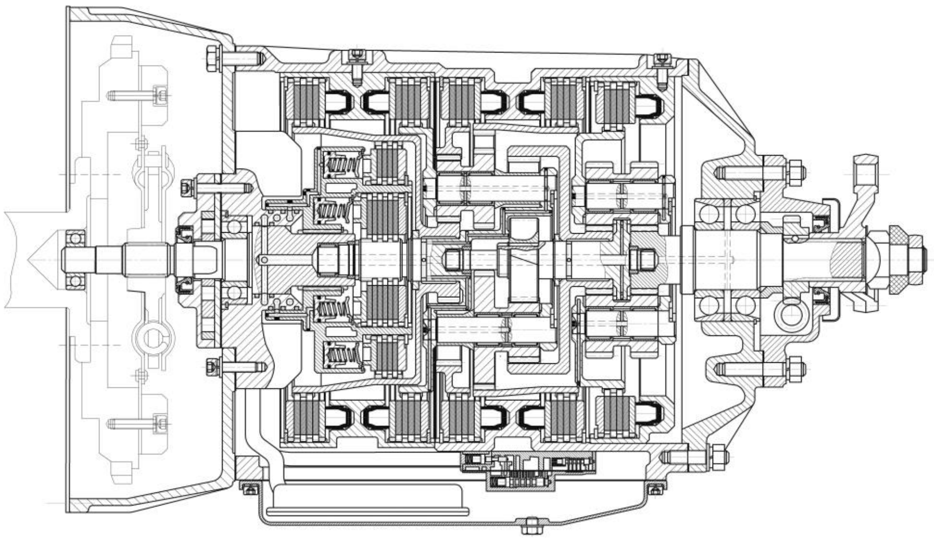

Figure 3: The design of the eight-speed planetary gearbox on the basis of universal multi-threaded differential mechanism

At changing to the sixth speed forward friction brake 18 is off, friction lock-up clutch 16 remains to be on and friction lock-up clutch 17 is also engaged.

When changed to the seventh speed forward friction lock-up clutch 17 shuts down, friction lock-up clutch 16 remains to be on and friction brake 14 turns on. At this time angular rate (rotary speed) of the sun central gear 4 becomes equal to zero.

When changed to the eighth speed forward friction lock-up clutch 16 remains to be on, friction brake 14 turns off and friction brake 11 turns on. This being done, angular rate (rotary speed) of the crown wheel 6 comes to zero.

At switching-in motion reverse gear friction lock-up clutch 17 and friction brake

13 are engaged. This being done, angular rate (rotary speed) of the crown wheel 7 of the second planetary set becomes equal to zero.
The engineering design of eight-staged planetary gear box for a rear drive vehicle is presented in figure 3.

5 Conclusion

Application of AMDGT planetary system in the function of a module makes possible to meet the challenge of synthesis for not only two-staged but also three-staged planetary gear boxes either at enlarging the number of control elements or additional planetary set. Not one but two links of AMDGT planetary system (sun central gear of the first planetary set and the cage) are used in the capacity of the leading link. Undoubtedly high efficiency is received due to short kinematic sequences and juxtaposition of two motions one of which being bulk motion. Considerable reduction of energy consumption for pressurization of PGB hydraulic steering system is implemented by way of friction lock-up clutches minimal use because their hydraulic systems are the places of maximal losses of hydraulic fluid consumption due to rotary seals. Gear changing is realized with no interruptions in torque delivery thanks to minimal number of synchronously switching control elements. The dimensions of the gear box are reduced owing to minimal number of the main links combined in AMDGT planetary system by means of the mutual cage as well as by virtue of less number of control elements at the same number of gears.

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