Volodymyr Malashchenko, Oleh Strilets*, Volodymyr Strilets*

Lviv Polytechnic National University, Lviv, Ukraine

National University of Water and Environmental Engineering, Rivne, Ukraine

FUNDAMENTALS OF CREATION OF NEW DEVICES FOR SPEED CHANGE MANAGEMENT

Abstract. The article deals with problem of speed change management in engineering, i.e., hoisting, building, road and land reclamation machines, cars and tractors, machine tools and other equipment. The review of the methods and devices for stepped, steppless and combined speed change process management have been performed. Their major disadvantages have been identified.

Among others, main disadvantages of stepped speed management devices are the complexity of design, their large material consumption, the emergence of dynamic loads during the transition from one speed to another, even in case of synchromesh using. Continuously variable speed devices have intense components wear due to the use of friction connections like brakes or lock-up clutches.

The authors lead to the conclusion about the need for new devices of speed change management to remove, or at least decrease, identified deficiencies.

New devices suggested for speed changes management via epicyclic gear trains with a closed circuit hydrosystem. Those new constructions are supposed to remove abovementioned shortcomings and can be the basis and foundation for new devices creation and their further research.

Introduction

Functioning of hoisting, building, road and land reclamation machines, cars and tractors, machine tools and other equipment requires the speed changes management of executive mechanisms for different processes. In the world of technology, there are widely known methods and devices for speed management that provide stepped or continuously variable change in velocity by the value and direction with the use of respective gearboxes. However, these methods have many shortcomings. The main disadvantages of stepped speed management devices are the complexity of design, their large material consumption, the emergence of dynamic loads during the transition from one speed to another, even in case of synchromesh using. On the other hand, continuously variable speed devices have intense components wear due to the use of friction connections like brakes or lock-up clutches. All abovementioned reduces the durability and reliability of certain device components and machine in general.

Problem statement

Based on the review of existing devices for speed change management in technique and analysis of their shortcomings the challenge is to create a new speed controlling devices, which eliminate identified deficiencies.

Analysis of modern information sources on the subject of the article

Recently, review of the methods and devices for speed change process management have been performed and their extensive analysis has been held [1–6]. The new classification for those methods and devices has been offered [7]. The possibility of speed change management via epicyclic gear train has been justified [8]. The new patent-protected devices for speed change management via epicyclic gearing with closed circuit hydrosystem have been developed. Nevertheless, for the implementation of any mechanical means, the comprehensive study of kinematic and power parameters is required, but currently fulfilled insufficiently.

Statement of purpose and problems of research

In this article, an attempt was made to lay the foundations for the creation of devices for speed change process management through epicyclic gearing with closed circuit hydrosystem. Among the advantages of these devices, one can highlight the possibility of speed change process automation, dynamic loads and wear reduction due to the absence of friction connections and simplified design..

The main material presentation

Among others, world machinery uses stepped gearboxes as a speed change devices. For example [1], classical tractor MTZ-50 has a nine-speed transmission (Fig. 1). It consists of frame 1, shafts: primary 4, intermediate 3, secondary 15, downshift, also the shifting mechanism and gears. Transmission frame is made of cast iron separately from back axel frame. Primary and secondary shafts arranged coaxially. The secondary shaft rotates on tapered roller bearings, and the rest are on the ball bearings. The posterior bearing of the secondary shaft is mounted on bushing in the rear of the transmission frame. Gaskets 14 are set between the bushing flange and frame surface for axial gap regulation in tapered roller bearings. The secondary shaft 15 is made integrally with the driven gear of gearbox first step. Its internal gearing is designed for direct speed (IX) shift. Driven gear of gearbox second step driving bevel pinion of the main gear are placed at the secondary shaft splines. By picking thickness of the washer 16, bevel pinion set after, a certain relative position of this bevel pinion and transmission rear plane can be provided. On end face of the secondary shaft there is boring hole with the bearing pressed into, which is to be the rear support of primary shaft.
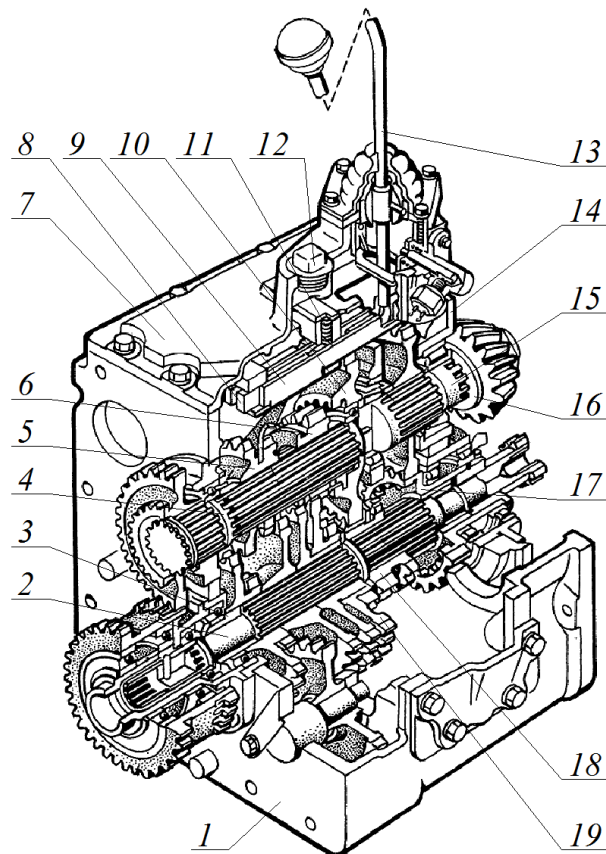

Fig. 1. *MTZ-50 transmission*

The intermediate shaft is hollow. Inside of it, there is a shaft 2 of the independent drive of PTO (power takeoff). The set of gears is mounted on the splines of the intermediate shaft. A lock ring fixes first four of them axially. Fifth and sixth gears 19 revolve feely about fourth one. Seventh and eighth gear set is able to move along shaft splines. Moving forward shifts the first speed of the gearbox, when backward move engages second speed. The second speed shifting mechanism includes gear 17 with both internal and

external gearing. On its end face extending beyond the gearbox frame there is gearing provided for PTO synchronizing device. Inside the gear there is a needle bearing - rear support of the intermediate shaft 3. The gear itself is mounted on two roller bearings at the gearbox frame.

Two mobile sets of gears are set on the splines of primary shaft 4. The forward movement of the front gear set shifts either fifth or eight speed, backward - forth ore seventh speed.

Rear gear set has three operating positions. The front position engages third or sixth speed, the rear one is for ninth (direct) speed. The middle position serves for rotation transition to the downshift shaft 6 through gear 7. On the splines of the shaft 6, there is a gear set 5. Its backward move shifts first or second speed, when opposite will engage the reverse gear.

Intermediate reverse gear 2 rotates about fixed axle and is constantly meshed with front gear 4 of intermediate shaft.

The shifting mechanism consists of lever 13, crosshead 8 with forks, locking plates 9 and retainers 11. Every shifting fork actuates one gear set and welded to its own crosshead. Locking plates prevent double shifting and ball retainers ensure crossheads defined movement.

The truck-mounted crane KS-3561 [1] gearbox serves for operations speed change. It is two-speed constantly meshed-gears gearbox (Fig. 2) manufactured as an independent aggregate and placed on slewing unit of the crane.

The rotation inputs goes from the intermediate bevel gearbox through a gear coupling to the driving shaft 2. The driving gear 3 is mounted at the driving shaft on splines. Further rotation is transmitted to the intermediate gear 4 which is set on the intermediate shaft together with other gear 20. Rings and sleeve prevent axial movement of abovementioned gears. There is a permanent meshing between gear 4 and main shift gear 11, same as gear 20 and downshift gear 19. Gears 11 and 19 are freely set on shaft 7 and separated with toothed hub 18, which in its turn meshes with shafts 7 splines internally, and switching clutch 16 externally.
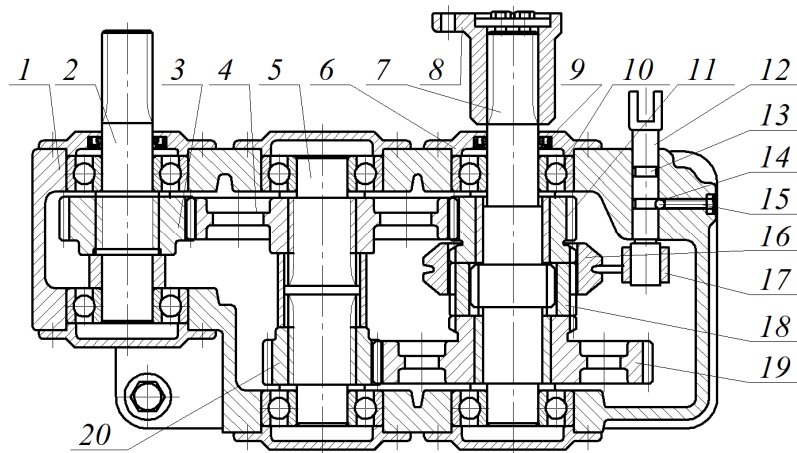

Fig. 2. *Crane KS-3561 gearbox*

In extreme top position clutch meshes with main shift gearing 11 and rotation is transmitted to shaft 7 though gear 4 and hub 18.

Downshifting is executed via gear 20 and hub 18. This time clutch has its extreme low position and meshes with gearing 20. Displacement of the clutch is performed by fork 17 connected to lead 12. Lead 12 is fixed in extreme positions by ball retainer 14 that can be regulated by screw 15. Further motion is transmitted to the driveshaft that connects to the flange 8.

All shafts are mounted in gearbox frame 1 on ball bearing 10. Cups 9 prevent bearing 10 from axial movement and protect inner cavity from environment harmful effect. Cup has an indicator that shows oil level. In addition, the openings for oil refill and drain provided.

Five-speed automobile gearbox of MAZ-5335 [1] shown on Fig. 3 has the following components: crankcase, driving shaft, intermediate shaft with gears, driven shaft with gears and synchromesh, shifting mechanism.
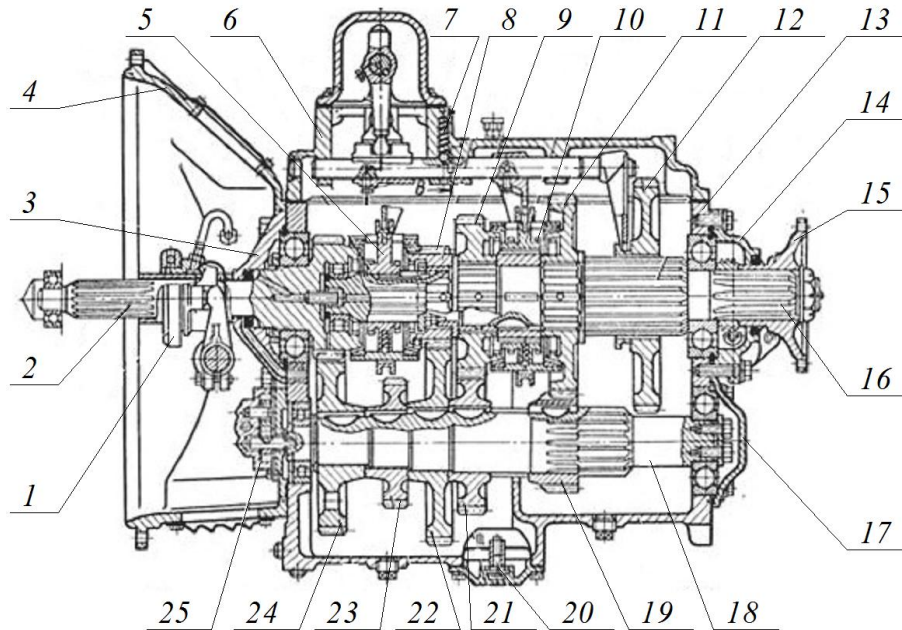
Fig. 3. *MAZ-5335 automobile gearbox*

Driven shaft 2 mounted at front side of crankcase 13 on ball bearing has splines for clutch disk setup at its front end and at other end – gear that permanently meshes with mate gear 24 set on intermediate shaft 18. Gears 8, 9 and 11 of the driven shaft 16 are mounted freely and mesh with the respective gear from intermediate shaft 18. In case of second, third or fifth speed shift, gear-shaft locking if executed via synchromesh 5 and 10. First speed and reverse shifts by gear 12 longitudinal displacement across driven shaft 18.

Steel bearing bushing of driven shaft 16 gears are lubricated under pressure from the pump 25. Oil is fed from the pump through channels in bearing cup, axial cavity in shaft 16 and radial channels leading to every bushing. Gears are lubricated by crankcase oil sprinkling.

Smooth shifting of speed is secured by inertial syncromeshes. Thus, synchromesh 10 shifts second and third speed, when synchromesh 5 shifts fourth and fifth speed.

Fig. 4 shows the construction of screw-cutting lathe gearbox [2]. This gearbox operates as follows. The workpiece is held in chuck attached to the spindle 13. Rotation from the engine 1 is transmitted through belt transmission 2 and clutch 3 to shaft 5.
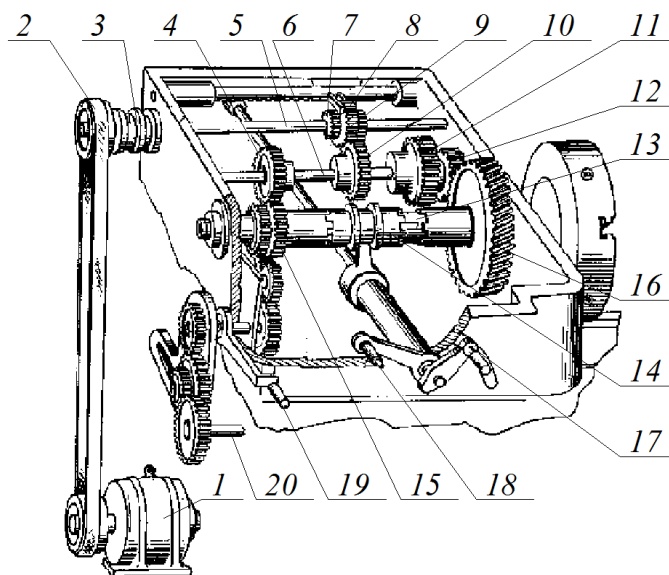

Fig. 4. *Screw-cutting lathe gearbox*

Set of three gears 7, 8 and 9 mounted on shaft 5 is connected with lever 17 by rack-and-pinion system. With help of this lever, gear set engages with gear 4 or 10 or 11 fixed on shaft 6. Gears 4 and 12 mesh respectively with gear 15 and 16 which transmit torque to spindle through gear coupling 14 connected to lever 18. In case of coupling right position, the spindle obtains its rotational movement from gear 16 and in left coupling position - from gear 15. Therefore, gearbox provides six spindle rotation speeds.

Fig. 5 shows the continuously variable transmission (CVT) with variable-diameter pulley. It can be used for metal-cutting lathe drives modernization. At the bottom of transmission frame 6 there is an engine 12 with hollow splined shaft 1. Right sheave 11 of splited V-belt pulley is fixed on the shaft 1. Left sheave 5 can move along splined shaft 1.
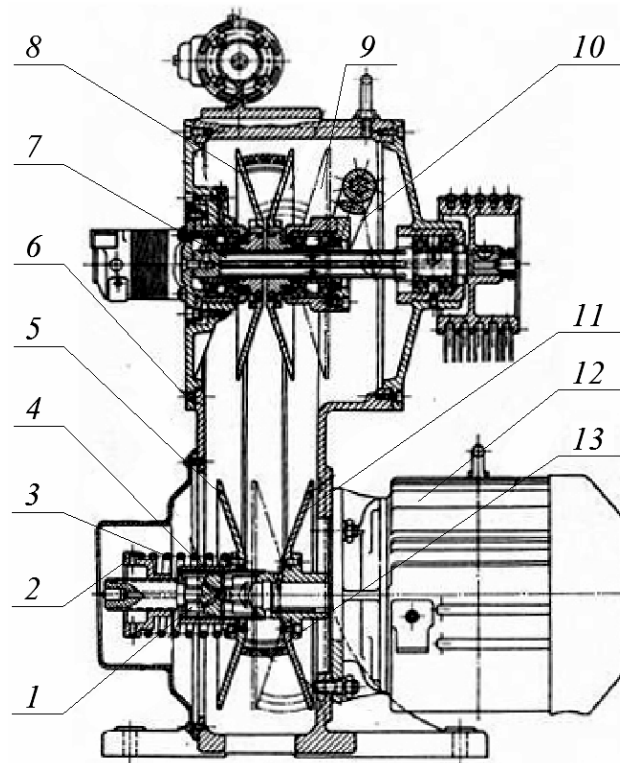

Fig. 5. *Variable-diameter pulley drive*

At the top of CVT there is a driven shaft 7. Here we have left sheave 8 fixed when right one 9 can move. Both driving and driven pulleys are connected with wide V-belt 13. Spring 3 keeps the belt under tension. Smooth rotation speed change of the CVT driven pulley is provided by the forced moving of sheave 9 so that V-belt changes its relative position causing change of ratio.

The other kind of CVT - toroidal, is shown on Fig. 6. Toroidal CVT can provide a wide range power transfer from 2 to 1000 kW and up to 6 ratio. The frame 17 has two hubs 3 and 22 pressed into and hollow shafts 9 mounted there on roller bearings 8. At the adjacent cone ends of the shafts 9 toroidal pulleys 11 and 14 are set on Woodruff keyed joints. The opposite end of the shaft bell-shaped with inward-facing teeth, which carry moving hubs 7 and spherical cogged washers 6 with grooves for balls 5. In the cavities of hollow shaft plugs 15 and anchor spring 10 are mounted. Electric engine 19 with ball coupling 21 is installed on the side of the CVT frame via intermediate flange 18. On the other side of the transmission frame on the outside of hub 3 on the ball bearings two roller 4 are mounted. The inner end of the latter is designed as a ball coupling 21 sheave. At the outer end of the roller 4 there is output pulley 1. Between the frame bottom and lid 16 on turning pin 12 two intermediate spherical disks 13 are installed in contact with toroidal pulleys 11 and 14. Spherical disc can rotate relative to toroidal pulleys while rotating on pin.

On Fig. 7 one can see another CVT construction - torque converter [4, 5] or hydrotransformer. It is used widely in hydrodynamic transmissions of trucks, buses or tractors.
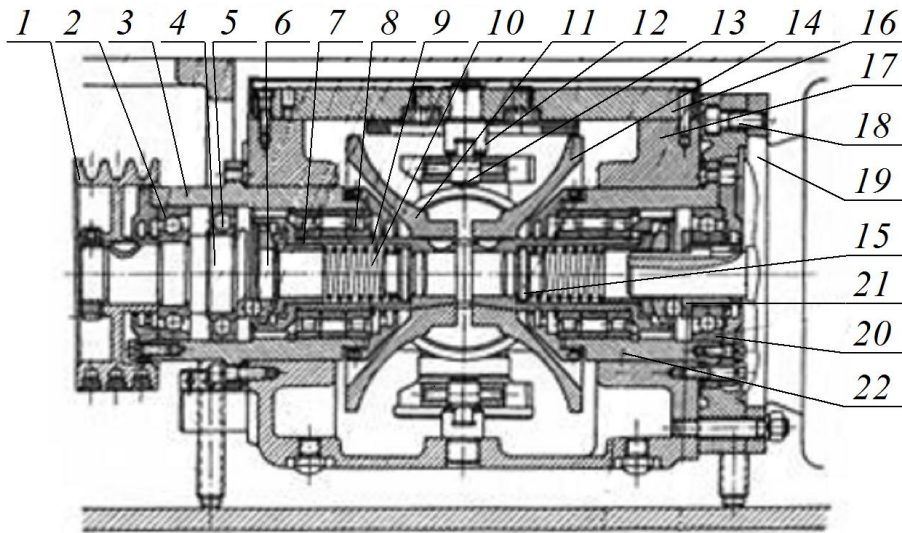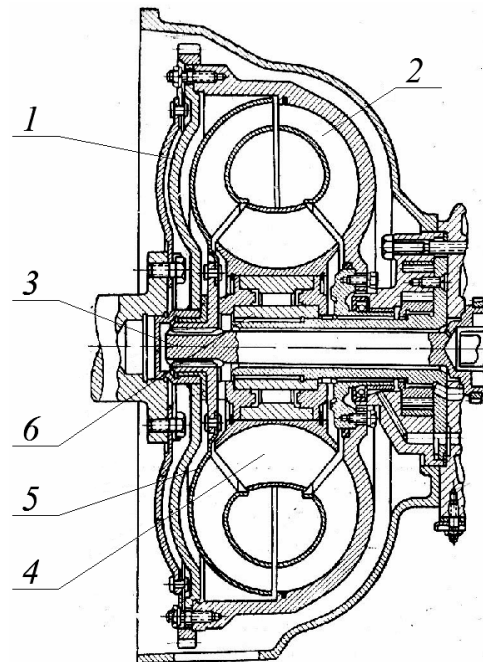
Fig. 6. *Toroidal CVT*

Fig. 7. *GAZ-13 Torque converter*

Torque converter consists of impeller 2, stator 4, turbine 5 and locking mechanism. All components are assembled in frame situated next to automobile prime mover. It is filled with liquid that continuously mixing inside. Impeller 2 is rigidly connected with the torque converter frame; during the rotation of the motor shaft, it creates a flow of fluid that rotates the stator 4 and turbine 5. The stator 4 is connected with the impeller 2 through overrunning clutch. When the difference in speed between impeller and turbine is big, stator is automatically blocked and sends to the impeller greater volume of fluid. Due to the stator, the torque increase up to 3 times at starting.

Because of the absence of rigid kinematic connection during the torque transmission, shock loads on the transmission are excluded and the automobile gets great smooth ride. The negative effect is the "slipping" of turbine relative to the impeller which leads to increased heat (some torque converter modes may produce more heat than the engine itself) and increased fuel consumption.

In order to increase fuel efficiency modern designs of torque converter introduce locking mechanism that allows to bind impeller and turbine rigidly. Locking turns on automatically when reaching sufficient speed (usually over 70 km / h).

Ford was the first who used CVT in form of torque converter with epicyclic system [3] in the front-wheel drive ATX transmission. In this design, the property of epicyclic system to add on output unit the power supplied to its two inputs is used.

The design of torque converter with epicyclic system (Fig. 8) resembles the design of one with friction locking clutch. The casing using bolts fastens to a flange of the crankshaft of the engine. There is also the damper attached inside, which serves to absorb engine torque oscillations. Built-in epicyclic system in torque converter is designed to add power of two streams, one of which comes directly from the engine, and other pre passes torque converter 1.
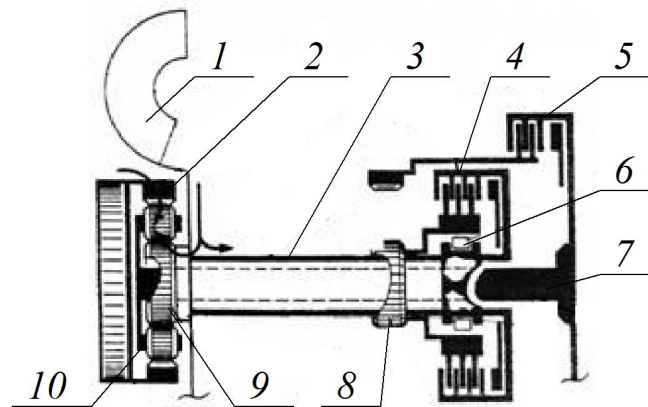

Fig. 8. Torque converter with epicyclic system

Central gear 2 of epicyclic gear train through splines connected to the torque oscillations damper, and therefore its angular velocity is always equal to the speed of rotation of engine shaft.

Sun gear 9 of this gear train is also the driving element, and the power from the torque converter 1 enters the epicyclic system through it.

Power adding is carried out on the carrier 10, which is connected via splines with intermediate shaft of the gearbox.

On the first speed, overrunning clutch 6 operates in active mode and coupling 5 of the second speed is off because the power from the engine completely passes through the torque converter 1 and bypassing the epicyclic system, it goes through overrunning clutch 6 at one of the two central gears 8 of the planetary gearbox.

When you shift to second speed clutch 5 the overrunning clutch 6 operation becomes inefficient and friction clutch of direct transmission 4 remains off. As a result, 62 % of the engine power goes through torque converter 1 and gets to the sun gear 9, when 38 % gets to the central gear 2 through damper from the crankshaft of the engine. Power adding is held in epicyclic system of torque converter 1, and then all the power goes to the gearbox via intermediate shaft 7.

At the third speed friction clutches of the second speed 5 and direct speed 4 are off. Planetary gearbox locked, and its ratio equals to one. Epicyclic system of torque converter 1 is also practically blocked. Total ratio equal to one. In this case, about 93% of the power from the engine goes to central gear 2, and 7 % to the sun gear 9. At the reverse second speed clutch 5 is off and all power goes through torque converter 1. To provide the rigid kinematic connection between input and output shafts in the planetary gear train it is necessary to use certain combinations of controls, which are friction elements as in CVT: belt, friction disk, overrunning clutches.

Fig. 9 shows the scheme of CTV Asian Warner A140E, [3].

In studies [11-25], the new devices for speed changes management in machines due to the epicyclic gear trains with a closed circuit hydrosystem are reviewed. One of such devices is shown on Fig. 10.

Here on Fig. 10 you can see the scheme of a simple epicyclic gear train with a speed change device, which consists of sun gear 1, planets 2, ring gear, 3 (annulus) and carrier 4 mounted in framework 5. The closed circuit hydrosystem 6 attached to the framework 5 and engaged with the control gear, i.e. annulus, through the gearing 7. Hydrosystem 6 consists of hydraulic pump 8, pipelines 9, regulating throttle valve 10, check valve 11 and fluid reservoir 12.
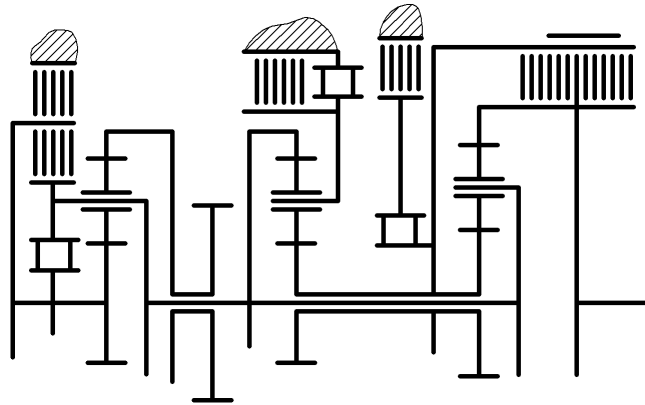

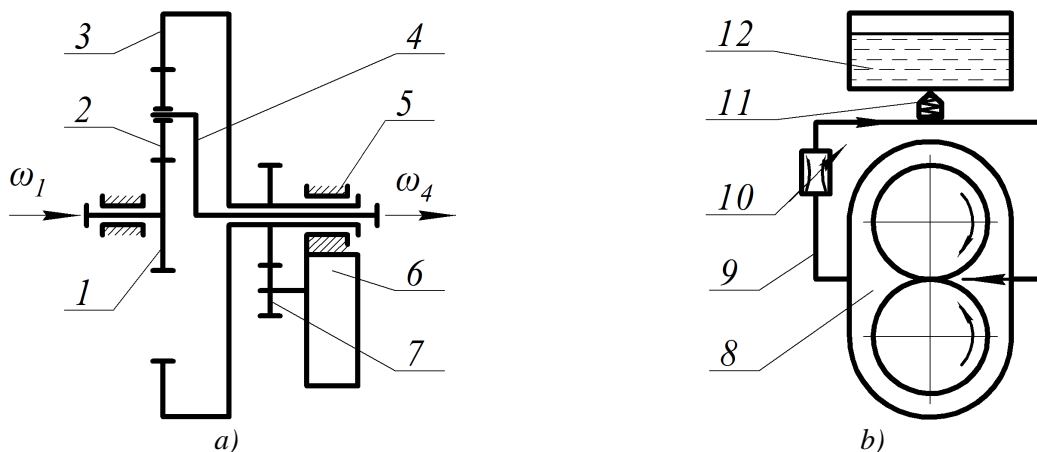**Fig. 9.** *Scheme of continuously variable transmission Asian Warner A140E*

Fig. 10. *Schemes: a – epicyclic gearing with closed circuit hydrosystem where sun gear used as inputs and annulus – control element; b – closed circuit hydrosystem*

In this case, speed change of the driven element – carrier 4, is performed via annulus 3. Sun gear 1, being a driving element, revolves with angular velocity $\omega_1 = const$, so by means of closed circuit hydrosystem 6 it is possible to change smoothly the velocity of the driven element – carrier 4 (ω_4). Annulus 3, through the gearing 7, sets in motion the pump 6, which pumps the fluid in closed circuit hydrosystem 6 consisting of pipelines 9 and so far open throttle valve 10. If we close the regulating throttle valve 10 – it will stop the flow in hydrosystem 6, the pump 6 will turn immovable and so will the annulus 3 ($\omega_3 = 0$). Thus, depending on the bandwidth of regulating throttle valve 10, angular velocity of annulus 3 (ω_3) changes from zero point to certain maximum value ω_{3max}, while changing the speed and possibly rotation direction of the carrier 4.

Conclusions

The analysis of stepped and CVT devices for speed changes management allow to draw following conclusions.

1. Considered devices in form of gearboxes are complex because of a large number of gears, costly in material and during the transition from one speed to another having large dynamic loads.

2. The proper use of a large number of gear transmission in high traffic, with frequent starting and stopping the car, steep descents and ascents, car operator should put a lot of strength and have high qualifications.

3. In existing designs of the stepped transmissions, it is difficult to implement automation of drives of wheeled, tracked vehicles or other equipment.

4. The main disadvantages of continuously variable speed devices are the intense components wear due to the use of friction connections like brakes or lock-up clutches, the complexity of the design.

5. Suggested devices for speed changes management due to the epicyclic gear trains with a closed circuit hydrosystem supposed to remove abovementioned shortcomings and are the basis and foundation for new devices creation and their further research.

References

[1] Грабар А. М. Атлас конструкцій агрегатів, вузлів та систем автомобіля / А. М. Грабар, Є. М. Рибалкін, Є. Г. Опанасюк, А. В. Ільченко. – Житомир : ЖІТІ, 1998. – 288 с.

[2] Кучер А. М., Металлорежущие станки / А. М. Кучер, М. М. Киватицкий, А. А. Покровський. – М. ; Л. : Машиностроение, 1985. – 284 с.

[3] Харитонов С. А. Автоматические коробки передач / С. А. Харитонов. – М.: Астрель АСТ, 2003. – 280 с.

[4] Петров А. В. Планетарные и гидромеханические передачи колесных и гусеничных машин / А. В. Петров. – М.: Машиностроение, 1966. – 383 с.

[5] Петров В. А. Гидрообъемные трансмиссии самоходных машин / В. А. Петров. – М.: Машиностроение, 1988. – 248 с.

[6] Стрілець О. Р. Огляд і аналіз способів керування змінами швидкості у техніці / О. Р. Стрілець // 12-й Міжнародний симпозіум українських інженерів-механіків у Львові: тези. – Львів : КІНПАТРІ ЛТД, 2015. 28–29 травня 2015 року. – С. 152–153.

[7] Малащенко В. О. Класифікація способів і пристроїв керування процесом зміни швидкості у техніці / В. О. Малащенко, О. Р. Стрілець, В. М. Стрілець // Підйомно-транспортна техніка. – Одеса. – 2015. – № 3. –С. 70–78.

[8] Стрілець О. Р. Обґрунтування можливості керування змінами швидкості за допомогою диференціальних передач / О. Р. Стрілець // Вісник Інженерної академії України. – К. : НАУ, 2015. – № 2. – С. 177–181.

[9] Пат. 2211796 Российская Федерация, МПК F 16 D 57/06. Останов для груза перемещаемого механизмом подьема / Куденко Н. М., Стрелец В. Н.; заявитель і обладатель патента Украинский государственный университет водного хазяйства и природопользования. – № 2001107699; заявл. 21.03.2001; опубл. 10.09.03, Бюл. № 25.

[10] Пат. 2211797 Российская Федерация, МПК F 16 D 57/06. Останов для груза перемещаемого механизмом подьема / Куденко Н. М., Стрелец В. Н.; заявитель і обладатель патента Украинский государственный университет водного хазяйства и природопользования. – № 2001113324/28; заявл. 14.05.2001; опубл. 10.09.03, Бюл. № 25.

[11] Пат. 7328 Україна, МПК F 16 H 48/20. Зубчастий диференціал з пристроєм для керування змінами швидкості / Стрілець В. М., Ріло І. П., Стрілець О. Р.; заявник і власник патенту Національний університет водного господарства та природокористування. – № 20041109781; заявл. 29.11.04; опубл. 15.06.05, Бюл. № 6.

[12] Пат. 11121 Україна, МПК F 16 H 1/28. Зубчастий диференціал з пристроєм для керування змінами швидкості / В. М. Стрілець, І. П. Ріло, О. Р. Стрілець, В. П. Поліщук; заявник і власник патенту Національний університет водного господарства та природокористування. – № u 2005 04847; заявл. 23.05.05; опубл. 15.12.05, Бюл. № 12.

[13] Пат. 11267 Україна, МПК F 16 H 1/46. Спосіб зв'язку між двигуном внутрішнього згорання і трансмісією автомобіля / І. П. Ріло, В. М. Стрілець, О. Р. Стрілець, А. В. Федитник; заявник і власник патенту Національний університет водного господарства та природокористування. – № u 2005 05938; заявл. 16.06.05; опубл. 15.12.05, Бюл. № 12.

[14] Пат. 11398 Україна, МПК F 16 H 1/46. Спосіб зв'язку між двигуном і трансмісією автомобіля / І. П. Ріло, В. М. Стрілець, О. Р. Стрілець, Ю. М. Сорока; заявник і власник патенту Національний університет водного господарства та природокористування. – № u 2005 06656; заявл. 07.07.05; опубл. 15.12.05, Бюл. № 12.

[15] Пат. 14701 Україна, МПК F 16 H 1/00. Спосіб зв'язку між двигуном і трансмісією автомобіля / Ріло І. П., Стрілець В. М., Стрілець О. Р.; заявник і власник патенту Національний університет водного господарства та природокористування. – № u 2005 12107; заявл. 16.12.05; опубл. 15.05.06, Бюл. № 5.

[16] Пат. 18514 Україна, МПК F 16 H 48/00. Зубчастий диференціал з пристроєм для керування змінами швидкості / В. М. Стрілець, І. П. Ріло, І. Т. Шинкаренко, О. Р. Стрілець; заявник і власник патенту Національний університет водного господарства та природокористування. – № u2006 04863; заявл. 03.05.06; опубл. 15.11.06, Бюл. № 11.

[17] Пат. 18587 Україна, МПК F 16 H 48/00. Зубчастий диференціал з пристроєм для керування змінами швидкості / О. Р. Стрілець, В. М. Стрілець, І. Т. Шинкаренко; заявник і власник патенту Національний університет водного господарства та природокористування. – № u200605205; заявл. 12.05.06; опубл. 15.11.06, Бюл. № 11.

[18] Пат. 25335 Україна, МПК F 16 H 1/28. Зубчастий диференціал з пристроєм для керування змінами швидкості / Стрілець О. Р.; заявник і власник патенту Національний університет водного господарства та природокористування. – № u 2005 04847; заявл. 23.01.07; опубл. 19.08.07, Бюл. № 12.

[19] Пат. 28463 Україна, МПК F 16 D 71/00. Вантажоупорний зупинник / Стрілець О. Р., Стрілець В. М.; заявник і власник патенту Національний університет водного господарства та природокористування. – № u200708884; заявл. 01.08.07; опубл. 10.12.07, Бюл. № 12.

[20] Пат. 28489 Україна, МПК B 60 K 17/06. Планетарна коробка передач / Стрілець О. Р.: заявник і власник патенту Національний університет водного господарства та природокористування. – № u 2007 09132; заявл. 09.08.07; опубл. 10.12.07, Бюл. № 20.

[21] Пат. 41191 Україна, МПК F 16 D 71/00. Вантажоупорний зупинник / Куденко М. М., Стрілець В. М.; заявник і власник патенту Рівненський державний технічний університет. – №2001032016; заявл. 27.03.01; опубл. 15.07.03, Бюл. № 7.

[22] Пат. 44135 Україна, МПК B 66 D 5/32. Вантажоупорний зупинник / Куденко М. М., Стрілець В. М.; заявник і власник патенту Український державний університет водного господарства та природокористування. – №2001053400; заявл. 21.05.01; опубл. 15.03.05, Бюл. № 3.

[23] Пат. 48608 Україна, МПК F 16 H 48/00. Зубчастий диференціал з пристроєм для керування змінами швидкості / Стрілець О. Р., Стрілець В. М.; заявник і власник патенту Національний університет водного господарства та природокористування. – № u 200910172; заявл. 07.10.09; опубл. 25.03.10, Бюл. № 16.