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Increasing the Vehicle's Dynamic Performance by Developing a Continuously Variable Transmission

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Abstract: To improve the traction and dynamic characteristics of small class cars, this study proposes the modernization of the power unit. The power unit includes a powerplant, clutch and gearbox. This study proposes a modernized driven pulley to increase the range of transmission ratios. The calculation of the gearbox to include the rear gear, the kinematic calculation of the V-belt transmission. Modernization of V-belt transmission was to create a two-section driven pulley with automatic locking of the outer section relative to the inner, increasing the diameter of the driven pulley compared to analogues.

Key words: Car, transmission, gearbox, variator, traction-dynamic characteristics, V-belt

INTRODUCTION

When traveling by car in the city, there is a need for frequent breaks and stops associated with traffic conditions. Frequent gear changes and disengagement reduce driver attention. Frequent periodic loads also increase wear on transmission mechanisms and lead to increased fuel consumption. As a solution to these problems, the use of an upgraded CVT transmission is proposed. The solution to the problem of reducing fuel consumption and increasing traction and dynamic performance is relevant primarily for urban cars (Grishkevich *et al*., 1985).

Many cars of the middle and business class are equipped with a continuously variable transmission. Small cars are practically devoid of such an opportunity as the variator gearbox has large longitudinal dimensions. In this regard, there are difficulties in its placement in the engine compartment. To improve the traction and dynamic characteristics of the car is necessary:

- Reduce the size of the continuously variable transmission
- Increase the ratio of the variator

Earlier in the variators used klinoremennaya, chain, rarely tore friction gear. Chain and V-belt transmission have a similar design, differ only in the driving link. To ensure performance, a maximum gear ratio of 3, ..., 3.6 is required. Previous designs of variators could realize a gear ratio of 2, ..., 2.5. In order to achieve a higher gear ratio corresponding to the first gear, planetary gears are installed in the case of a variable-speed gearbox. For additional mechanical gears, body expansion is required. And because of this, friction losses increase and the transmission efficiency decreases (Kosenkov, 2003; Pronin and Revkov, 1980). As a result of marketing research, the following variators used in the automotive segment were identified.

Mercedes Autotronic 722.8 variator is installed only on A-Class and B-Class models; CVT Jatco CVT diagram. These variators are produced by the Japanese company JATCO. Since, 2007, installed on cars brands Jeep Compass, Patriot and Dodge Caliber. First, the Jatco CVT diagram CVT was tested on small Japanese cars (Serena, Altima) and European (Renault - Clio) with a small engine size from 1.3-1.8 L.

CVT Subaru lineartronic is also a Japanese development. It is applied on cars of an average and business class. Table 1 provides an assessment of the technical level of the described variators as well as the identified advantages and disadvantages.

This study proposes the modernization of a continuously variable transmission due to a constructive increase in the transmission ratio of the V-belt variator. As a result it increases to 3 and corresponds to the maximum value in the first gear of the manual transmission. This will allow to abandon the planetary and cylindrical gears as well as significantly reduce the overall dimensions of the variator. This will enable the placement in the engine compartment of the variator on cars of small and extra small class. The use of stepless transmission will reduce fuel consumption and wear of transmission mechanisms when driving in city conditions.

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Table 1: Evaluation of the technical level of analogues

Designation	Maximum moment (N/m)	Transmission range relationship	Advantages	Disadvantages
Autotronic 722.8	152	0.75-2.2	Ease and comfort of control in all driving modes. Effective engine operation mode, hence, the inability to overload the engine	The complexity of the mechanism of its device and relative fragility. From this follows the high price of the service (including expensive transmission oil for the variator)
Jatco CVT diagram	167	0.8-2.3	Effective engine performance, change gear ratio depending on the load on the engine and transmission	Difficulties in towing the car, resulting from the design features of the variator
Subaru CVT lineartronic	141	0.75-2.2	More rapid change in gear ratio due to a constructive change in the pulleys and their faster movement along the drive shaft	Slow gear shift; the complexity of off-road operation; road roughness sensitivity

MATERIALS AND METHODS

Based on the fact that the V-belt transmission is installed between the engine and the main transmission, the torque and speed on the drive pulley is chosen equal to the frequency and torque on the engine crankshaft, taking into account the gear ratio of the cylindrical gear transmission (Lykin *et al*., 1984).

The LADA Granta car was chosen as a prototype. For further calculations, you will need the technical characteristics shown in Table 2.

Figure 1 shows the assembly drawing of the variator with the indication of the shaft location numbers. When moving forward, the rotation is transmitted through a gear, synchronizer clutch, variator, main gear. Shaft 4 is used for reverse rotation when reverse gear is engaged.

Kinematic calculation of the proposed variator: The developed design of the variator is able to realize the minimum gear ratio u = 0.33. But by analogy with a manual gearbox to move at maximum speed it is enough to accept the gear ratio u = 0.85. At this value, the vehicle acceleration approaches zero. The control range of the variator D can be determined by Eq. 1:

$$D = \frac{u_{max}}{u_{min}} \quad (1)$$

where, u_{max}, u_{min} – maximum and minimum variator ratio. If u_{max} = 3 and u_{min} = 0,85 then D = 3.53. Determine the overall efficiency of the drive η_{tr} (10, c. 6) Eq. 2:

$$\eta_{tr} = \eta_{ab} \cdot \eta_{cs} \cdot \eta_{g} \cdot \eta_{pb} \cdot \eta_{b} \cdot \eta_{pb} \cdot \eta_{g} \quad (2)$$

According to reference data Grishkevich *et al*. (1985), Ivanov *et al*. (1970), Kosenkov (2003), Lykin *et al*. (1984) and Pronin and Revkov (1980) takes the following values of drive efficiency:

Table 2: Technical characteristics of the car "LADA Granta"

Characteristics	Values
Maximum torque	120 (N.m)
Crankshaft frequency at maximum torque	3800 (min^{-1})
Maximum engine power	64 (kW)
Crankshaft speed at maximum power	5100 (min^{-1})
Drag coefficient	0.356
Frontal area	1.82 (m^2)
Maximum vehicle speed	160 (km/h)
Tire size	155/65 R13

η_{pb} = Efficiency of a pair of bearings (η_{pb} = 0.99)
η_{cs} = Synchronizer coupling efficiency (η_{cs} = 0.98)
η_{g} = Gear efficiency (η_{g} = 0.97)
η_{b} = Belt drive efficiency (η_{b} = 0.95)

As a result of calculations, the overall efficiency of the drive is determined η_{tr} = 0.841. The main characteristics of the kinematic calculation of the proposed continuously variable transmission are shaft speeds, shaft power and torques on shafts. Shaft rotational speeds n_i are determined by Eq. 3:

$$n_i = \frac{n_1}{u} \quad (3)$$

Where:
n_1 = Input shaft rotational speed (min^{-1})
u = Mechanical gear ratio
i = Number of gear (shaft) from Fig. 1

The power on the shafts P_i is determined by Eq. 4:

$$P_i = \frac{P_1}{\eta_i} \quad (4)$$

Where:
P_1 = Input shaft Power (kW)
η_i = Transmission efficiency

Torques on the shafts T_i are determined by Eq. 5 (Pronin and Revkov, 1980):

$$T_i = 9550 \cdot \frac{P_i}{n_i} \quad (5)$$

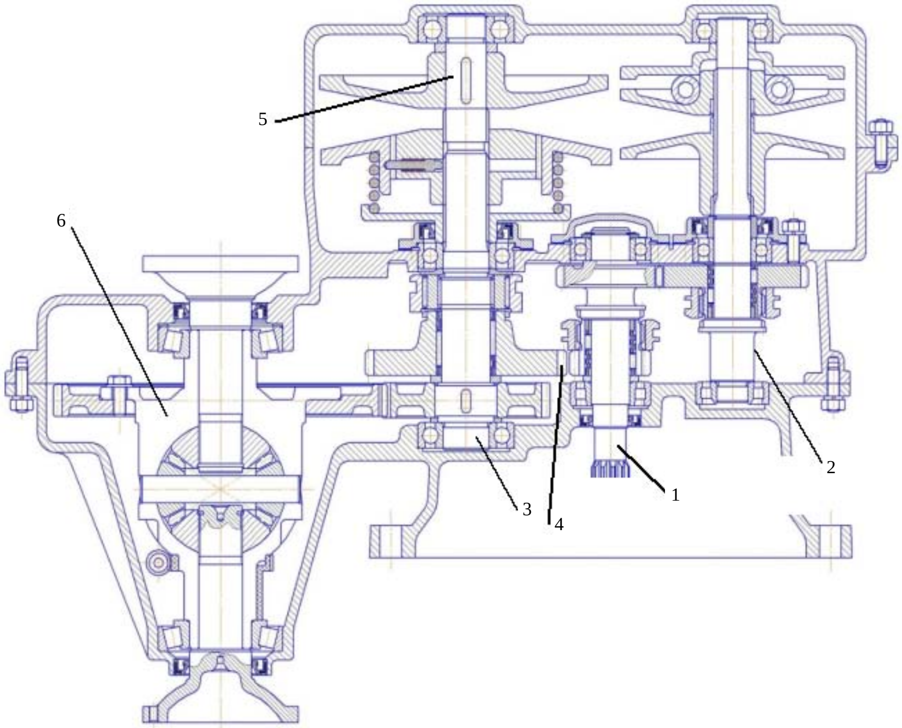

Fig. 1: The layout of the shafts in the gearbox: 1) Primary shaft; 2) the drive shaft of the variator; 3, 5) Variator driven shaft; 4) Shaft used for reverse rotation and 6) Differential housing

Table 3: The calculated values of the kinematic characteristics

Shaft number	Shaft rotation frequency n_i (min^{-1})	Shaft Power P_i, (kW)	Torque shaft T_i (N.m)
1	3800	40.49	101.8
2	3040	38.49	120.9
3	1013.3	36.20	341.2
5	1013.3	36.20	341.2
6	361.9	34.76	917.3

where, P_i is the shaft power. The calculated values of the kinematic characteristics are presented in Table 3. For structural and technological reasons in gearboxes, stepped shafts with different diameters of individual stages are used.

Traction dynamic calculation of transmission: The required maximum engine power P_{me} is calculated from the condition for maximum speed, i.e., power balance Eq. 6:

$$P_{me} = \frac{\left(F_\psi + F_W\right) \cdot V_{a_{max}}}{\eta_{tr} \cdot k_c} \qquad (6)$$

Where:

F_ψ = Road resistance Force (N)
F_w = Air resistance Force (N)
$V_{a_{max}}$ = Maximum Vehicle speed ($V_{a_{max}}$ = 160 km/h)
η_{tr} = Transmission efficiency in driving mode with maximum speed, for non-AWD vehicles with mechanical transmission (η_{tr} = 0.92)
k_c = Coefficient taking into account the loss of power in the air cleaner, muffler, radiator, compressor and auxiliary units (k_c = 0.9)

Air resistance force F_w Eq. 7:

$$F_W = C_X \cdot A \cdot q_v \qquad (7)$$

Where:

C_x = Vehicle drag Coefficient (C_x = 0.356)
A = Frontal Area (A = 1.82 m^2)

Calculate the velocity head q_v Eq. 8:

$$q_v = \frac{\rho_a \cdot V_{a_{max}}^2}{2} \qquad (8)$$

where, ρ_a = air density (ρ_a = 1.25 kg/m^3). As a result of calculations, the following values are obtained; q_v = 1231.1 kg/m·sec^2, F_W = 797.65 N. Road resistance power F_ψ Eq. 9:

$$F_{\psi} = G_a \cdot \psi_V \tag{9}$$

Where :

ψ_v = Road drag coefficient

G_a = Car weight ($G_a = 1.21 \cdot 10^4$ N)

The road resistance coefficient ψ_v is determined when the vehicle is moving at maximum speed Eq. 10:

$$\psi_V = \left(f_0 + 5{,}6.10^{-16} \cdot V_{a_{max}}^2\right) \tag{10}$$

where, f_0 coefficient of resistance to movement ($f_0 = 0.015$). As a result of calculations, the following values are obtained, $\psi_v = 0.026$, $F_{\psi} = 314.6$ N, $P_{me} = 59.62$ kW.

External speed characteristic is a graphical dependence which characterizes the dependence of the effective engine power and torque on the crankshaft rotation speed. Before the dynamic calculation, this characteristic needs to be built as it shows the transmitted power and torque transmitted to the wheels through the transmission minus losses (Kosenkov, 2003). The power values at different crankshaft rotational speed can be calculated by Eq. 11:

$$P_{e_1} = (n_e) = \left(P_{me} \cdot \left(C_1 \cdot \left(\frac{n_e}{n_{ep_{me}}}\right) + C_2 \cdot \left(\frac{n_e}{n_{ep_{me}}}\right)^2 - C_3 \cdot \left(\frac{n_e}{n_{ep_{me}}}\right)^3\right)\right) \tag{11}$$

Where:

$C_1 = 1$, = Coefficients for a gasoline engine
$C_2 = 1$, depending on the shape of the combustion
$C_3 = 1$ = chamber

P_{me} = Effective engine Power (kW)

n_e = Effective engine speed (min^{-1})

$n_{ep_{me}}$ = Crankshaft speed corresponding to maximum power (min^{-1})

Calculate the torque on the crankshaft $T_e(n_e)$ by Eq. 12:

$$T_e(n_e) = 9554 \cdot \frac{P_{e_1}(n_e)}{n_e} \tag{12}$$

According to the results of the calculations, we construct a graph of the external velocity characteristic (Fig. 2).

Traction characteristic is a graph of the change in thrust at the wheels of the vehicle when moving in each gear. According to this graph, you can determine the maximum speed of the vehicle, it will correspond to the point of intersection of the curve in high gear with the graph of the actual resistance forces (Ivanov *et al*., 1970; Lykin *et al*., 1984).

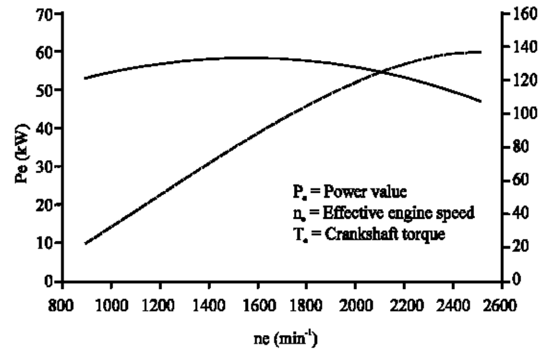

Fig. 2: External speed response

For a continuously variable transmission, the pull curve will be in the area bounded by two curves. When driving off-road-to the left on a flat surface-to the right.

When calculating the tires of a car (Ivanov *et al*., 1970) on the prototype 155/65/13 it is necessary to determine the free radius of the wheel r_w Eq. 13 and the rolling radius of the wheel r_r Eq. 14. The free radius of the wheel r_w can be defined as:

$$r_W = \frac{d_W \cdot 0.0254 + 2 \cdot H \cdot 10^{-3}}{2} \tag{13}$$

Where:

d_w = Wheel disc diameter (inch)

H = Tire profile Height (mm)

The rolling radius of the wheel r_r can be defined as Eq. 14:

$$r_r = 0.85 \cdot r_w \tag{14}$$

The following values of these values were obtained: r_w = 0.274 inches, r_r = 0.225 m. The calculation of the minimum transmission ratio $u_{tr_{min}}$ is carried out from the condition of the possibility of movement with a maximum speed according to Eq. 15:

$$u_{tr_{min}} = \frac{0.105 \cdot n_{eP_{me}} \cdot r_K}{V_{a_{max}}} \tag{15}$$

where, $n_{eP_{me}}$ is the rotational speed of the crankshaft at maximum power. If $n_{eP_{me}} = 5300\ min^{-1}$, then $u_{tr_{min}} = 2.82$. Vehicle speed V_{a_i} is calculated with the maximum and minimum gear ratio of the variator (Pronin and Revkov, 1980) Eq. 16:

$$V_{a_i} = 0.105 \cdot n_e \cdot \frac{r_K}{u_i \cdot u_{mt}} \tag{16}$$

Where:

u_i = Gear ratio on the calculated gear

u_{mt} = Gear ratio main gear

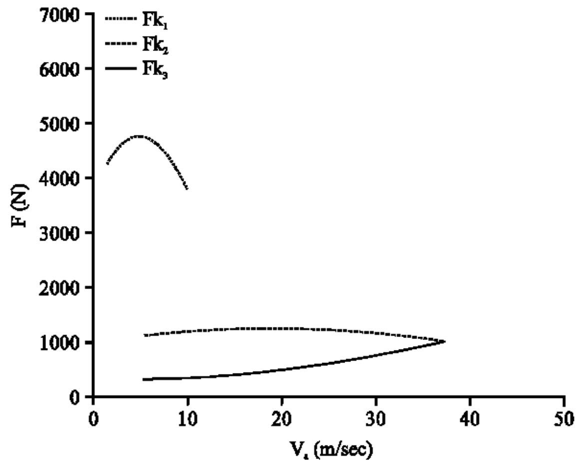

Fig. 3: Vehicle traction. F_{k1}, F_{k2}-traction force on wheels at maximum and minimum gear ratio. F_{12}-road resistance force; Traction force on wheels

The thrust force on wheels F_{ki} is calculated when driving with maximum and minimum speed using Eq. 17:

$$F_{ki} = \frac{T_{e_i} \cdot u_i \cdot u_{mt} \cdot \eta_{tr} \cdot k_c}{r_r} \quad (17)$$

Forces of air F_w resistance can be determined by Eq. 18:

$$F_w = C_x \cdot A \cdot \frac{\rho_a \cdot V_{a_{max}}^{\ 2}}{2} \quad (18)$$

Based on the calculated values, we build a graph of the traction characteristic of the car (Fig. 3). Dynamic characteristic is a graphical dependence of the dynamic factor of the car on the speed at each gear. The dynamic factor characterizes the ability of the car to reach maximum speed and overcome the resistance of rolling and lifting. It is equal to the ratio of the free thrust force to the weight of the car (Ivanov *et al.*, 1970). The calculation of the dynamic factor Di according to the formula was carried out according to Eq. 19:

$$D_i = \frac{F_{k_i} - F_{w_i}}{G_a} \quad (19)$$

Based on the calculated values, we construct a graph of the dynamic performance of the car (Fig. 4). The acceleration characteristic is a graphic dependence of the vehicle acceleration on the speed of movement in each gear. It characterizes the ability of the car to reach the

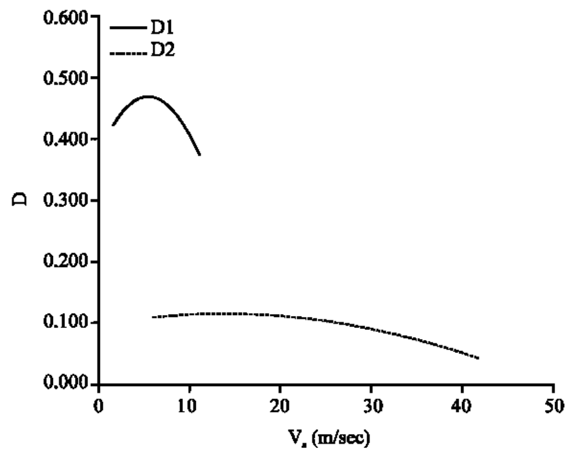

Fig. 4: Graph of dynamic characteristics. D1-dynamic performance at maximum gear ratio. D2-at maximum gear ratio; Dynamic factor

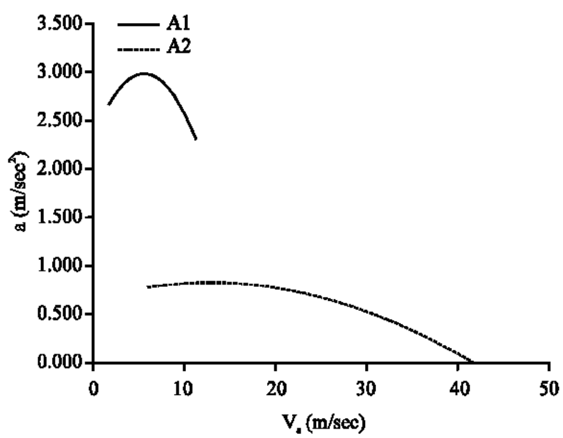

Fig. 5: Graph of acceleration characteristics. A1-acceleration at maximum gear ratio. A2-acceleration with a minimum gear ratio; Car acceleration time

maximum speed at which acceleration approaches zero (Kosenkov, 2003). The acceleration of the car $a_i(n_e)$ was calculated by Eq. 20:

$$a_i(n_e) = \frac{\left(D_i(n_e) - \psi_i(n_e)\right)}{\delta_i} \cdot g \quad (20)$$

where, g = Acceleration of gravity (g = 9.8 m/sec^2). The coefficient of rotating masses δ_i is defined as:

$$\delta_i = 1.04 + 0.05 \cdot u_{tr}^{\ 2}$$

Based on the calculated values, we build a graph of the vehicle's acceleration characteristics (Fig. 5).

RESUTLS AND DISCUSSION

Based on the chosen scheme and the calculations performed in the CAD SOLIDWORKS has developed a continuously variable transmission for a small car (Fig. 6). A feature of this design is the use of a V-belt transmission with an increased gear ratio which greatly increases the traction and dynamic characteristics of a passenger car compared to mass-produced automatic transmissions.

In this design, the gearbox housing is made in one piece along with the clutch housing. The gearbox is connected to the engine by means of a bolt joint for this purpose, four holes are made on the clutch housing, concentric with the holes on the engine.

The continuously variable transmission under development combines a V-belt transmission with an increased range of gear ratio, reverse mechanism, final drive, differential and protective flywheel housing in one case.

V-belt drive is installed in a separate case, behind the gearbox. This constructive solution is made for more convenient replacement of the variator belt and to increase the resource of the V-belt transmission. The bodies of the reverse mechanism and the main transmission are separated, so that, the oil from the lubrication system does not fall into the variator. This is ensured by lip seals (Lykin *et al*., 1984). In the upper part of the case of the variator, the tide is made to install the gearshift mechanism.

Figure 7 shows the design of a continuously variable transmission, consisting of a body cover, a cylindrical main gear and a reverse mechanism. The gearbox is fixed to the car body with the help of brackets. All actuators are located in the housing. The V-belt drive is made in the rear part of the case for more accessible replacement of the variator belt, a cover is provided.

The inclusion of reverse gear by using gear couplings. This design solution allows to increase the resource of the V-belt transmission.

When moving forward, the rotation is transmitted directly through the variator. When the reverse gear is engaged, the torque is transmitted to the main gear through a cylindrical gear train and the V-belt drive is completely stationary (Lykin *et al*., 1984). The design of the rear cover of the variator makes it easy to open access to the variator belt.

The rear cover of the variator allows you to access the main components of the gearbox. In order to carry out repairs or periodic maintenance of continuously variable

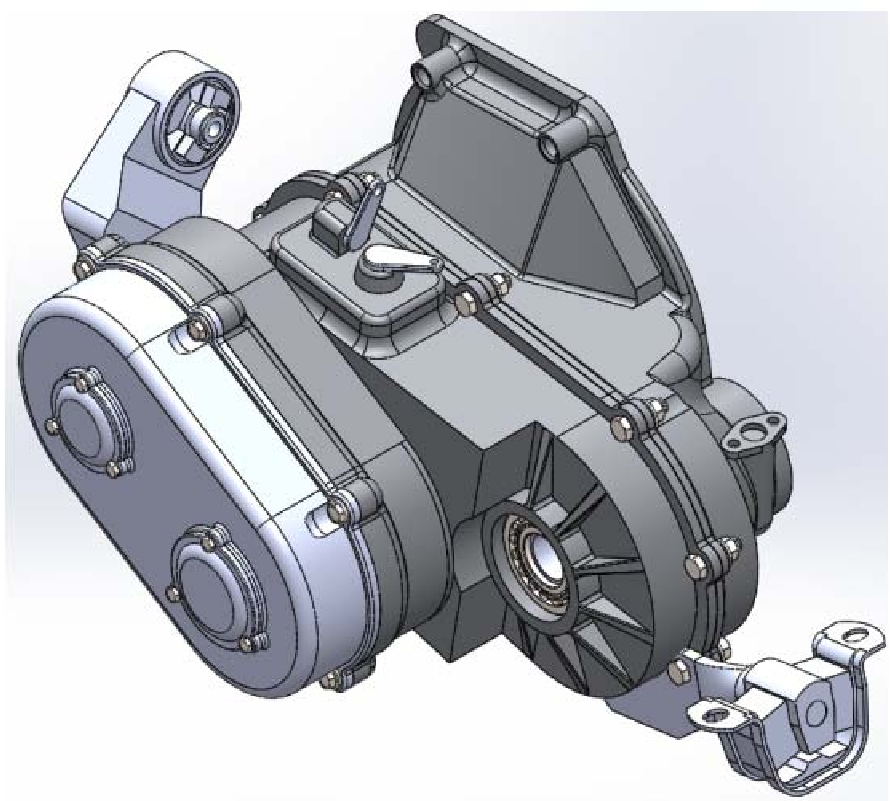

Fig. 6: The continuously variable transmission (isometric)

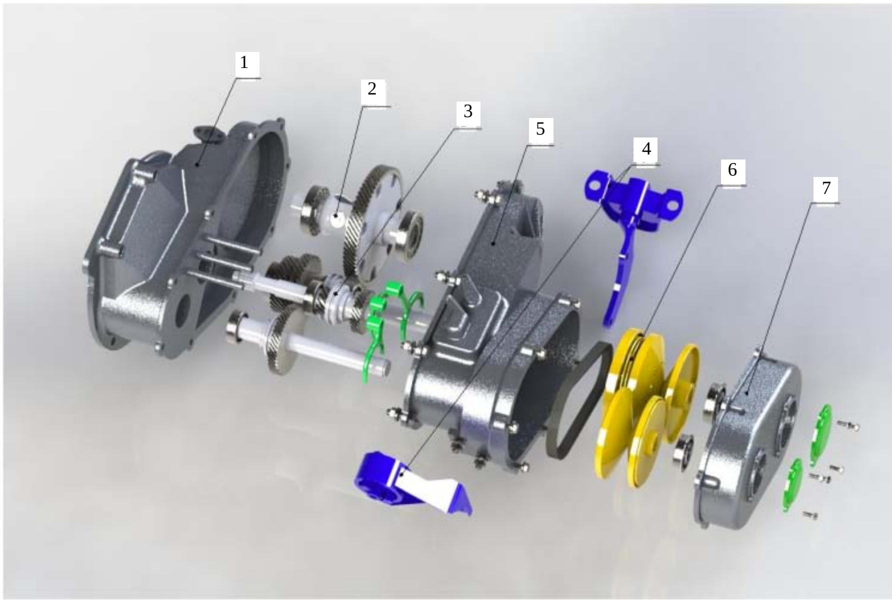

Fig. 7: Transmission design: 1) Gearbox cover; 2) Cylindrical main gear; 3) Reverse rotation mechanism; 4) Mounting brackets to the car body; 5) Variator housing; 6) V-belt drive and 7) V-belt gear cover

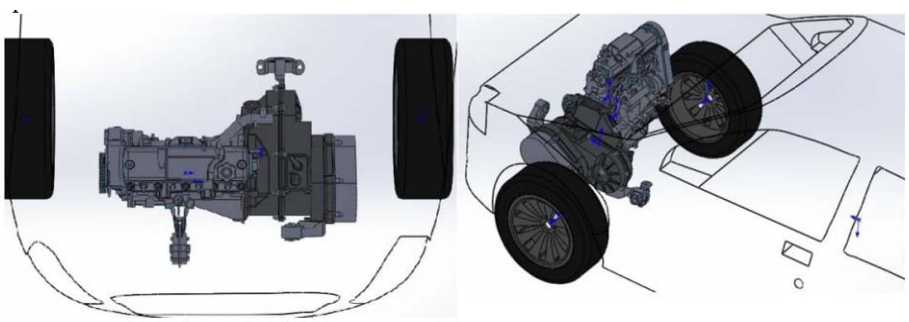

Fig. 8: Location of the power unit on the car

transmissions it suffices to remove the rear cover of the variator. This opens access to the main nodes it becomes possible to replace the variator belt or pulleys.

The power unit is located transversely in the engine compartment of the car. Its attachment to the body is accomplished with the help of three brackets that combine rubber pads to damp vibrations. Two brackets are attached to the gearbox, one to the engine (Grishkevich *et al*., 1985).

To transfer torque to the wheels of the car used hinges equal angular velocity. They connect the axle gears of the differential and the hub of the drive wheels. They have advantages in comparison with cardan transmission such as large angles of rotation and high reliability.

In the design of the continuously variable transmission used V-belt variator. The transmission of rotation to the wheels when the rear speed is engaged is carried out by means of a gear, this allows you to increase the resource of the variator belt. The main gear is located in the same housing with other actuators. This solution allows to reduce the overall dimensions of the transmission and reduce the unsprung weight of the car (Fig. 8). Fastening mechanisms by using special brackets.

When you remove the cover, you get access to the main gearbox mechanisms. The main gearbox and differential support are roller bearings which in turn are located in the housing and gearbox cover.

The drive shaft is mounted in bearings which are located on the crankshaft of the engine and in the variator

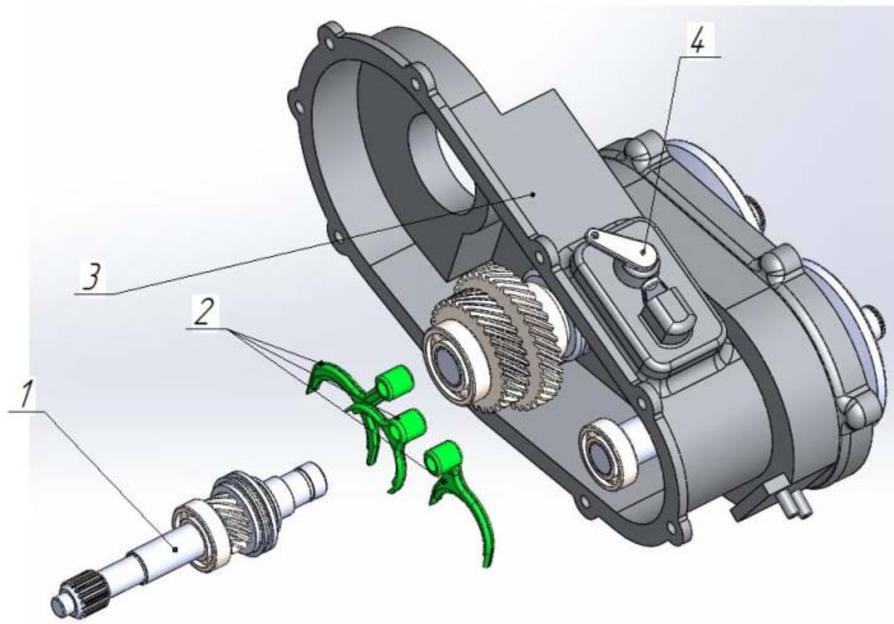

Fig. 9: Reverse gear engagement mechanism: 1) Drive shaft; 2) Reverse gear forks; 3) Variator housing and 4) Switching mechanism

housing. Forks serving to activate the gear couplings when the reverse gear is engaged, move along the guides. They are controlled by a lever from the passenger compartment which is connected by rods with the switching mechanism (Fig. 9). Gear couplings allow you to completely eliminate from the work of the variator when the reverse gear. This will increase the life of the belt (Ivanov *et al*., 1970).

In the developed design of the gearbox incorporation rear speed there are three gear couplings. The first is responsible for turning off the variator from work and switching on the neutral gear it is installed on the drive shaft. The second is located on the primary shaft and activates the transfer of reverse rotation to the main gear (with the first clutch off). The third gear coupling serves to completely exclude from the work of the V-belt transmission it is located on the driven shaft of the variator (Kosenkov, 2003).

There are advantages over analogues in the choose scheme of rotation transmission when the rear gear is turned on.

First, planetary gear is excluded from the design. In prototypes it is installed on the primary shaft and changes the frequency of rotation of the variator. In this case, the transmission of torque through the V-belt transmission. The belt is forced to work under alternating load conditions and quickly fails as a result of fatigue failure (Kosenkov, 2003). Secondly, the planetary gearing reduces the overall efficiency of the drive due to the large number of gear pairs.

The drive and driven shafts of the variator are mounted in a housing on bearings. Gears are connected by means of gear couplings with shafts when the reverse gear is engaged.

The stationary half of the driven pulley is mounted on the shaft and fixed with the help of a parallel key. The movable half moves along the axis of the shaft using rectangular slots made on the inner surface of the pulley and the outer surface of the shaft. The belt is held in constant tension by means of a coil spring (Fig. 10).

At the moment when the hub moving along the shaft becomes opposite to the groove, the pins under the action of the springs descend. In this case, the outer section will be released and will continue to move in the direction of the hub under the action of the spring. Thereby, pushing the belt to a larger diameter and increasing the gear ratio of the variator.

A groove is made on the driven shaft to block the outer section. The locking pins are installed in the hub bores. Springs are installed on them to ensure automatic unlocking of the outer section (Fig. 11).

Groove on the shaft is performed at a certain distance from the end of the shaft. This distance is chosen, so

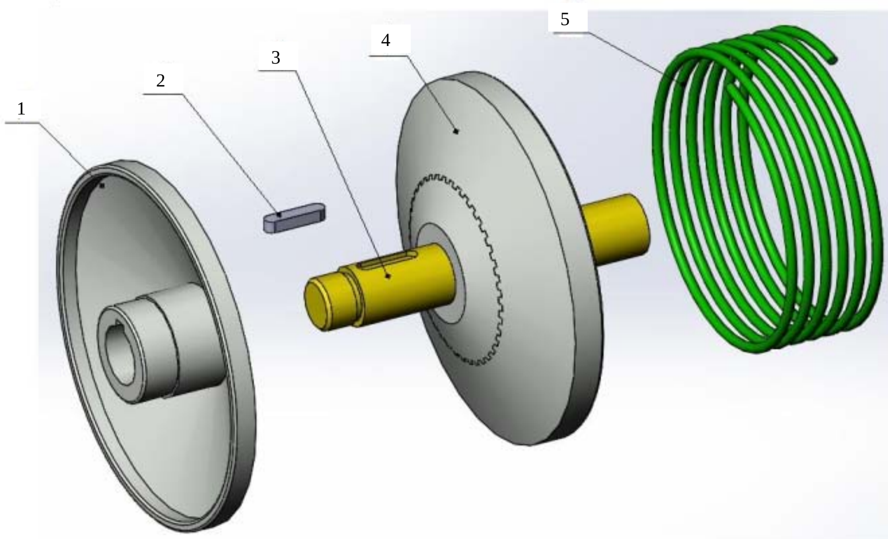

Fig. 10: The driven pulley of the variator: 1) Fixed half; 2) Key; 3) Driven shaft; 4) Movable half and 5) Cylindrical spring

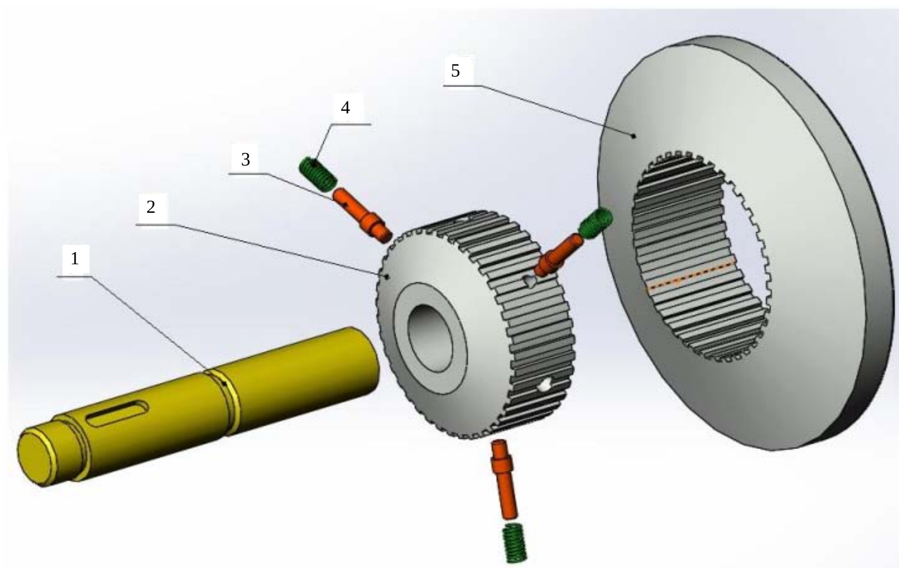

Fig. 11: Interlock: 1) Groove on the shaft; 2) Hub; 3) Locking pin; 4) Cylindrical spring and 5) Outer section

that, the outer section is locked and unlocked while the belt is located on the outer section. This condition is required in order to reduce belt wear on sharp ends of the spline joint.

CONCLUSION

The design of the continuously variable transmission which has smaller weight and size characteristics compared with analogues. Selected the location of this transmission in the engine compartment of the car.

The device of this continuously variable transmission has an increased range of gear ratios as compared with analogues.

The necessary kinematic design calculation was performed which allows determining the kinematic parameters of the main working units of the gearbox.

The traction and dynamic calculation of the transmission is carried out which allows to determine the acceleration and traction and dynamic characteristics. Evaluation of the results shows the design performance and compliance with the specified technical requirements.

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