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Synthesis of the Energy-Saving Dry Dual Clutch Control Mechanism

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Abstract: A promising direction in developing up-to-date transport vehicles is the use of transmissions, an essential element of which is a dual-clutch. Improving functional performance, energy efficiency, and environmental friendliness are relevant and require appropriate research. The object of study is a dry dual-clutch working with a manual transmission with high energy efficiency. In the proposed design scheme, a rotary lever and a movable carriage can change the structural diagram of the force interaction between the pressure spring and the clutch discs. The developed mathematical model of the clutches control mechanism allowed for analyzing the process of controlling the dual-clutch transmissions (DCT) and obtaining functional dependencies between pushing forces in friction pairs and the carriage position. As a result, a method for synthesizing DCT was proposed to ensure the transmission of a given torque. Furthermore, it was determined and numerically proven that the same radial movement of the carriage when switching-on the clutches does not guarantee equal loading for the frictional discs of each clutch. This fact increases the uneven dynamics of torque and disc wear. Overall, the synthesis of the energy-saving dry dual-clutch control mechanism providing the same clutch margin for each clutch was developed. The method can be generalized and applied to designing dry dual-clutch transmissions for machines of various purposes.

Keywords: robotic transmission; transport vehicle; dual-clutch transmission; torque; clutch margin; process innovation; energy efficiency

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1. Introduction

One of the essential elements of the transmission is a clutch. With the development of transmission designs, it becomes necessary to develop new clutches and control systems. They should increase the vehicle and the transmission elements resource, positively influence dynamics and performance, and control and operation energy efficiency.

DCT has advantages in manufacturing, operating, and maintenance costs. They work without interruption in power flow or can switch clutches rapidly. Unlike traditional design schemes, existing automatic transmissions use closed clutches. They require permanent energy consumption when changing switches and maintaining the frictional pairs of one of the clutches in the closed position.

In DCT, opening one clutch and closing the second one occurred in a reduced or overlapped time. Thus, gear shifting is ensured with no interruption in power flow. This significantly reduces gearshift time, improves acceleration dynamics, and reduces fuel consumption and emissions to the environment.

One of the ways to improve such clutches is to reduce the energy consumption for control while ensuring reliable transmission of the required torque and safety indicators. Studies of the energy-saving dry clutch control mechanism allow recommending rational choice and a unification of the designs while ensuring the proper functioning of the drive and control system.

Recently, significant progress has been reached in the development of transmissions. Particularly, the number of speeds has been increased, the range of gear ratios has been extended, and the gearshift rate has been raised [1].

Electric drives are permanently expanding in response to tightening emission control regulations and urgent fuel economy requirements in automotive transmissions. As a result, integrating electric drives [2] into conventional transmissions occurs. Such an approach simplifies modular designs and the production of transmissions. However, it does not solve the problems of developing and improving the mechanical parts of transmissions.

Improving transmissions involves researching dry clutches related to vehicle dynamics [3–9], reliability [10], clutch architecture [5], durability, noise, switching strategy, and control technology, as well as energy efficiency and comfort [11]. The importance of dry clutch control was previously discussed in refs. [12–14].

Based on the models of workflows in dry clutches, several control algorithms have been proposed: optimal control [15], branched control [16], adaptive control [17], control based on fuzzy logic [8], robust control [18], combined management strategy [19], and predictive control [5]. Notably, effective clutch control requires developing a mathematical model for torque transmission with a minimum deviation from the actual process [20–23]. In ref. [4], a detailed analysis of the dry clutch architecture was carried out, allowing one to understand the main phenomena affecting torque transmission by friction forces. However, the literature does not provide enough information on the justification and analysis of DCT's design schemes with reduced energy costs for control, functional features, and rational selection of types and parameters for drives and actuators.

The clutch design is mainly based on the types of transmission and gearbox [24]. Nowadays, a number of vehicles are equipped with preselective gearboxes. Based on the positive effects of their application, a rational choice of a gearbox is realized [25]. Moreover, preselective transmissions are structurally simpler, have higher energy efficiency, and do not need to be repaired by highly qualified specialists. Using a dry dual-clutch eliminates the need to use expensive components and materials from the hydraulic system. During the operation of DCT, the switched-off multi-plate wet clutch creates additional resistance due to the difference in the angular velocities of the driving and driven discs.

Ref. [26] is devoted to shifting gears with friction clutches in the transmissions of wheeled and tracked vehicles. It analyzes the gear-shifting process with and without interruption of the power flow. However, more attention should be paid to investigating the properties of dry DCTs with reduced energy consumption for control.

The dynamic behavior of automotive dry clutches depends on the friction characteristics of the contact between the material of the friction lining, the flywheel, and the pressure disc when the clutch is engaged [27,28]. When switched on, the friction causes contact heat due to high interphase sliding and relatively high contact pressures. This affects the behavior of the material and the related frictional characteristics [29].

One of the critical problems in designing DCT is clutch durability. When the temperature is exceeded in dry clutches, the friction lining begins to suffer irreversible damage. For a dry DCT, a thermal model should be developed to predict parameter changes. In this regard, it is essential to compare experimental and theoretical results in the contact pressure and wear distribution [30]. However, only a single-clutch transmission was considered.

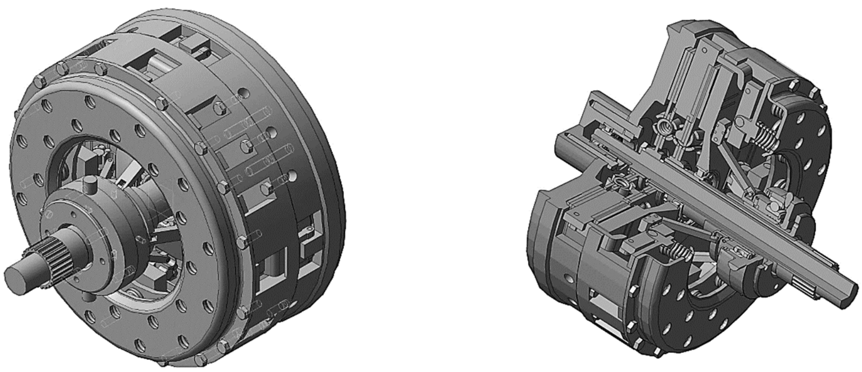
Overall, parameters and the design scheme of the mechanism that ensure the compression of the friction pairs and the control system predominantly affect the pressure force and the switching-on/off time for each clutch.

Due to the mentioned above, the research aims to analyze the operational process and evaluate the impact of the parameters of the dry DCT's control mechanism with high energy efficiency on changing the force. Simultaneously, the synthesizing parameters of the control mechanism should ensure uniform clutch margins for each clutch.

2. Materials and Methods

2.1. The Design Model

The energy efficiency of drive control for DCTs of different designs was analyzed. A comparison with the original DCT with reduced control costs was also carried out. As a result, it was determined that the proposed transmission [31] requires energy only when starting the vehicle and shifting gears (Figures 1 and 2).



(**a**) (**b**)

Figure 1. 3D model of the DCT: (**a**)—general view; (**b**)—cross-sectional view.

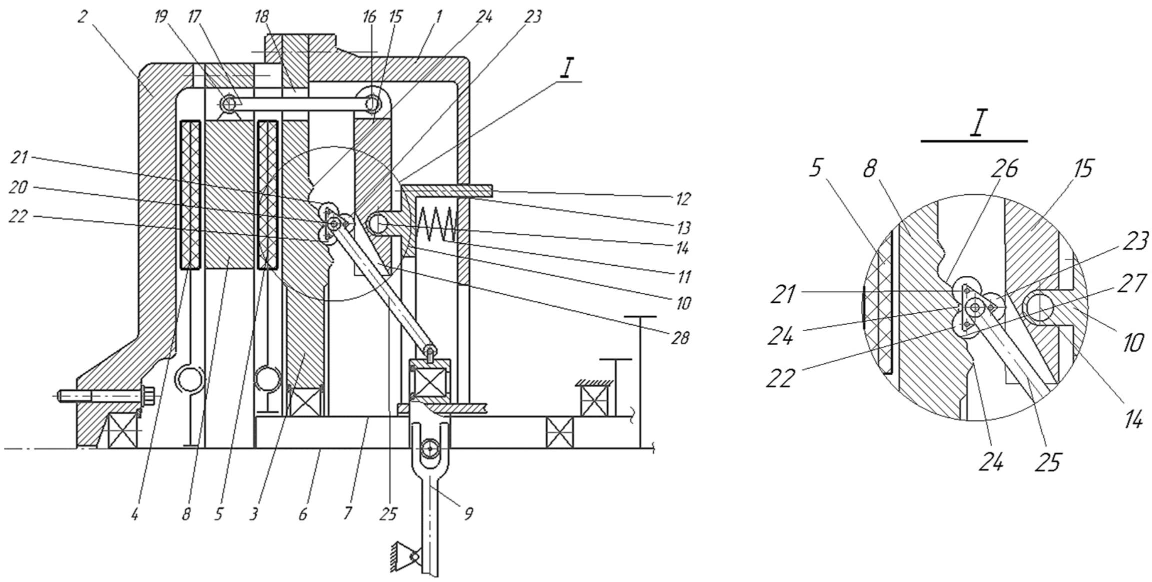


Figure 2. The original DCT design: 1—casing; 2—flywheel; 3—platter; 4,5—driven discs; 6, 7—shafts; 8—pressure plate; 9—drive mechanism; 10—additional disc; 11—pressure springs; 12—elements; 13, 18—holes; 14, 16, 19—hinges; 15—rotary stops; 17, 25—levers; 20—movable supports; 21–23—rollers; 24—clamps; 26, 27—surfaces; 28—inclined grooves.

DCT contains casing 1 (mounted on the engine's crankshaft through the leading flywheel 2), platter 3, driven discs 4 and 5 (mounted on splines on coaxially located main shafts 6 and 7), and shift drive mechanism 9. Flywheel 2 (used as drive disc), platter 3, and casing 1 are rigidly interconnected in the axial direction along the periphery, forming the driving elements of the clutches.

Between platter 3 and casing 1, additional disc 10 is installed in the axial direction. It is loaded with pressure springs 11 and moved through elements 12 with the possibility of interaction with holes 13. On an additional drive 10, from the platter side in the radial direction, stops 15 are fixed with hinges 14. They can rotate in radial planes. The ends of swivel stops 15 are connected with levers 17 from the periphery of the discs (in a horizontal plane) with hinges 16. The last ones (passed through holes 18 in platter 3 through hinges 19) are fixed on pressure plate 8. This plate has a possibility of forced axial movement.

The second free end of the rotary stops 15 interacts with movable supports 20. These supports are made as carriages with rollers 21, 22, and 23, placed between stops 15 and platter 3. The pair of rollers 21 and 22 on carriages can interact with clamps 24. Movable supports 20 are pivotally connected to drive mechanism 9 for switching clutches through levers 25.

The designed DCT operates as follows. In the switched-off (neutral) position, tdrive mechanism 9 is installed in such a way that the supports 20 are fixed in the middle position, excluding the rotation of stops 15 in the radial planes. Driven discs 4 and 5 do not interact with the leading flywheel 2, pressure plate 8, and platter 3. Main shafts 5 and 6 of even and odd gears rotate freely through driven discs 3 and 4 without transmitting torque from the engine.

When switching on the 1st clutch (Figures 2 and 3), drive mechanism 9 is installed in such a way that the supports 20 move from the periphery of the discs to the rotation axis of the shafts through levers 25. In this case, support 20, overcoming the resistance of the clamps 24, moves along surface 27. It is installed in a fixed position, where torque occurs between hinges and rollers 23.

Additional disc 10 moves in the axial direction towards support disc 8 under the action of springs 11. Stops 15 transmit the pushing force to levers 17, turning on hinges 14 through hinges 16. Levers 7 transmit the force to pressure plate 8 through hinges 19. Pressure plate 8 acts on the driven disc 4 and flywheel 2. The friction pairs circuit for the 1st clutch closes. As a result, torque from the engine is transmitted to the input shaft 6 of the gearbox through drive flywheel 2, pressure plate 8, and drive plate 4.

When the 1st clutch is switched off, drive mechanism 9 is set to a position in which stops 20 moves to the periphery of the discs (Figures 2b and 3b). In this case, rollers 21 and 22 move along surface 27, overcoming the resistance of clamps 24. These rollers are installed in a fixed position. However, rollers 23 are mounted on the same horizontal axis as hinges 14 of swivel stops 15 and pressure springs 11. Additional disc 10 moves in the axial direction away from support disc 3, overcoming the resistance of springs 11. Stops 15 transmit the pulling force to levers 17, turning on hinges 14 through hinges 16. Levers 17 transmit the force to pressure plate 8 through hinges 19. Pressure disc 8 opens the power circuit, stopping interaction with driven disc 3. As a result, the torque transmission from the engine to shaft 6 of the even rows of the gearbox stops.

When the 2nd clutch is switched on (Figures 2c and 3c), mechanism 9 is located in such a way that stops 20 moves to the periphery through levers 25. In the rotary stops 15, inclined grooves 28 are made to ensure free movement of the levers 25. As a result, the contact of rollers 23 is ensured along one initial plane of stops 15. In this case, support 20 moves along surface 26, overcoming the resistance of clamps 24. Torque occurs between the axes of hinges 14 and rollers 23. Additional disc 10 moves in the axial direction towards support disc 3 under the action of compression springs 11. Stops 15 transmit the pulling force to levers 17, turning on hinges 14 through hinges 16. Levers 17 transmit the force to pressure plate 8 through hinges 19. Pressure disc 8 closes the power circuit, acting on driven disc 5 and platter 3. As a result, torque from the engine is transmitted to shaft 7 through pressure plate 8, driven disc 5, and support disc 3.

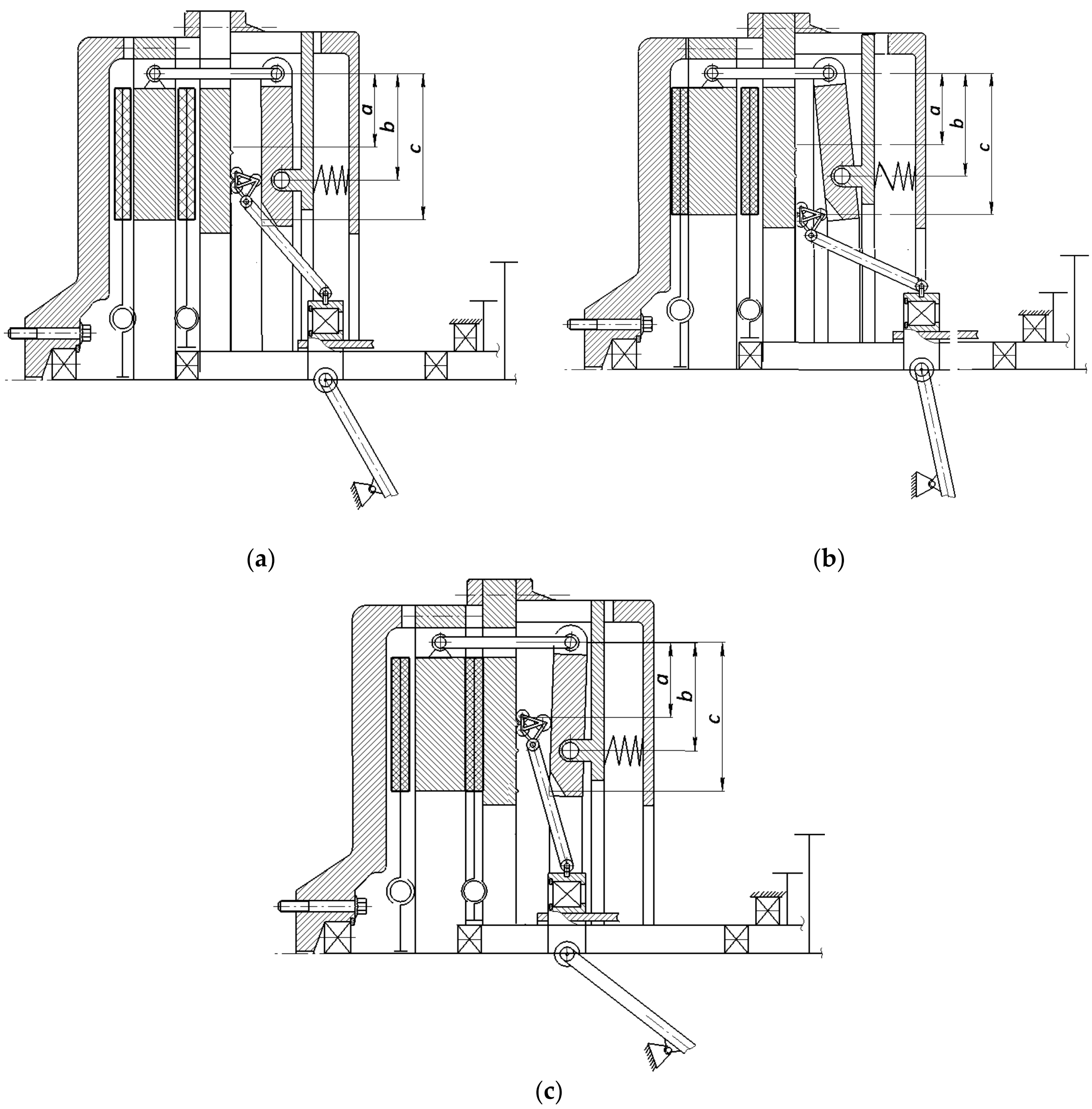
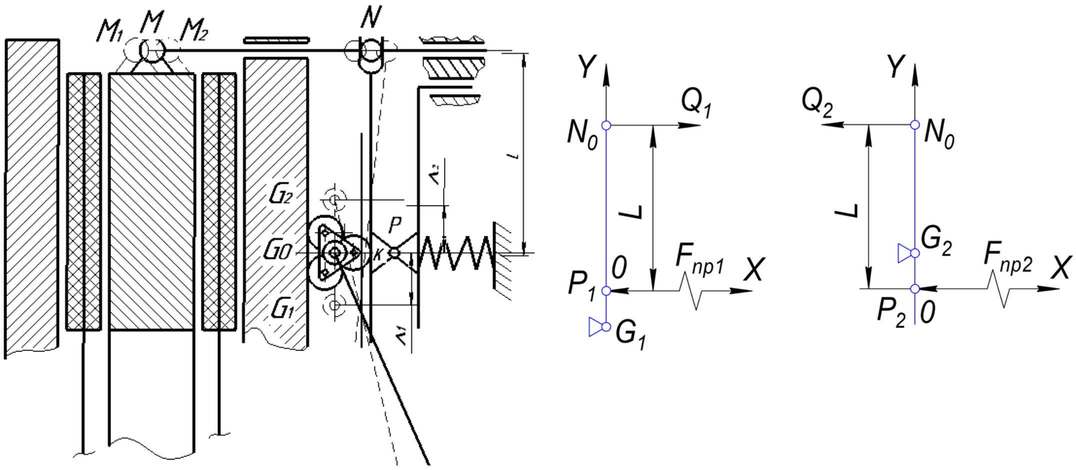


Figure 3. Positions of the movable carriage of the control mechanism: (**a**)—neutral position; (**b**)—1st clutch; (**c**)—2nd clutch; *a*—length determining the position of the carriage under switched-on 2nd clutch, *b*—length determining the position of the carriage under switched-off 1st and 2nd clutches, *c*—length determining the position of the carriage under switched-on 1st clutch.

When the 2nd clutch is switched off, mechanism 9 is set to a position in which stops 20 move from the periphery of the discs to the center through levers 25 (Figure 3a). Rollers 21 and 22 move along inclined conical surface 26, overcoming the resistance of clamps 27. These rollers are installed in a fixed position. Rollers 23 are mounted on the same axis of horizontal hinge 14. The additional disc 10 is pushed into holes 13 of casing 1 through protrusions 12, overcoming the resistance of compression springs 11. They also move axially away from platter 3. In this case, stops 15 transmit the pushing force to levers 17, turning on hinges 14 through hinges 16. Levers 17 transmit the force to pressure plate 8 through hinges 19. Pressure disc 8 stops interacting with driven disc 5 and opens the circuit. As a result, the torque transmission from the engine to shaft 7 stops.

2.2. The Mathematical Model

A design model of the clutch shift mechanism is proposed to analyze the effect of DCT's carriage movement on clutch margins (Figure 4).



(**a**) (**b**) (**c**)

Figure 4. The original DCT: (**a**)—the design scheme; (**b**)—1st clutch; (**c**)—2nd clutch.

The maximum engine torque, the clutch margin, and the number of friction pairs determine the required compression force Q for the clutch friction pairs. The clutch margin for the light vehicle is β = 1.2, 1.7–2.0 for all-terrain vehicles and 3.0 for heavy machinery. The calculation conditionally considers a single rotary lever and pressure spring. However, the number of levers can reach 3–4, and the number of springs can reach 18–42, depending on the transmitted engine torque.

The total pressing force F_{np} is summarized by all the pre-compressed springs. It is conventionally attached to the clutch swing lever NP, which is connected to the clutch pressure plate via trailing lever NM at point M. When moving the carriage from position G_0 to positions G_1 and G_2, the pressure plate moves from position M to positions M_1 and M_2 (Figure 3a). The action line of the total force Q coincides with the lever NM. It is also located at a distance L from the pressure spring line of action.

The coordinate system XOY is used to study the clutch operation. The axis OX is directed along the compression spring action line. The axis OY is directed along the swivel disc in its neutral position ON_0. The full switch-on of the clutch is considered. Vertical lines M_1 and M_2 determine the extreme positions of the friction pressure plates for the switched-on 1st and 2nd clutch, respectively, with coordinates x_{M1} and x_{M2}.

A pre-compressed pressure spring is used to provide the required force Q. The proposed design scheme allows determining dependencies between the force Q_j (j = 1, 2) on the pressure plate and force F_{npj} of the clutch pressure spring when switching on the 1st (Figure 3b) and the 2nd (Figure 3c) clutch, so during vertical movement of the carriage from the neutral position G_0 to working positions G_j by the distance of Δ_{Kj} (Figure 3a).

The equilibrium condition of the j-th lever relative to the reference point G_j is as follows:

$$\Sigma M_{G_j}(F) = 0, \tag{1}$$

or for the cases of switching on the 1st and the 2nd clutches, respectively:

$$F_{np1}\Delta_{K1} = Q_1(L + \Delta_{K1});\ F_{np2}\Delta_{K2} = Q_2(L - \Delta_{K2}). \tag{2}$$

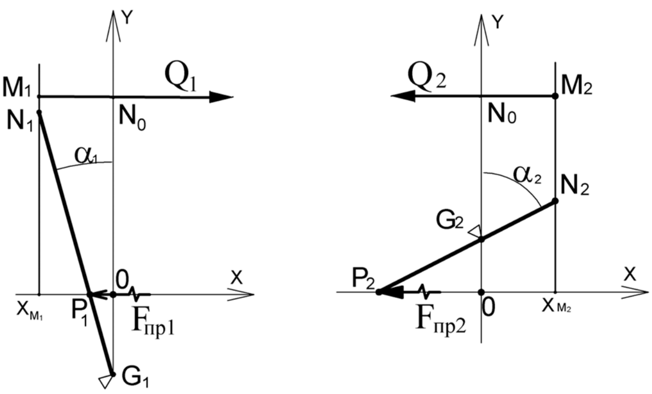
from which the following expressions can be obtained:

$$Q_1 = \frac{F_{np1}\Delta_{K1}}{L + \Delta_{K1}} = \frac{F_{np1}}{\gamma_1 + 1};\ Q_2 = \frac{F_{np2}\Delta_{K2}}{L - \Delta_{K2}} = \frac{F_{np2}}{\gamma_2 - 1};\ (\gamma_2 > 1), \quad (3)$$

where the following parameter is introduced:

$$\gamma_j = \frac{L}{\Delta_{Kj}}. \quad (4)$$

Formula (3) shows that the inequality $F_{np1} > F_{np2}$ ensures equal forces on the pressure disc ($Q_1 = Q_2 = Q$) in the case of similar carriage movements. To fulfill this condition, the design schemes in Figure 5 are considered.



(**a**) (**b**)

Figure 5. The design schemes of the switched-on DCT: (**a**)—1st clutch; (**b**)—2nd clutch.

When the support carriage G is moved to position G_j, the support lever rotates under the force F_{npj} from the pressure spring at point P_j. This fact provides the required compression force Q_j of the clutch friction pairs. For this case, pressure spring forces on the platter surface will be as follows:

$$F_{npj} = {}_{np}\left(\Delta_0 - \Delta_j\right), \quad (5)$$

where c_{np}—compression spring stiffness [kN/mm]; Δ_0—spring pre-deformation [mm] for the neutral position of the carriage (when clutches are switched off); Δ_j—additional deformation of the pressure spring when turned on the j-th clutch (from Figure 4, $\Delta_1 = OP_1$, $\Delta_2 = OP_2$). Particularly, after switching on the 1st clutch (Figure 4), the rotary lever reaches the position N_jG_j, where $x_{N1} = -z_1$. Gap neglecting z_{c1} occurs for the freewheel, and elastic deformation of friction linings equals z_{y1}. Therefore, $z_1 = z_{c1} + z_{y1}$.

Assuming that the lever N_jG_j does not deform, the following geometric expression can be written:

$$\Delta_j = \Delta_{Kj} tg\alpha_j, \quad (6)$$

where α_j—j-th lever angle [deg], determined from the equation:

$$L\ sin\ \alpha_j = z_j{}^{-}1j + 1\Delta_{Kj} tg\alpha_j. \quad (7)$$

Thus, Equation (5) can be rewritten as follows:

$$F_{npj} = {}_{np}\left[\Delta_0 - \Delta_{Kj} tg\alpha_j\right]. \quad (8)$$

The dependencies (1)–(8) allow studying the switching-on clutches in the DCT.

3. Results

Figure 6 shows the effect of carriage displacement Δ_K on the force F_{npk} in the pressure spring and the force Q_k on the pressure disc when switching on the k-th clutch.

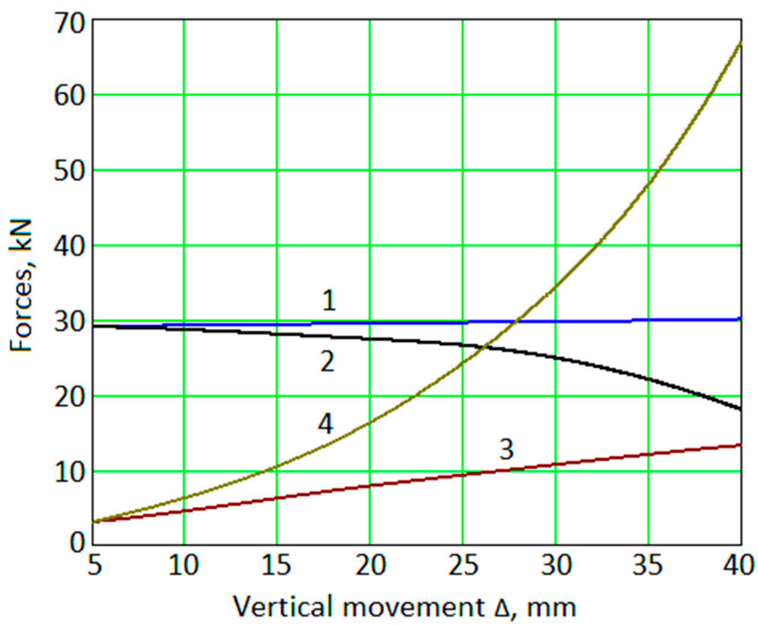


Figure 6. Dependencies of forces Q_k and F_{npk} [kN] on the pressure disc (1,2) and spring (3,4) on the vertical movement Δ [mm] of the carriage: 1—F_{np1}; 2—Q_1; 3—F_{np2}; 4—Q_2.

A dry clutch is considered a test case for simulating the DCT workflow. This clutch is widely used in the automotive and agricultural industries. It also has the following parameters: L = 0.053 m, $z_1 = z_2 = z = z_c + z_y$, z_c = 2.0 mm, z_y = 0.8 mm, c = 950 kN/m, Δ_0 = 30 mm.

The analysis results presented in Figure 6 shows that the force on the pressure plate will be different for a uniform vertical movement of the carriage Δ_K (when switching on the 1st and the 2nd clutches). Moreover, $Q_2 > Q_1$ and $F_{np2} > F_{np1}$. In this case, F_{np1} changes insufficiently.

An increase in Δ_K increases the difference $Q_{21} = Q_2$—Q_1 between forces on the pressure disc and decreases the clutch margin. Notably, such a difference in the pressure disc also increases but is lower. Hence, it is not possible to provide the same clutch margin. Moreover, equality $Q_1 = Q_2$ cannot be provided by choosing different carriage displacements ($\Delta_{K1} \neq \Delta_{K2}$).

Therefore, to equalize the clutch margin, it is proposed to abandon the strictly vertical movement of the carriage. It is also suggested to constructively realize an additional horizontal movement of the carriage. In this case, to equalize forces Q_{K1} and Q_{K2}, the force Q_{K1} should increase, and the force Q_{K2} should be decreased. The corresponding change in the position of points P_1 and P_2 of the carriage contact with the pressure spring can achieve the last condition. In other words, the platter profile along which the carriage moves under switching on/off the DCT should be changed.

In the proposed improved DCT design [32], rollers 21 and 22 move on the inclined conical surfaces 26 and 27 of the support disc. The corresponding design schemes are presented in Figure 7.

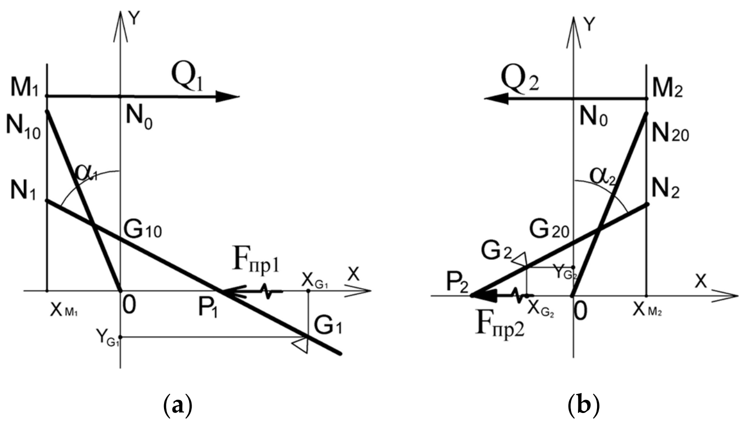


Figure 7. Design schemes of the improved DCT: (**a**)—1st clutch; (**b**)—2nd clutch.

It is convenient to characterize the clutch operation when the k-th clutch is switched on by changing the position of the rotary lever N_kP_k with the inclination angle α_k. The extreme positions of the friction pressure plates are conditionally marked by vertical lines M_k. Point N_k of the rotary lever has the coordinate x_{Mk} = const.

The following dependencies determine the coordinates of characteristic points N_k and P_k for the swivel disc:

$$\begin{array}{lll} N_k: & x_{N_k} = (-1)^k z; & y_{N_k} = L\cos\alpha_k; \\ P_k: & x_{P_k} = -(-1)^j (L\sin\alpha_k - z_k); & y_{P_k} = 0, \end{array} \tag{9}$$

where $z_k = \mid x_{Mk} \mid$.

When the clutch is entirely switched on, gap neglect occurs, and the friction linings are elastically deformed. Therefore, the initial position of the rotary lever (at the initial position of the carriage G_0) can be represented by levers ON_{10} and ON_{20}. In this case, $x_{N10} = x_{M1}$, $x_{N20} = x_{M2}$, and the initial inclination of the rotary disc is characterized by angles α_{10}, α_{20}.

At a specific position of the rotary lever, the carriage position is determined from the following expressions:

$$\begin{array}{lll} G_1: & x_{G_1} = x_{P_1} + \Delta_{K1} tg\alpha_1; & y_{G_1} = -\Delta_{K1}; \\ G_2: & y_{G_2} = -\Delta_{K2}; & x_{G_2} = x_{P_2} - \Delta_{K2} tg\alpha_2, \end{array} \tag{10}$$

where Δ_{Kk}—vertical movement of the carriage [mm].

The pressing force Q_k of the friction pairs and the force F_{np1} from the pressure spring (when the j-th clutch is switched on) are determined by Formulas (5)–(8).

The current deformation Δ_k of the pressure spring (the value of OP_k) is determined by the coordinate x_{Pk}. For the investigated clutch design (Figure 1b), under the switching-off of the 1st clutch, the pressure spring is additionally compressed ($\Delta_1 > 0$). For the 2nd clutch, the force in the pressure spring is decreased ($\Delta_2 < 0$). This fact increases the value of Q_1 and decreases the value of Q_2.

Figure 8 shows the impact of specified horizontal carriage displacement (considered by the angle α_k) with a vertical displacement of the carriage by $\Delta_k = \mp 15$ mm for the considered case study.

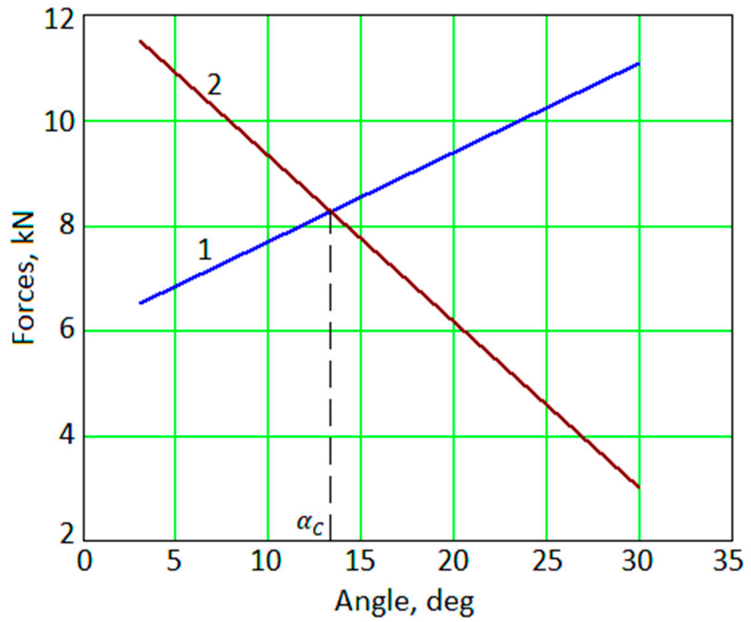


Figure 8. Dependencies of forces Q_k [kN] on the lever rotation angle α_k [deg]: 1—Q_1; 2—Q_2.

As the study results show, after considering the horizontal displacement of the carriage, an increase in the angle α_k (an increase in the value of $|x_{Gk}|$) decreases Q_2 and increases Q_1. However, despite the previous design scheme when $x_{Gk} = 0$, the following equality $Q_2 = Q_1$ takes place under a specific value of the angle α_c. Therefore, the problem of equality for the clutch margins has been solved.

Figure 9 shows results on changes in the forces $Q_1(\alpha_1)$ and $Q_2(\alpha_2)$ on the pressure disc at different values of vertical displacements of the carriage Δ_K = 5–30 mm with a step of 5 mm.

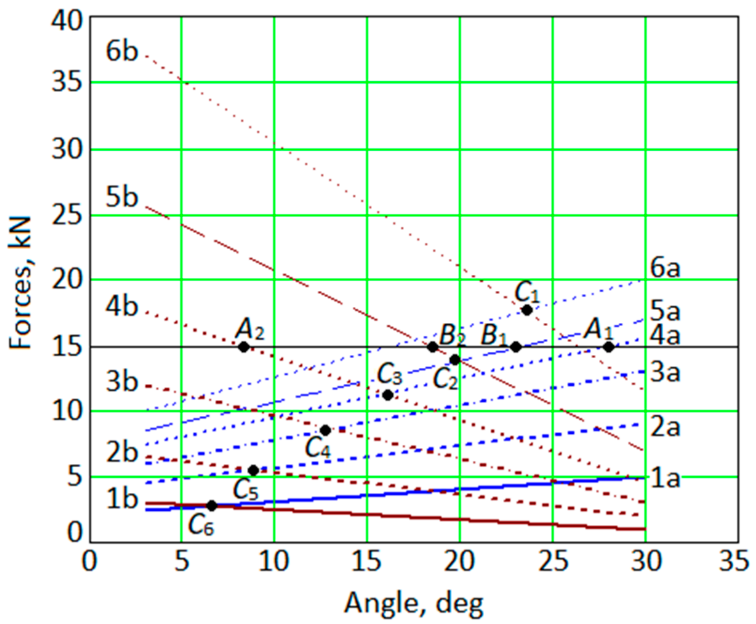


Figure 9. Dependencies of $Q_k(\alpha_k)$ on the angle α [deg] for different displacements Δ [mm]: a—Q_1; b—Q_2; 1—Δ = 5 mm; 2—Δ = 10 mm; 3—Δ = 15 mm; 4—Δ = 20 mm; 5—Δ = 25 mm; 6—Δ = 30 mm.

After applying the obtained results, it is possible to select the *k*-th position of the rotary lever and, accordingly, the position of the support carriage, which provide equal friction coefficients (since $Q_1 = Q_2$) in the case of choosing similar vertical displacements of the carriage ($\Delta_{K1} = \Delta_{K2}$) when the clutches are switched on.

Particularly, ensuring equal forces on the pressure plate ($Q_1 = Q_2 = 15$ kN) under the switched-on clutches 1 and 2 can be realized using the vertical movement of the carriage by $\Delta_{K1} = \Delta_{K2} = 20$ mm (points A_1, A_2) or through vertical displacement by $\Delta_{K1} = \Delta_{K2} = 25$ mm (points B_1, B_2). The levers' positions (angles α_1 and α_2) are different in this case. This complicates surface treatment technology for the platter on which the carriage moves and the corresponding control system.

However, as the results show, this shortcoming can be overcome. Particularly, for the chosen vertical movement Δ_K of the carriage, the lever position (points C_i, $I = 1, 2, \ldots, 6$) can be determined to satisfy the following equalities: $Q_1(C_i) = Q_2(C_i)$, $\alpha_1(C_i) = \alpha_2(C_i) = \alpha(C_i)$. This case corresponds to equal ratios for the 1st and 2nd friction clutches. It also facilitates the platter surface design on which the carriage moves.

Figure 10 shows the results for the dependencies between the forces difference $Q_{21} = Q_2 - Q_1$ (on the pressure plate) and the corner $\alpha = \alpha_1 = \alpha_2$ for some values of vertical movements $\Delta_K = \Delta_{K1} = \Delta_{K2}$ of the carriage.

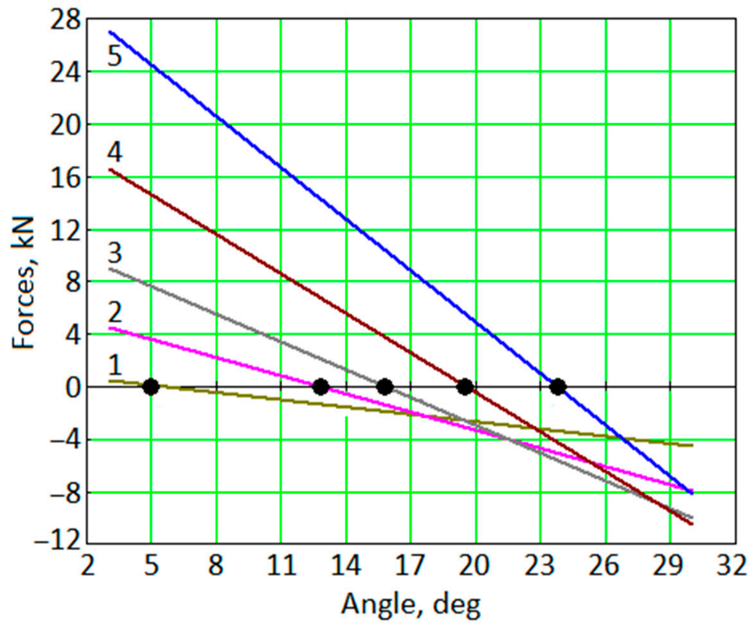


Figure 10. Dependencies of $Q_{21}(\alpha_{21})$ [kN] on the displacement Δ [mm]: 1—$\Delta = 5$ mm; 2—$\Delta = 10$ mm; 3—$\Delta = 20$ mm; 4—$\Delta = 25$ mm; 5—$\Delta = 30$ mm.

When synthesizing the clutch control mechanism, based on the conditions $\Delta_K = \Delta_{K1} = \Delta_{K2}$ for providing the equal position of the rotary lever ($\alpha_1 = \alpha_2 = \alpha_c$), the primary attention should be focused on the results presented in Figure 11.

Based on the required loading Q_P for the clutch friction pair compression, values for the carriage movement Δ_K and the inclination angle α_c should be chosen properly. Such a rational choice allows for designing the clutch disc's surface profile, providing equal ratios for the 1st and 2nd clutches. The following case study illustrates this. Let the required force $Q_P = 11$ kN be given, which corresponds to point *A* (Figure 11). Furthermore, point *B* (on the curve *Q*) should be found, which corresponds to the vertical movement $\Delta_{K1} = \Delta_{K2} = \Delta K = 20$ mm of the carriage (point *D*). This point *D* allows determining point *C* (on the curve α_C) and the angle $\alpha_c = 16°$ (point *E*).

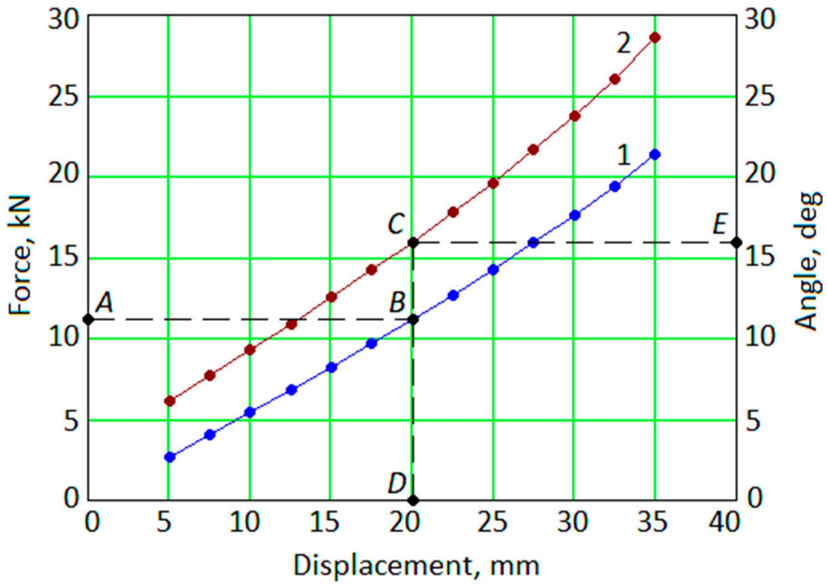


Figure 11. Dependencies of $Q(\Delta_K)$ [kN] (1) and $\alpha_c(\Delta_K)$ [deg] (2) on the displacement Δ [mm].

Suppose that according to the conditions of the task, it is required to change the force Q_P. In that case, new values of Δ_K and α_c can be similarly determined, or the calculated values Δ_K = 20 mm and α_c = 16° can be left (which correspond to the force Q^* = 11 kN), but the stiffness c_{np} of the pressure spring should be changed $\chi = Q^*/Q_P$ times.

Remarkably, when choosing the vertical displacement Δ_K of the carriage, the following limits should be considered: $\Delta_{K2} < \Delta_{K2max}$, where $\Delta_{K2max} = \Delta z_c + y_{K2max}$ (Figure 12).

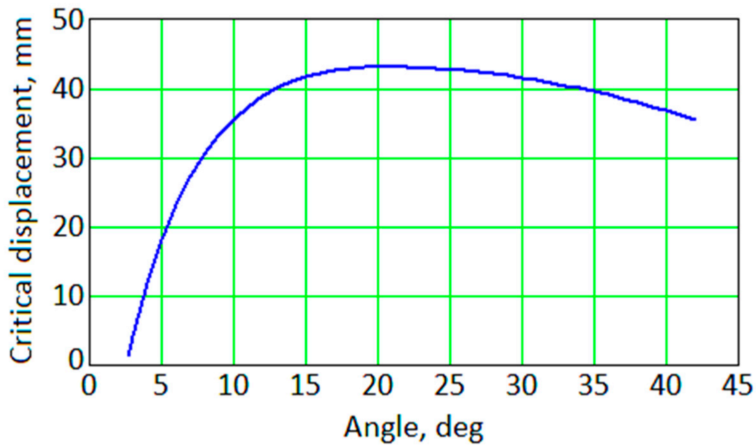


Figure 12. Limitation for the vertical displacement Δ_{k2max} [mm] of the carriage for different values of the angle α [deg] when the 2nd clutch is switched-on (L = 53 mm and z = 2.8 mm).

The above recommendations are mainly related to ensuring the equal vertical movement of the carriage ($\Delta_K = \Delta_{K1} = \Delta_{K2}$) when switching on the clutch. However, there may be other criteria for synthesizing the clutch control mechanism. Particularly, if there is a need to ensure the equal displacement ($S_1 = S_2$) of the carriage, these displacements should be determined as follows:

$$S_k = OG_k = \sqrt{x_{G_k}^2 + y_{G_k}^2}, \tag{11}$$

where x_{G_k}, y_{G_k}—coordinates of the carriage on the platter surface.

In this case, the equation of a straight line that passes through the points N_K and P_K (Figure 4) should be written as follows:

$$\frac{y - y_{P_K}}{y_{N_K} - y_{P_K}} = \frac{x - x_{P_K}}{x_{N_K} - x_{P_K}}. \quad (12)$$

Using expressions (9) for the coordinates of points N_K and P_K, this dependence can be rewritten as follows:

$$y = (-1)^K ctg\alpha_K \cdot (x - x_{P_K}). \quad (13)$$

The movement of the carriage equals S_K [mm]. Therefore, point $G_K(x_{Gk}, y_{Gk})$ is placed both on the circle S_K and straight line $N_k P_k$:

$$\begin{cases} x_{G_k}^2 + y_{G_k}^2 = S_k^2; \\ y_{G_k} = (-1)^k ctg\alpha_k \cdot (x_{G_k} - x_{P_k}). \end{cases} \quad (14)$$

The solution of this system allows determining the angle α_K of the rotary lever for carriage located at point $G_K(x_{Gk}, y_{Gk})$ on the distance of S_K, using parameters α_K and $\Delta_K = y(G_k)$.

However, to synthesize the surface profile of the support disc, it is more convenient to use the polar coordinate system (S_k, β_k), where β_k is the angular coordinate (Figure 13):

$$\beta_K = -arctg\left(\frac{x_{G_K}}{y_{G_K}}\right). \quad (15)$$

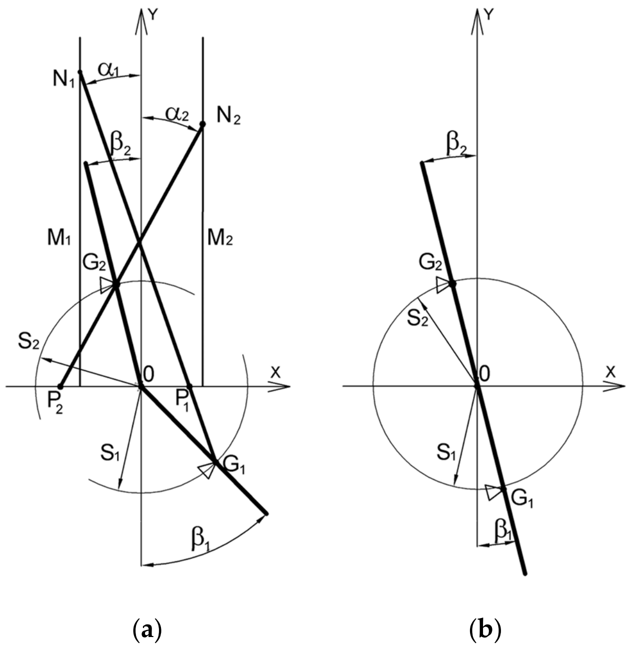


Figure 13. The conical contact surface of the platter: general (**a**) and simplified (**b**) case studies.

After considering the issues of simplifying the technology of manufacturing a conical surface, the clutch control system can be simplified using the following limitation:

$$\beta_1 = \beta_2. \quad (16)$$

Figure 14 shows the changes in the forces $Q_K = Q(S_k, \beta_k)$ acting on the pressure plate.

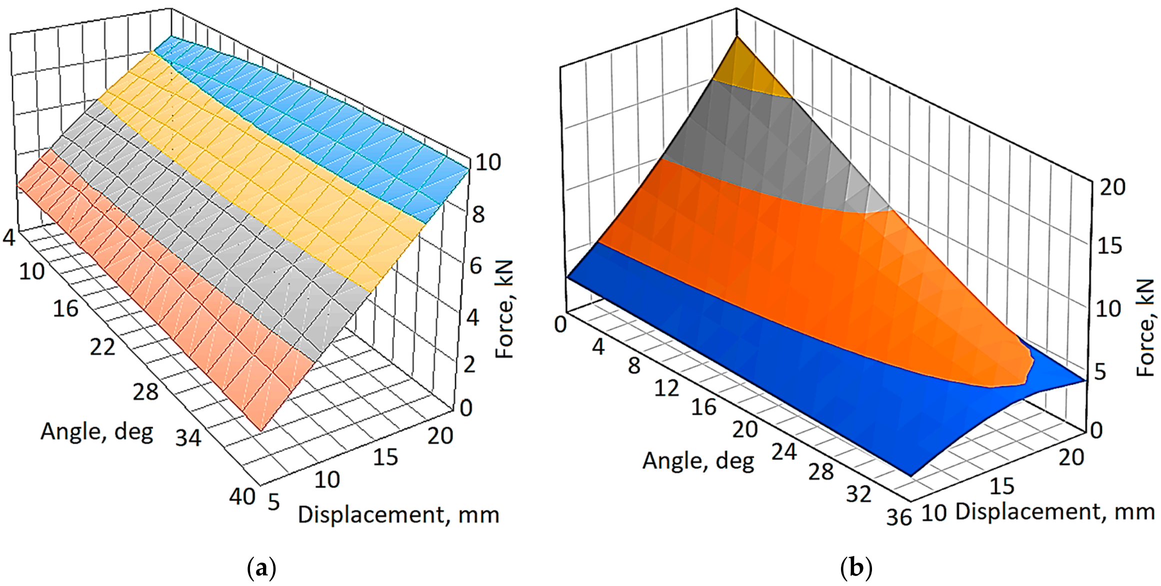


Figure 14. Surfaces $Q_K = Q(S_1, \beta_1)$ [kN] (**a**) and $Q_K = Q(S_2, \beta_2)$ [kN] (**b**) on the displacement S_k [mm] and the angle β_k [deg].

The results show that condition (16) can be satisfied. Particularly, the locus of these points that meets this condition is presented in Figure 15.

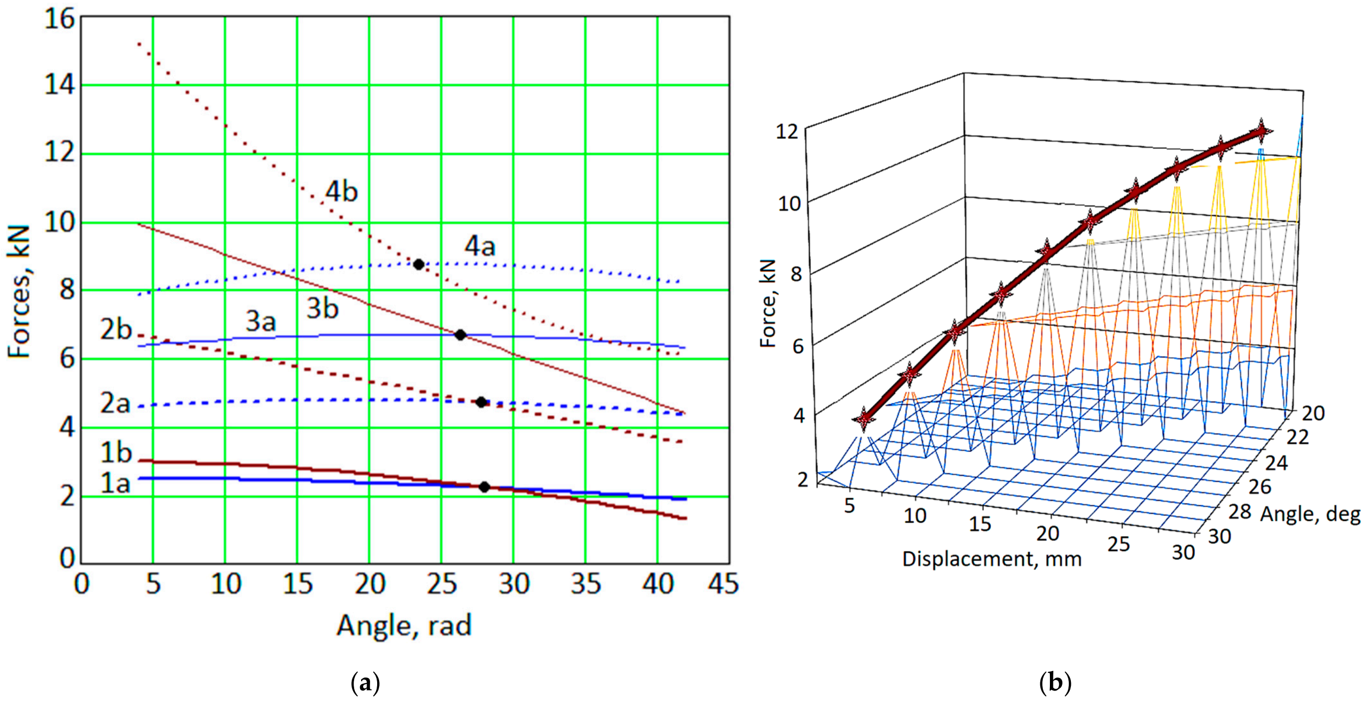


Figure 15. Dependencies of $Q_k(S_k, \beta_k)$ [kN] (**a**) and $Q(S, \beta)$ [kN] (**b**) on the displacements S, S_k [mm] and the angle β_k and β [deg]: a—Q_1; b—Q_2; 1—S = 5 mm; 2—S = 10 mm; 3—S = 15 mm; 4—S = 20 mm.

The synthesis results presented in Figure 16 are also of interest. Particularly, it demonstrates the dependence of the carriage movement on the force acting on the pressure disc.

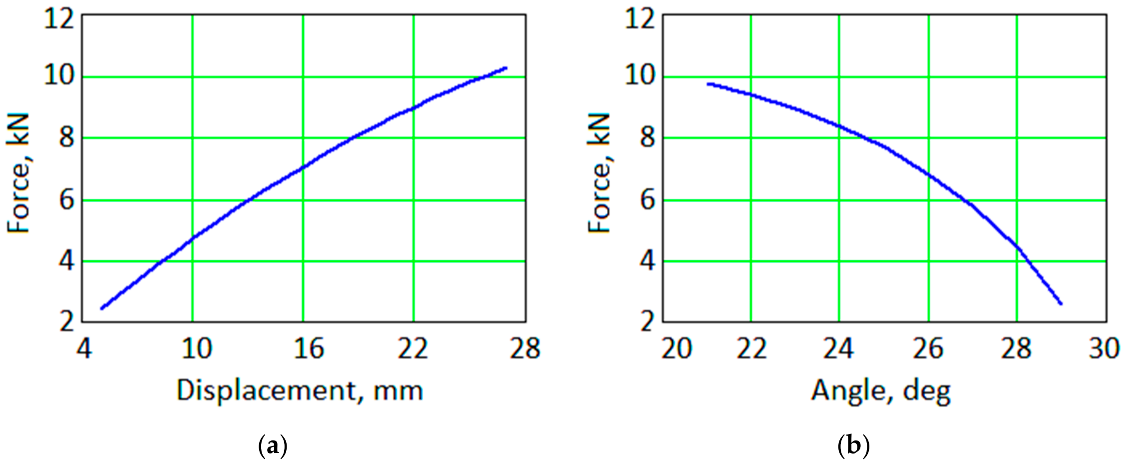


Figure 16. Dependencies of the forces Q [kN] on the displacement S_k (**a**) and the angle β [deg] (**b**).

Notably, the maximum carriage movement depends on the length L of the rotary lever and the parameter z. For the considered case study, $S < 24.5$ mm.

The minimum carriage movement S_{min} is determined by the wear resistance of the carriage movement mechanism. In this case, the inclination angle of the surface profile of the platter is in the range of $\beta = 20$–$29°$ (Figure 16).

Finally, the carried-out research, the developed methodology, and the corresponding algorithm allow for designing the platter surface profile for the DCT, providing equal clutch margins (Figure 17).

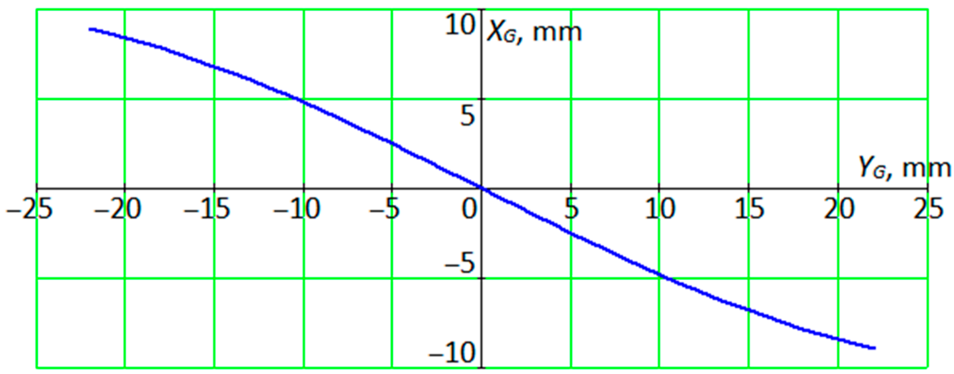


Figure 17. Clutch platter surface profile for equal clutch factors in coordinates X_G, Y_G [mm].

4. Discussion

Comparing the proposed DCT with traditional dual-clutch designs, it can be summarized that it has a reduced energy consumption for clutch control. Particularly, in a fairly common dry dual clutch of the automotive industry, each clutch is switched on by a pressure plate and actuator. The drive of such a clutch uses a separate mechanism and actuator (hydraulic cylinder) connected through a distributor to a pump. This requires additional energy costs. Remarkably, energy is permanently spent on maintaining the clutch in the switched-on position. However, there are no shortcomings in the proposed DCT.

Thus, the considered dry DCT provides fast and practically uninterruptible power flow from the DCT through the clutch. This can improve the vehicle's acceleration dynamics and reduce the energy costs of switching clutches.

Nevertheless, the kinematic and dynamic analyses of the device [33] showed that the loading of the driven discs for each clutch is uneven. Particularly, clutch margins when switching are different. This impacts the service life of each clutch and dynamic processes in the vehicle's transmission.

The novelty of the research is in the substantiation of the mathematical model, the development of the methodology, and the related algorithm for choosing the rational design parameters of the DCT and its drive. This increases the functional performance of the unit while reducing energy costs for control.

The practical significance of the results is to substantiate the possibility of increasing the efficiency of the original DCT. It is also in developing recommendations for determining the rational design parameters and the algorithm's versatility when designing such clutches for automotive, agricultural, and other industries.

5. Conclusions

In the article, the working processes in DCTs have been analyzed. A functional relationship has been established between the drive control parameters and the pressure forces (load capacity) on the clutch friction pairs. The impact of the vertical movement of the control carriage has also been analyzed.

It has been established that only vertical movement of the control carriage does not provide the equality of the clutch margins for the 1st and the 2nd clutch with the equal movement of the carriage.

As a result, a methodology for synthesizing the parameters of the DCT control mechanism has been proposed. The proposed approach is oriented towards the transmission of the required clutch torque. The proposed design solution for the control mechanism substantiates the equalization of the load capacity of the clutch.

The problem of synthesizing the parameters of the surface profile of the disc has also been solved. It allows for providing the equality of clutch margins for the 1st and 2nd clutches. However, the functional limitation regarding the equality of vertical (radial) movements or the general carriage movement for the control mechanism should be satisfied.

The economic effect from the implementation of the obtained results is primarily associated with a decrease in the energy losses of the internal combustion engine and the costs of a vehicle's driver during the operation. The efficiency of the proposed novel clutch is also due to the achievement of the same operating conditions for each clutch. Mainly, equal wear rates of friction pairs and dynamics equalization of the drive wheels (when switching clutches and transmitting the torque of the engine) have been provided. This extends the service life of the clutch and transmission, decreases fuel consumption, and improves the vehicle's environmental safety, traction, and dynamic performance.

Further studies will aim to generalize the proposed algorithm for synthesizing parameters of various types of DCT control mechanisms. Particularly, as part of the strategic problem of the systematic improvement of the vehicle's quality, it is planned to develop a multi-criteria optimization approach considering economic, energy, and environmental quality indicators. This will allow for the optimal design of DCTs with an extended number of parameters.

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