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MODEL OF A V-BELT CENTRIFUGAL VARIATOR WITH MECHANICAL ADAPTATION

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Introduction. Modern automatic V-belt CVTs (Continuously Variable Transmission) have in-built regulators as a part of their construction. These are centrifugal regulators which change the gear ratio [1, 2]. But the kinematic scheme of a motor vehicle CVT is based, in most cases, on an irrational converting of the centrifugal force. Therefore, one of the tasks of solving this problem is to create a variator with new properties.

Problem statement. Improving the performance of motor vehicles involves its automation, which ensures motion safety, ease of control and the ability to operate the engine in the optimal mode. In recent years, the tendency of the use of automated V-belt variators in motor vehicle CVTs has become stronger. The design of such a variator requires exact calculations, taking into account all the determining factors and design parameters of the CVT. The problem of selecting the most appropriate configuration of regulating devices and automatic clutch, providing enhanced dynamics of the acceleration of motor vehicles [2].

The article presents a model of a V-belt centrifugal variator for Honda Tact AF-24 scooter with the V-belt variator input link substituted for a new centrifugal link, designed by the authors. The output unit, together with the centrifugal clutch, remained unaltered.

Object of study. The constructional basis for building a prototype is the physical model, described in publications [3, 4]. Prototype tests were conducted in the modes of starting off and that of linear motion [5]. The tests have proven the performance of the new mechanical transmission of the CVT, based on the centrifugal variator (physical model) (fig. 1). Following the scheme of distribution of forces (fig. 2), we are going to determine a new gear ratio of the altered V-belt CVT.

The new V-belt CVT model. Under the scheme of distribution of working forces and control forces, they are divided into the centrifugal force of the sliding pulley, the centrifugal force of the flexible element (belt), belt tension, the jamming force between the guide rail and the sliding pulley, the load on the output shaft and the weight of the driven pulley. Control forces include: the centrifugal force of epy flexible element (belt), the load on the output shaft.

We will give here the equations describing each force:

The centrifugal force of one sliding pulley will constitute:

$$F_{CFS} = m \cdot \omega^2 \cdot R, \qquad (1)$$

where m is the pulley weight, g; ω will be angular velocity of the input shaft in Hz; R is the radius of the sliding pulley’s position in mm.

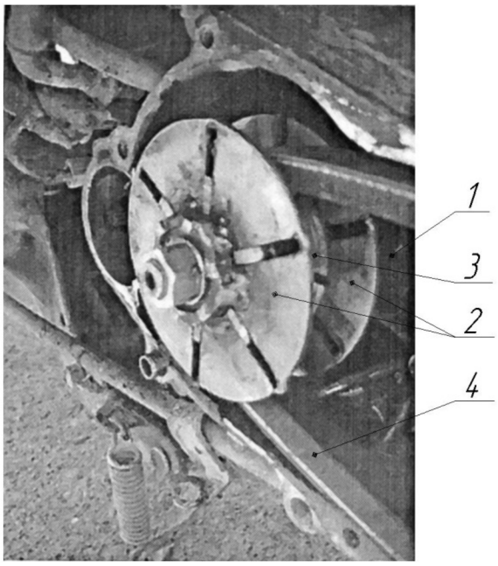

Figure 1. General view of the centrifugal element of the CVT scooter Honda Tact AF-24.
1 – the engine; 2 – guide disks of the sliding parts of the pulley; 3 – segments of the sliding pulley; 4 – the belt.

Centrifugal belt force:

$$F_{CFB} = m_1 \cdot \omega^2 \cdot R_1 \cdot \sin\beta, \qquad (2)$$

where m_1 is the mass of the part of the belt limited by angle β, g; ω is angular velocity of the input shaft, Hz; R_1 is belt radius, mm; β is the belt's looping angle of the pulley, deg.

Centripetal force of one sliding pulley:

$$F_{CPF} = \frac{m \cdot v^2}{R}, \qquad (3)$$

where m is the pulley weight, g; v is the sliding pulley's linear speed, mm/min.; R is the position radius of the sliding pulley, mm.

Belt tension force:

$$F_{BTF} = 500 \cdot \frac{(2{,}5 - C_\alpha) \cdot P_{DP} \cdot C_p}{C_\alpha \cdot v} + m_a \cdot v_1^2, \qquad (4)$$

where m_a is the belt's length weight, g; C_α is the looping angle coefficient; C_ρ is the dynamic load and single-shift operating mode coefficient; P_{DP} is the rated drive power, N; v_1 is the belt's linear speed, mm/min.

The wedging force of the single pulley's centrifugal link will be:

$$F_{WF} = n \cdot F_N \cdot \frac{1}{3} \cdot \kappa, \qquad (5)$$

where n is summary points of contact between the guide rail and the sliding pulley; F_N is the normal force acting on the pulley, N; κ is the gearing coefficient.

Output shaft load. This parameter is determined as the moment of resistance on the output shaft:

$$P = F_F \cdot R_{OS}, \qquad (6)$$

where R_{OS} is the output pulley's radius, mm; F_F is the friction force between the shaft and the load element:

$$F_F = F_N \cdot f\ , \tag{7}$$

where F_N is the normal force acting on the pulley, N; *f* is the coefficient of friction (here accepted as f = 0,6). By substituting the values in Formula 6, we will get the following equation:

$$P = F_N \cdot f \cdot R_{OS}\ , \tag{8}$$

Taking into consideration the direction of the force vectors (fig. 2) and having transformed the indicated equations (1-8), we obtain the equation of the centrifugal force of one sliding pulley:

$$F_{CFS} = \frac{\omega_{out} \cdot m \cdot \omega \cdot R_{OS}}{R}\ , \tag{9}$$

The total gear ratio of the variator is defined as the difference between the input power and the output power. The input power is constituted by the motor shaft torque multiplied by the shaft rotation frequency. The power output is constituted by the drive wheel torque of a motor vehicle multiplied by its rotational speed. Considering that the rotating moments are circular forces multiplied by the frequencies, we obtain the equation for determining the total gear ratio:

$$i_T = \frac{z \cdot m \cdot \omega}{P}\ , \tag{10}$$

in which Z is the number of the sliding parts of the centrifugal variator link, in pieces.

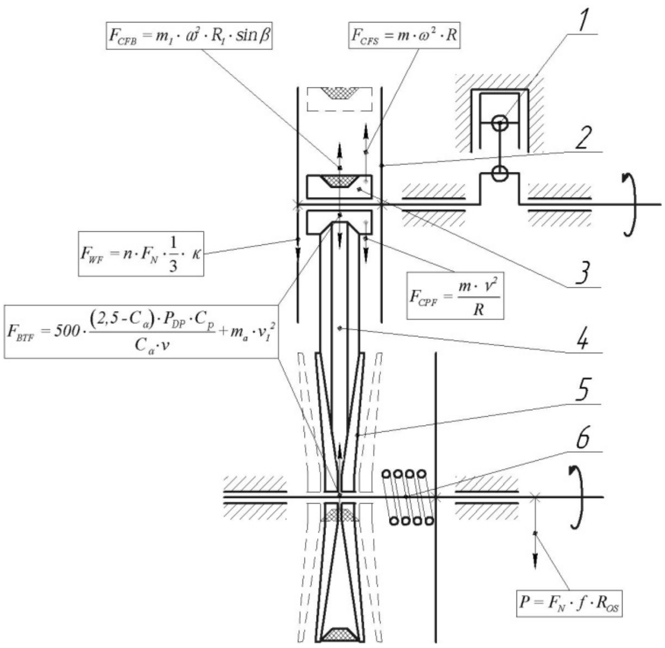

Figure 2. Scheme of the distribution of working and guiding forces proposed by the authors of the CVT with the centrifugal link of the Honda Tact AF-24 scooter:
1 – the engine; 2 – guide disks of the sliding parts of the pulley; 3 – segments of the sliding pulley; 4 – belt; 5 – sliding disks of the variator's output link; 6 – centrifugal clutch variator.

Conclusion. The authors offer a mathematical model of the V-belt variator, in which, due to the kinematics of the new variator, the input link serves as a centrifugal regulator of the rotation frequency of the engine shaft. The proposed equation of the gear ratio (10) is based on the total weight of the sliding parts of the centrifugal element, the frequency of rotation of the input shaft and the load of the output shaft. These properties characterize the mechanical adaptation of variators of this class.

References

1. Spiridonov O. V. Methods of increasing the efficiency of the operation of automatic V-belt variators in the transmission of motorized vehicles. / Abstract. dis kand. tech. Sciences - Specialty 05.02.02-05 - Kovrov, 2008. 26 pp.
2. Mikheev S. S. Design and optimization of parameters of an automatic V-belt variator of motor vehicles. / Abstract. dis kand. tech. Sciences - Specialty 05.02.02 - Vladimir, 1998. 24 pp.
3. Rational distribution of controlling forces of variators with flexible joints / Pilipenko, O. M., Litovchenko, V. V., Udodenko, V. S., Virevka D. I. - Newsletter of SevNTU, 2011 (№122), 118-120.
4. Investigation of step-by-step mechanical transmission on adaptation / Litovchenko, V. V., Pidhornyi M. V.// Materials of the VIIth International Scientific and Practical Internet Conference "Problems and Prospects for the Development of Automobile Transport" - Vinnytsia, 2019, 96-100.
5. Directive 2006/42/EC of the european parliament and of the council of 17 may 2006 on machinery.

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ОСОБЕННОСТИ ПРИМЕНЕНИЯ МЕТОДА ГРАФОВ ДЛЯ АНАЛИЗА ТОЧНОСТИ УГЛОВЫХ РАЗМЕРОВ ДЕТАЛЕЙ ПРИ МЕХОБРАБОТКЕ

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Методика анализа точности мехобработки путем выявления графов взаимосвязи линейных размеров детали и заготовки подобно рассмотрена в ряде источников [1, 2 и др.]. При этом не рассматриваются особенности угловых размеров, хотя, обычно, уровень их точности более высок. Поэтому, усовершенствование методики анализа техпроцессов мехобработки методом графов, с учетом специфики угловых размеров, является актуальной задачей.

В ходе теоретических исследований формирования угловых размеров деталей выявлен ряд особенностей построения графов и уравнений угловых размерных связей техпроцессов, а также особенности решения таких уравнений: 1) графы и уравнения угловых размеров следует составлять и решать раньше, чем графы и уравнения линейных размеров. Это позволит учесть угловые смещения элементов полуфабрикатов в составе линейных промежуточных припусков; 2) в общем случае достаточно составить исходный и технологический граф для трех координатных плоскостей детали. Одну из них следует совмещать с наиболее часто применяемой в анализируемом техпроцессе установочной, либо направ-