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TEST STAND FOR MODELLING HYDRAULICALLY CONTROLLED CONTINUOUSLY VARIABLE TRANSMISSION

STANOWISKO BADAWCZE DO MODELOWANIA WŁASNOŚCI HYDRAULICZNYCH PRZEKŁADNI CVT

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Summary

A test stand making it possible to analyse the hydraulic processes that take place in the hydraulically controlled automotive continuously variable transmission (CVT), operating with a pushbelt or chain, has been described. The test stand was developed at Lodz University of Technology (TUL), Department of Vehicles and Fundamentals of Machine Design.

The transmissions of this type and the transmission ratio control systems being in use at present have been briefly characterized. In the next part of the article, attention has been focused on the hydraulic control system, which is now most popular. In consideration of the known drawbacks of the classic system with a single fixed-displacement pump, a new engineering solution has been proposed to improve the system efficiency.

The test stand shown here makes it possible to carry out various tests of the new solution mentioned

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above. The method of obtaining similarity between the test setup and the arrangement of elements in a real Jatco CVT7 unit has been presented. This has been achieved by appropriate selection of the type and geometry of the lever system that interconnects the movements of two actuators on the test bench. A new balance of forces exerted by the actuators is achieved for every proportion of the pressures and the corresponding displacements on the test stand are proportional to the displacements in the reference CVT. Some test results have been given as an example.

Keywords: pressures, hydraulic system, CVT, geometrical relationships

Streszczenie

W artykule opisano stanowisko badawcze opracowane w Katedrze Pojazdów i Podstaw Budowy Maszyn Politechniki Łódzkiej, pozwalające na analizę zjawisk hydraulicznych zachodzących w samochodowej przekładni bezstopniowej o sterowaniu hydraulicznym współpracującej z pasem pchanym lub łańcuchem.

W artykule przedstawiono krótką charakterystykę tego typu przekładni oraz stosowane dziś rodzaje sterowania przełożeniem. W dalszej części skupiono się na najpopularniejszym dziś rozwiązaniu hydraulicznym. Wobec znanych niedoskonałości klasycznego układu w tego typu przekładniach z jedną pompą o stałej objętości jednostkowej, opracowane zostało nowe rozwiązanie w celu podniesienia sprawności.

Opisywane stanowisko badawcze umożliwia przeprowadzenie różnorodnych testów wspomnianego, nowego rozwiązania. Osiągnięto to poprzez odpowiedni dobór układu dźwigniowego, sprzęgającego ruch dwóch siłowników, dzięki któremu dla każdego stosunku ciśnień w obu siłownikach wypracowana zostaje nowa równowaga sił a odpowiednie przemieszczenia na stanowisku badawczym są proporcjonalne do przesunięć w referencyjnej przekładni CVT. Przedstawiono sposób w jaki uzyskano podobieństwo stanowiska do układu z rzeczywistej przekładni Jatco CVT7 oraz przykładowe wyniki badań.

Słowa kluczowe: ciśnienia, układ hydrauliczny, przekładnia CVT, zależności geometryczne

1. Introduction

More than a century has passed since the appearance of the first vehicle with a stepless transmission [6]; however, transmissions of this type were not introduced to production on a wider scale before 1965, when all the then DAF manufactured passenger cars were equipped with Variomatic continuously variable transmissions (CVT), developed by Hub van Doorne [12].

Thanks to the development in materials engineering and automatics, which took place in the recent two or three decades, the transmission of big torque values by a pushbelt [12] or chain has become possible. The performance of nowadays internal combustion (IC) engines can be fully used and the engine-CVT connection can be controlled in such a way that optimum conditions of operation of the engine-transmission system can be ensured. The optimization may be assessed on the grounds of e.g. maximum torque or power transmitted, minimum specific fuel consumption, or minimum pollutant emissions.

In result of these achievements, a continuous increase in the share of CVT systems in the production of automatic gearboxes can be observed. A comparison of these figures for 2010 and 2017 (a forecast) has been shown in Fig. 1.

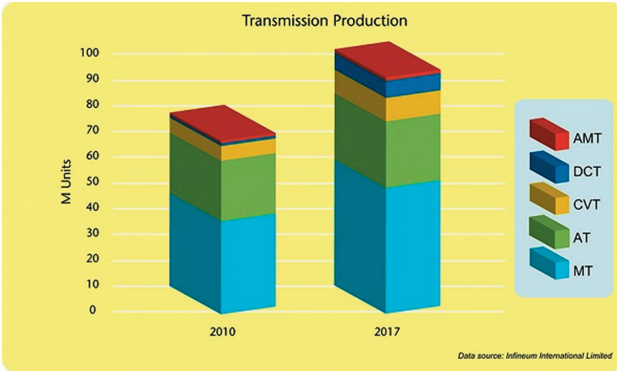

**Fig. 1. Comparison of shares of specific transmission types in the global production [13]
AMT – automated manual transmissions; DCT – dual-clutch transmissions;
CVT – continuously variable transmissions; AT – classic automatic transmissions;
MT – manual transmissions**

The next graph shows the world CVT production (the additional axis of ordinates), in millions, compared with the world production of motor vehicles, in millions as well. In 2020, about every fifth vehicle is to be provided with a transmission of this type, according to forecasts.

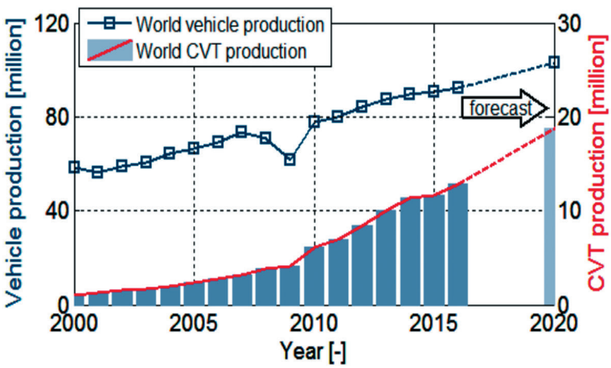

Fig. 2. World CVT production compared with the global production of motor vehicles [11]

The increasing sales volume and further technological progress result in ongoing development of CVTs. Apart from great CVT market players, such as Bosch Transmission Technology [12], LuK [14], or Jatco [15], there are also some smaller companies specializing in developing new CVT design, such as Gear Chain Industrial B.V. [16] or Varibox [17]. These issues are also addressed by numerous technical universities. One of them is the Lodz

University of Technology (TUL), where a new design of the hydraulically operated CVT ratio control system has been developed. The use of such a system on an industrial scale may bring significant energy savings [4].

In this article, the description of the current CVT ratio control methods is followed by presentation of a test stand designed and built at TUL, Department of Vehicles and Fundamentals of Machine Design, for testing hydraulic characteristics of the CVT control system without the need of time- and cost-consuming construction of a transmission prototype.

2. CVT ratio control methods

Among many ideas of stepless change of the transmission ratio, a solution with a pair of conical pulleys and a pushbelt (chiefly manufactured by Bosch) or a chain (chiefly manufactured by LuK) is now very popular.

As regards just the method of axial shifting of the pulley halves (sheaves) and thus controlling the transmission ratio, three construction types may be discerned [1]: hydraulic [9], electro-hydraulic [2], and electromechanical [8].

At present, the definitely greatest popularity has been gained by a system with hydraulic actuators and this is the solution to be addressed in the subsequent part of this article. A typical solution adopted by CVT manufacturers (e.g. Jatco) has been presented in Fig. 3.

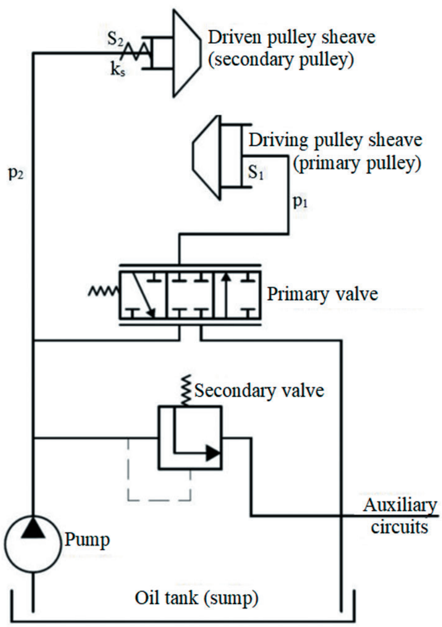

Fig. 3. Conventional hydraulic control system of a CVT [7]

According to [7], the clamping forces in the conical pulleys are hydraulically generated by the action of pressure (p_1 and p_2 for the driving and driven pulley, respectively) on the corresponding clamping surface (S_1 and S_2). Only the movable sheaves of the conical pulleys have been shown in the drawing. The secondary valve controls the pressure delivered by the pump and, thus, the pressure in the circuit of the driven pulley (i.e. in the secondary circuit). The overflow from the pump is directed towards auxiliary circuits (oil cooler and filter). Pressure p_1 in the circuit of the driving pulley (i.e. in the primary circuit) is controlled by the primary valve, which reduces the input pressure (p_2). If the electrohydraulic valves are properly set, pressures p_1 and p_2 may be so selected that any stationary working point of the transmission could be maintained. The clamping surface of the driving pulley actuator is bigger than the clamping surface of the driven pulley actuator ($S_1 > S_2$); thanks to this, the clamping force in the driving pulley may be higher than that in the driven pulley although the pressure in the driving pulley circuit cannot exceed that in the circuit of the driven pulley (i.e. $p_1 \leq p_2$).

The system as described above, although very popular (it is used in most cases), suffers from a serious drawback related to the principle of its operation: the system efficiency is quite low, especially at high vehicle engine speeds. This is because the fixed-displacement pump shaft is driven by the vehicle engine through a chain transmission (a solution used in the popular Jatco CVT7 transmission). In the case of a large-displacement pump being used, the pump delivery rate becomes markedly excessive at higher rotational speeds and the oil overflow is released to the sump, which results in energy losses. An answer to the question how much power is consumed by the CVT pump in conventional designs can be found in [5]; some information about the efficiency of continuously variable transmissions and the available methods of improving it has been provided in [1].

Another concept of the control system, the electrohydraulic one is described in the subsequent part of this article and has been shown in Fig. 5. The main difference is the use of electric motors to drive the pump (or pumps). This makes it possible to adjust the oil delivery rate to current needs. An additional good point of such a solution is the fact that the hydraulic system may be operated in vehicles with a start/stop system and in hybrid electric vehicles, where the IC engine is temporarily stopped. The use of a control system of this type may result in a reduction of the necessary transmission control power by more than 83 % and in about 5 % fuel consumption savings in comparison with a CVT controlled by a conventional hydraulic system [2].

One of the concepts of an electromechanical control system has been schematically presented in Fig. 4. In this solution, two double epicyclic gear sets with identical dimensions have been applied. In each gearset, one of the ring gears is connected with a servomotor. The first servomotor makes it possible to control the transmission ratio and the other one controls the clamping force in the conical pulleys. A control system of this type makes it possible to gain up to 10 % fuel consumption savings in comparison with a CVT controlled by a classic hydraulic system [8].

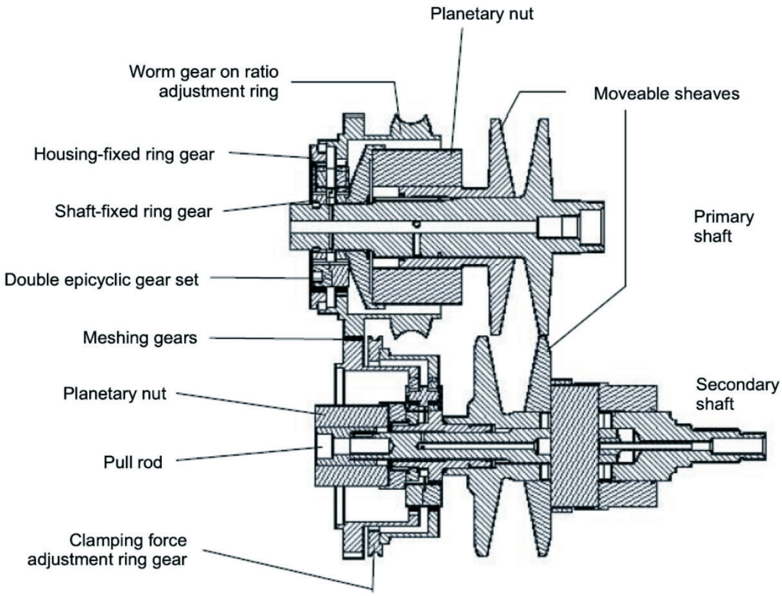

Fig. 4. Schematic diagram of the electromechanical control system of the Empact CVT [8]

3. Method of modelling a real CVT on a test stand

Within work on the hydraulically operated ratio control system for CVT (which is most popular at present), the engineering solution mentioned previously was developed, the use of which may considerably reduce the energy losses in transmissions of this type, especially in the periods when the vehicle moves with a constant transmission ratio.

Details of this solution have been presented in [4] and they will not be thoroughly discussed herein. To enable an analysis of the subsequent part of this article, only a hydraulic diagram will be shown (in Fig. 5), which will explain the system operation principle. A reversible pump "nr 1" is responsible for maintaining appropriate transmission ratio and pressure p_1. It pumps oil between two sheave actuators of equal piston areas. Pump "nr 2" is accountable for maintaining appropriate pressure p_2 in the system and for compensating changes in the volume of the oil contained in the system.

Information about other research centres working on solutions of this type can also be found in the literature (e.g. [16]).

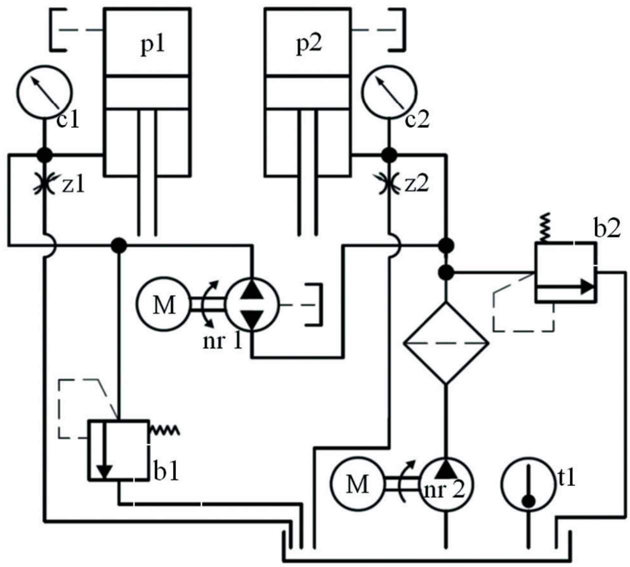

Fig. 5. Schematic diagram of the hydraulically operated CVT ratio control system in the proposed solution [4]

To enable the testing of the hydraulic system proposed as described above, either a transmission prototype or a test stand modelling the hydraulic processes had to be designed and built. In consideration of the time needed to make a complete prototype of the transmission unit with equal actuator piston areas and with two hydraulic supply pumps, a decision was made to develop a test stand to model the transmission behaviour, consisting of two hydraulic cylinders connected with each other by an appropriate lever system (Fig. 7).

An issue of critical importance was the necessity to maintain adequate similarity between changes in the positions (and thus ratio) of the hydraulic actuators used in the test stand and in a real CVT. As the reference transmission, a Jatco CVT7 unit was chosen. While changing the ratio (by change of proportion of clamping forces) from one steady state to a new steady state, the actuator piston displacements in the model and in the reference transmission unit should be proportional to each other. As a result of displacement proportion, the volume of the oil pumped between actuators are as well proportional.

For this problem to be solved, much work had to be done because of a non-linear relation between changes in the transmission ratio and changes in the proportion between the forces exerted by the two actuators. The proportion of the force exerted by the primary actuator to the one of the secondary actuator will be hereafter represented by a variable named KF. In the solution as proposed, with two hydraulic pumps, this proportion is identical with the pressure proportion because of equal actuator piston areas. The said non-linearity of the relation between the clamping force and the transmission ratio in the Jatco CVT7 unit shows the curve plotted in Fig. 6.

It should be clearly stated here that in consideration of participation of a French industrial partner in the development of the solution discussed here, the transmission ratio is defined here as the quotient of the rotational speed of the driven shaft to the rotational speed of the driving shaft, i.e. as a reciprocal of the ratio that is customary in Poland.

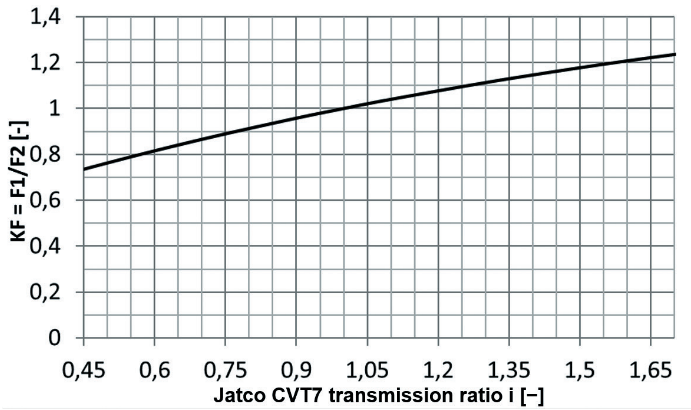

Fig. 6. Jatco CVT7 characteristic curve

The required similarity of the test stand to the real transmission was achieved by appropriate shaping of the lever system that connects the two actuators with each other. The selected position of the pivoting point of both actuators ensures adequate similarity mentioned above in the whole range of transmission ratios; thanks to this, moreover, a new equilibrium of forces exerted by the actuators is developed for every ratio between the oil pressures in the actuators.

The geometrical relations that enable the finding of the appropriate position of the pivoting point have been presented below, in the form of equations (1)-(9); for the notation, see Fig. 7.

The red colour indicates the points and distances whose value or position changes depending on the position (simulated ratio) of the lever system, except for the arm lengths, variable as well, but drawn in green. The elements or dimensions whose position or value is constant have been drawn in blue.

H_1, H_2 – displacement of actuator piston rods from the positions corresponding to equal pressures in both actuators; the positive sign means increased overall actuator length;

B – overall actuator length at equal pressures;

$L = L_1 = L_2$ – lengths of actuator arms;

C – distance between the centre of rotation "O" and the midpoint of lever arms "S";

$\Delta\alpha$ [rad] – deviation from the initial position; the positive sign corresponds to $H1 < 0$;

ram_1, ram_2 – lengths of the arms of the actuator forces relative to the centre of rotation;
$\alpha 0$ – initial deflection;
$\alpha 1$ $= \alpha 0 - \Delta\alpha$
$\alpha 2$ $= \alpha 0 - \Delta\alpha$

$$B + H_1 = \sqrt{(OM^2 + OP^2) - 2 \times OM \times OP \times \cos(\alpha_1)} \tag{1}$$

$$H_1 = \sqrt{(OM^2 + OP^2) - 2 \times OM \times OP \times \cos(\alpha_1)} - B \tag{2}$$

$$B + H_2 = \sqrt{(OM^2 + OP^2) - 2 \times OM \times OP \times \cos(\alpha_2)} \tag{3}$$

$$H_2 = \sqrt{(OM^2 + OP^2) - 2 \times OM \times OP \times \cos(\alpha_1)} - B \tag{4}$$

$$A_1 = \frac{OP \times OM \times \sin(\alpha_1)}{2} \tag{5}$$

$$A_2 = \frac{OP \times OM \times \sin(\alpha_2)}{2} \tag{6}$$

$$ram_1 = \frac{2 \times A_1}{B + H_1} \tag{7}$$

$$ram_2 = \frac{2 \times A_2}{B + H_2} \tag{8}$$

$$i_{sym} = \frac{ram1}{ram2} \tag{9}$$

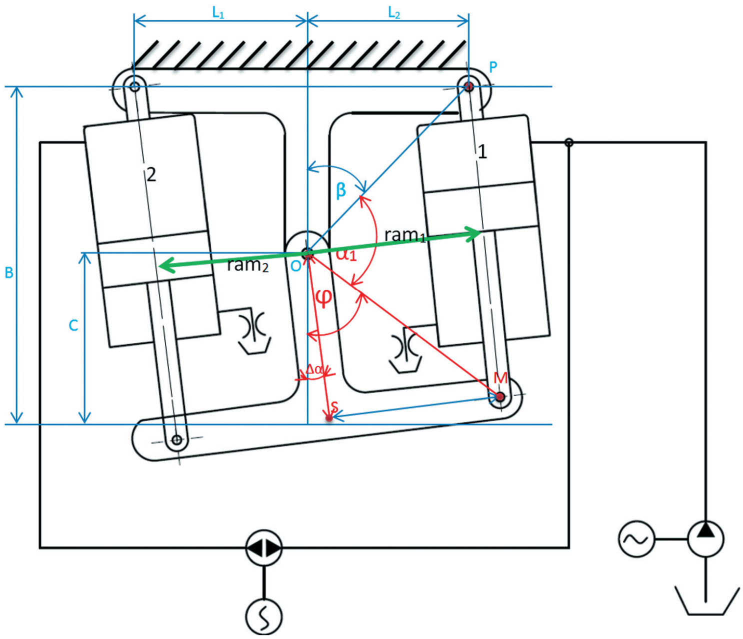

Fig. 7. Schematic representation of the test stand, with the parameters having been shown that are necessary for determining the appropriate geometry of the lever system

If the Jatco CVT7 characteristic curve, i.e. the interdependence between the transmission ratio and the KF value (see Fig. 6), as well as the above equations describing the test stand geometry are known then the position of the pivoting point can be so selected that the said proportionality would be as close to the optimum as possible.

The relation between the positions of the lever system actuators and the simulated ratio of a real transmission may be determined with using the variable KF. With this objective in view, the graph shown in Fig. 6 should be compared with the characteristic curve of the test stand (KF as a function of the displacement of the secondary actuator piston rod) shown in Fig. 8. Here, the displacement of the secondary actuator piston rod was assumed as the independent variable, because the piston rod of this actuator was the test stand component on which the position sensor was installed.

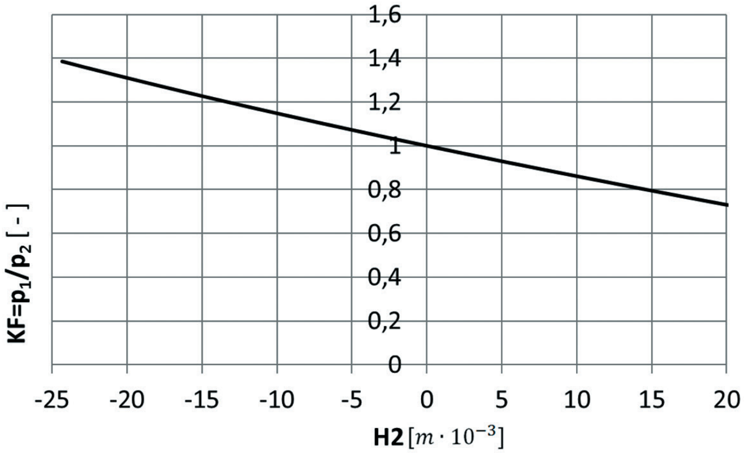

Fig. 8. Test stand characteristic curve

Fig. 9 visualizes the method of determining the ratio of the test stand lever system to model the real CVT ratio, exemplified by three different ratio values. As previously mentioned, the parameter common for the real CVT and the test stand is the proportion between the forces exerted by the two actuators- KF. For the lever system, this proportion is related to the position of the piston rod chosen as a basis- H2 (i.e. the piston rod of the secondary actuator) and for the Jatco CVT unit, it is related to the ratio between the two current push-belt radii, i.e. the transmission ratio.

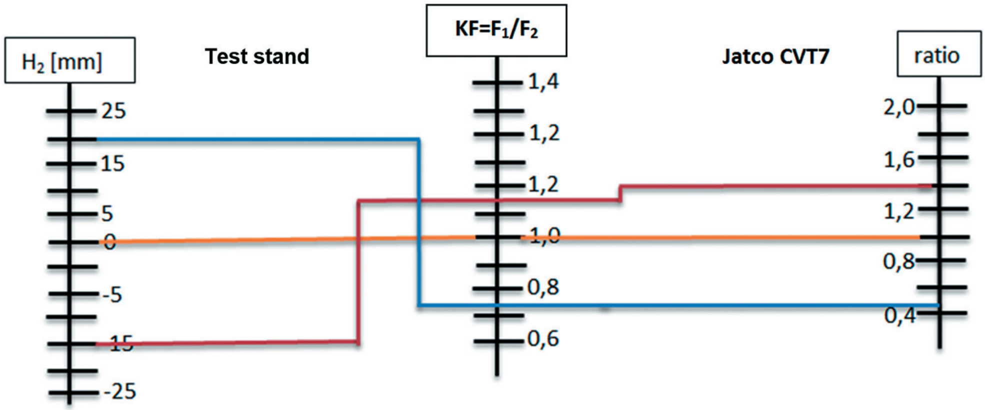

Fig. 9. Relation between the position of the secondary actuator piston rod of the test stand lever system and the ratio of the Jatco CVT7 unit modelled

4. Test stand description

The test stand consists of the following major parts (Figs 5 and 10):

- positive-displacement pump 1 with a fixed displacement of 2 cm^3, operable in both directions, accountable for controlling the transmission ratio, and driven by an inverter-controlled electric motor, where the preset rotational speed value is obtained from a controller;
- positive-displacement pump 2 with a fixed displacement of 11 cm^3, accountable for making up for system leakage, and driven by an electric motor, where the preset rotational speed value is obtained from a controller;
- set of hydraulic actuators $p1$ and $p2$, with a piston diameter of 100 mm and stroke of 50 mm each, and with negligible internal leaks;
- safety valves $b1$ and $b2$ (one per each pump delivery line);
- two choke valves $z1$ and $z2$, making it possible to simulate bigger leaks in the actuators;
- actuator pressure sensors $c1$ and $c2$;
- piston rod displacement sensor, installed on actuator $p2$ and denoted by $s2$;
- oil temperature sensors, situated in the oil sump and denoted by $t1$.

The original actuator seals severely distorted the system characteristics by large hysteresis caused by friction. Thanks to changing the seals, the friction could be significantly reduced with the leaks being kept at a negligibly low level [3].

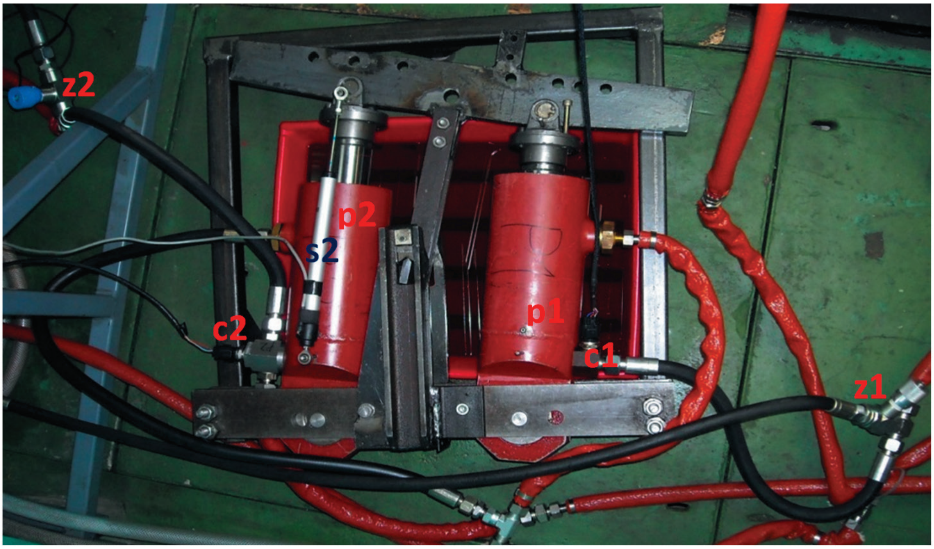

Fig. 10. The photograph shows the most important component part of the test stand, i.e. two actuators coupled with each other in a way that models their cooperation in a CVT

The range of the transmission ratio values obtainable on the test stand is i_{sym} = 0.43-2.4, at the corresponding range of pressures being $\frac{p_1}{p_2}$ = 0.7-1.35. In the CVT operated in the Jatco CVT7, the ratios can vary in similar range.

In the Jatco CVT7 unit, the piston area of actuator $p2$ is almost identical to that of each actuator on the test stand. The stroke of CVT 7's actuator piston rod is 14.5 mm. On the test stand, the actuator piston rod is displaced by 34.8 mm for the ratio to be changed from 0.45 to 1.7. The difference between the piston rod strokes in the real transmission unit and in the test stand system results in a time scale, which is equal to $\frac{34.8}{14.5}$=2.4 if pumps with the same displacement are used on the test stand and in the reference transmission. Thus, all the processes that are modelled on the test stand run slower than the same processes in the Jatco CVT7 unit, with the speed modelled being reduced to a value of 1/2.4 of the real process speed, according to the time scale. This brings a benefit similar to that when certain phenomena are recorded with the use of high-speed cameras and then observed at a reduced speed, i.e. when quick-changing processes are viewed in a slower and more precise manner.

5. Example test results

Thanks to the test stand design as described above, a reliable transmission model could be obtained without the necessity to build a prototype of the transmission unit. With using the hydraulic system more comprehensively described in [4], a number of tests were carried out and their results have been presented below for illustration.

An important and, undoubtedly, very favourable characteristic of the hydraulic system proposed has been shown in the form of a graph in Fig. 11. The graph represents a time history of pressure p_1 at a constant rotational speed of the first pump and a stepwise-changed speed of the second pump. It should be noted that a constant rotational speed of pump 1 ensures an approximately constant value of the pressure in the primary actuator, regardless of pressure p_2. This is because the pressure in the first actuator is exclusively determined by the rate of flow of the oil delivered to the actuator and by the oil leakage inside the actuator; the leaks through the primary pump (1) between the actuators are negligibly small and, thanks to this, the suction pressure of the pump does not play any important role for the value of the delivery pressure of this pump. Similar results were obtained for various values of both pressures, including the case where pressure p_1 was higher than pressure p_2.

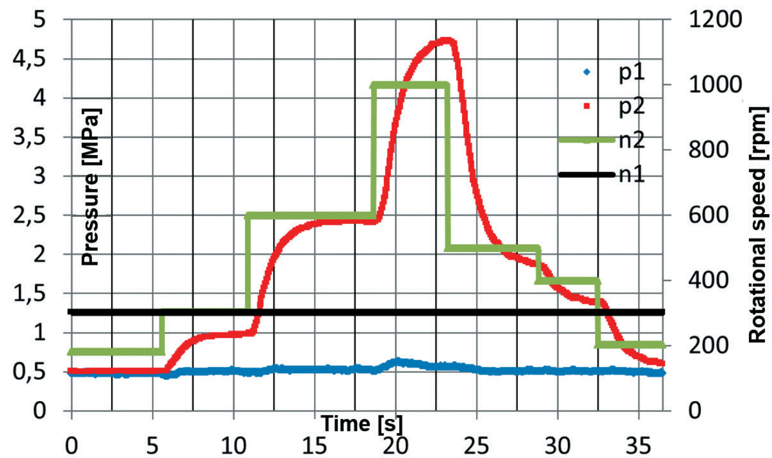

Fig. 11. Time history of pressure p_1 at various values of pressure p_2 (delivery rates of pump 2) [4]

The next graph shows a time history of the transmission ratio achieved on the test stand in response to a sinusoidal signal applied as an input request. Noteworthy is the fact that in spite of the varying value of the pressure in the first actuator, the system was able to maintain an approximately constant value of pressure p_2, which is of critical importance for a continuously variable transmission, because this pressure is responsible for fastening the pushbelt to prevent slippage. Another very important achievement was the maintaining of similar dynamic behaviour of the control system over the whole range of transmission ratios (in spite of the system non-linearity), which was successfully achieved by means of methods similar to those described in [10].

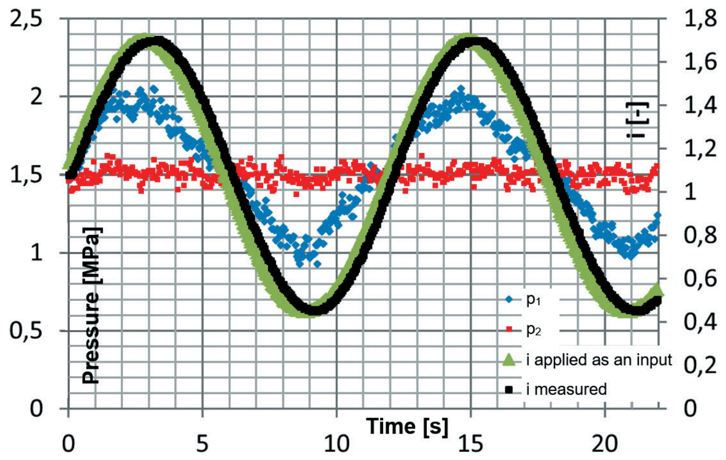

Fig. 12. System response to a sinusoidal request of ratio

6. Recapitulation

Among various methods of axial shifting of the conical sheaves of a continuously variable transmission (CVT), the hydraulic system is most popular.

To test the proposed CVT control system, a test stand was built, based on two hydraulic actuators and a lever system. For adequate similarity to be obtained between the real (reference) transmission and the test stand, the geometry of the latter, especially of its lever system, had to be appropriately designed.

Some results of the tests carried out with using the test stand presented have been given for illustration.

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The full text of the article is available in Polish online on the website http://archiwummotoryzacji.pl.

Tekst artykułu w polskiej wersji językowej dostępny jest na stronie http://archiwummotoryzacji.pl.

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