UDC: 623.438.3-23(047)=20
COSATI: 19-03, 13-09, 14-02

# Identification of hydromechanical transmission behaviour during turning of high-speed tracked vehicles

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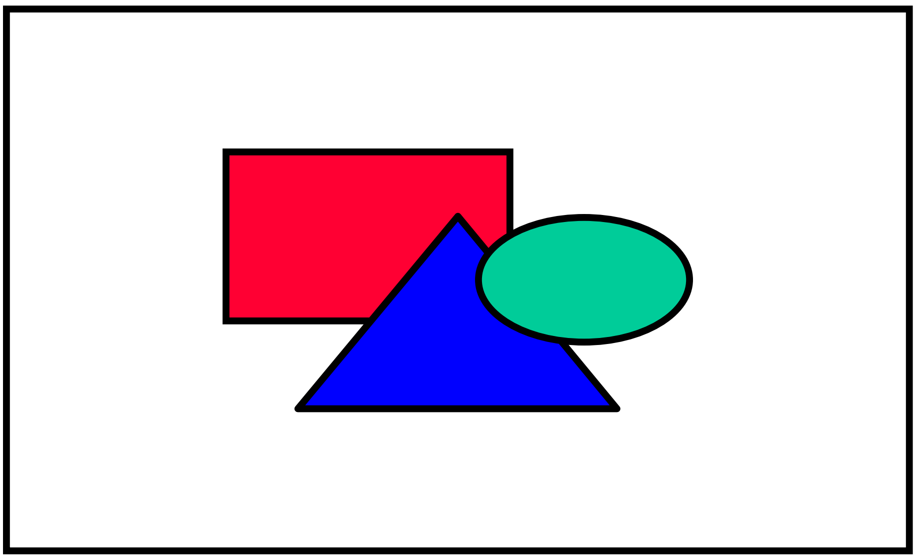
**This paper presents an analysis of tracked vehicle turning with the hydromechanical transmission in which a hydrodynamic gearbox is used for vehicle rectilinear motion and turning. The results of realized functional model laboratory testings obtained by the test stand simulation of transmission function during vehicle turning are presented.**

*Key words:* **tracked vehicle, hydromechanical transmission, hydrodynamic gearbox, turning, testing.**

## Introduction

THE rectilinear motion of tracked vehicles is realized in the same way as in wheel vehicles, by changing gear ratios in gearboxes. For tracked vehicle turning it is necessary to achieve different track revolve speeds. This difference can be achieved in a few ways: by independent mechanisms for steering, by steering systems in transmissions in block, and by sideway gearboxes, depending on transmission design.

For high traction and speed characteristics during rectilinear motion and turning, it is necessary to use a gearbox with continual change of transmission ratio and a steering system that enables continual change of turning radius. The transmissions with both mechanical and hydraulic components (hydrodynamic gearbox and hydrostatic transmission), so-called hydromechanical transmissions (HMT), have such characteristics.



The result of theoretical research work is the HMT functional model solution, with a hydrodynamic gearbox (HDM) in the main drive arranged in the parallel link, while the auxiliary drive power is obtained from the HDM pump circuit. The purpose of laboratory testings was to confirm the theoretical research results with regard to gear ratio continuity in rectilinear motion and in the process of turning. This paper presents a kinematic scheme of a transmission functional model with the description of its functioning, analyzes the transmission functioning during the vehicle turning and gives the laboratory testing results.

## Functional model of hydromechanical transmission

Fig.1 shows a kinematic scheme of the HMT functional model where the hydrodynamic gearbox is connected parallelly with the planetary differential gear train (DP). The mechanical part of the HMT contains a part of transmission in the block with nonsymmetrical drive, which during turning, provides the decrease of the output shaft angular velocity on the inner track. During this process, the angular velocity of the output shaft on the outser track does not change. The transmission consists of the following: HDM, main shaft (GV), differential gear train (DP), change speed gearboxes (M), drive shaft (PV), summarizing planetary gear trains (SPP), brake for rectilinear motion (K), auxiliary drive (PP), auxiliary drive shafts (V) and clutches for turning (S).

When turning, it is necessary to switch off the brake for rectilinear motion (K) in the transmission (Fig.1) on the inner track,

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and to switch on the clutch for turning (S). In this way the power from the auxiliary drive is transmitted to the SPP sun gear. Due to the fact that outer track the speed is the same as before (during rectilinear motion) because of different output shaft speeds, the vehicle turns. After the turn, the clutch is switched off and the brake is switched on, so the vehicle continues its rectilinear motion.

As it can be seen, the power is transmitted from the HDM pump circuit via the PP gear to the auxiliary drive shaft (V). Such conception enables the power supply to be independent of the hydraulic gear ratio in HDM.

The vehicle turning process with such a transmission can be divided into two phases:

**The first phase** comprises gradual switching off the brake on the inner track, so the speed of this track depends on both the hydraulic gear ratio in the HDM and the speed of the SPP sun gear (Fig.1) depending on brake slipping.

**The second phase** starts with a gradual switching on the clutch on inner track (the brake being completely switched off) and ends when the clutch is switched on. By switching on the clutch, the power is transmitted to the SPP sun gear via the auxiliary drive, providing the reduction of rotational speed of the output shaft.

## Turning kinematics of vehicles with the hydromechanical transmission

As the speed of output shaft on the inner track, and thus the gear ratio, in the process of switching off the brake and switcing on the clutch, is the function of hydraulic gear ratio in HDM and slipping of the clutch and brake frictional elements, relations for transmission ratios and relative turning radius will be made, but just for the end of the second phase, when the clutch for turning is completely switched on. In this case, gear ratios and relative turning radius are only the functions of hidraulic gear ratio.

**The gear ratio of the output shaft on the inner track** is derived from the kinematic relation for SPP, which has the following form

$$n_{au} + \alpha \cdot n_{bu} - (1+\alpha) \cdot n_n = 0 \tag{1}$$

where

$n_{au}$ – speed of sun gear

$n_{bu}$ – speed of ring gear,

$n_n$ – speed of carrier and

$\alpha$ – inside characteristic.

The inside characteristic of SPP is defined by the following expression

$$\alpha = \frac{Z_b}{Z_a} \tag{2}$$

where $Z_a$ and $Z_b$ are numbers of teeth of sun gear and ring gear.

The speeds of sun gear and the carrier are given in the following expressions

$$\begin{aligned} n_a &= n_{au} = \frac{n_M}{i_{pp}} \\ n_n &= \frac{n_M}{i_m \cdot i_{HMP}} \end{aligned} \tag{3}$$

where

$n_M$ - speed of input (drive) shaft,

$i_m$ - gear ratio in gearbox,

$i_{pp}$ - gear ratio in auxiliary drive and

$i_{HMP}$ - gear ratio in hydromechanical gear train (HMP), consisting of HDM and DP.

The relations for gear ratios in the gearbox, auxiliary drive and HMP are the following

$$\begin{aligned} i_{mI} &= \frac{Z_{3M}}{Z_{1M}} \\ i_{pp} &= \frac{Z_{4p}}{Z_{1p}} \cdot \frac{Z_{2s}}{Z_{1s}} \\ i_{HMP} &= \frac{1+\alpha_D}{1+\alpha_D \cdot i_H} \end{aligned} \tag{4}$$

where

$Z_{1M}$, $Z_{3M}$ – numbers of gear teeth in gearbox,

$Z_{1p}$, $Z_{4p}$ – numbers of gear teeth in auxiliary drive,

$Z_{1S}$, $Z_{2S}$ – numbers of gear teeth from the S clutch to SPP,

$\alpha_D$ – inside characteristic of DP and

$i_H$ – hydraulic gear ratio of HDM.

The inside characteristic of DP and the hydraulic gear ratio of HDM are defined by relations

$$\begin{aligned} \alpha_D &= \frac{Z_{bD}}{Z_{aD}} \\ i_H &= \frac{n_T}{n_P} \end{aligned} \tag{5}$$

where

$Z_{aD}$, $Z_{bD}$ — numbers of sun gear teeth and ring gear teeth and

$n_P$, $n_T$ — speed of HDM pump and turbine wheel.

The gear ratio of output shaft on the inner track is derived from relations (3), (4) and (1)

$$i_u = \frac{n_M}{n_{bu}} = \frac{\alpha}{\dfrac{1+\alpha}{i_m} \cdot \dfrac{1+\alpha_D \cdot i_H}{1+\alpha_D} - \dfrac{1}{i_{pp}}} \tag{6}$$

For numerical values: $\alpha$=$\alpha_D$=2.5455, $i_{mI}$=1 and $i_{pp}$=0.4783, the gear ratio is only the function of hydraulic gear ratio ($i_H$)

$$i_u = \frac{1}{i_H - 0.4285} \tag{6a}$$

**The gear ratio of the output shaft on the outer track** is obtained from the kinematic equation for SPP, in the following form

$$n_{as} + \alpha \cdot n_{bs} - (1+\alpha) \cdot n_n = 0 \tag{7}$$

where $n_{as}$ and $n_{bs}$ are speeds of both the sun gear and the ring gear.

The relation for the carrier speed is the same, and the speed of SPP sun gear is

$$n_{as} = 0 \tag{8}$$

From relations (8), (3) and (7), the gear ratio of the output shaft on the outer track is obtained

$$i_s = \frac{n_M}{n_{bs}} = \frac{1+\alpha_D}{1+\alpha_D \cdot i_H} \cdot i_m \cdot \frac{\alpha}{1+\alpha} \tag{9}$$

For numerical values of parameters: $\alpha$, $\alpha_D$ and $i_m$, the gear ratio is the following

$$i_s = \frac{1}{i_H + 0.3929} \tag{9a}$$

The dependence of transmission ratio of the output shafts in the function of hydraulic gear ratio of HDM, obtained from relations (6a) and (9a), is presented in Fig.4.

The mutual dependence of output shaft speeds obtained from basic equations for SPP on the inner and outer track is given in the form

$$n_{bs} = n_{bu} + \frac{n_M}{i_{pp} \cdot \alpha} = n_{bu} + 0.8214 \cdot n_M \tag{10}$$

**The relative turning radius.** The relation which is often used for the relative turning radius instaed of the relation for the vehicle turning radius is defined by the following relation

$$\rho = \frac{R}{0.5 \cdot B} = \frac{V_s + V_u}{V_s - V_u} = \frac{n_{bs} + n_{bu}}{n_{bs} - n_{bu}} \tag{11}$$

where

$R$ – turning radius of vehicle centre,

$B$ – track trails width and

$V_s$, $V_u$ – circumferential speed of the outer track and the inner track, respectively.

Based on relations (6) and (9) for output shaft speeds the relative turning radius is determined as follows

$$\rho = \frac{1 + \alpha_D \cdot i_H}{1 + \alpha_D} \cdot \frac{1 + \alpha}{\alpha} \cdot \frac{2 \cdot i_{pp} \cdot \alpha}{i_m} - 1 \tag{12}$$

For numerical values of parameters $\alpha$, $\alpha_D$, $i_{pp}$ and $i_m$, the relation for relative turning radius is the following

$$\rho = 2.4348 \cdot i_H - 0.0435 \tag{12a}$$

Based on relation (12a), it can be concluded that the relative turning radius is the function of hydraulic gear ratio in HDM. As the hydraulic gear ratio change is continual, depending on outside resistances, the relative turning radius change is also continual.

## Hydromechanical transmission testing

The objective laboratory testings of the functional transmission model was also to determine the gear ratio character change, or relative turning radius, by the test stand simulation of appropriate conditions.

During these tests, the operating conditions of transmissions were similar to those that appear in the real utilization. The testings were carried out with some simplifications because the achievement of utilization conditions in a laboratory was impeded.

A lot of parameters were measured during the testings. The recorded values of speed on the transmission input ($n_M$) and output ($n_{bu}$ and $n_{bs}$) were important for the determination of gear ratio and relative turning radius.

*Test stand*

The testing of transmission was carried out on the test stand the scheme of which, with measured points is presented in Fig.2. The transmission was driven by the electric motor (EM) with continual changes of speed. The power was transmited from the electric motor via torsion dynamometer (TDU) and propeller shaft (KV) to the hidromechanical transmission (HMT). The output shafts of transmission are connected with generators (G), used as electrical brakes, via middle shaft (MV) and appropriate torsional dynamometers (TDL and TDD).

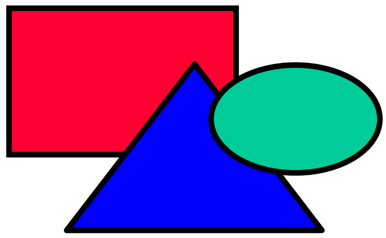

**Figure 2.** Scheme of test stand for transmission testings

In the course of transmission testings on the test stand, it was necessary to confirm the continual change of turning radius during the vehicle turning simulation. The simulation of transmission functioning during the turn was carried out for the two cases:

- With a changeable load on the transmission output shafts and a constant speed of the drive shaft. The realization of some turning radius values is simulated by these tests.
- With a constant load on the transmission output shafts and a changeable speed of the drive shaft, which simulates the starting of the vehicle.

*Simulation of transmission function with changeable load*

During the simulation of transmission function in this case, the drive shaft speed was constant and the load changed on the inside output shaft. After the achievement of stationary values, speed values on the driving ($n_M$) and output ($n_{bu}$ and $n_{bs}$) transmission shafts were measured, Fig.3.

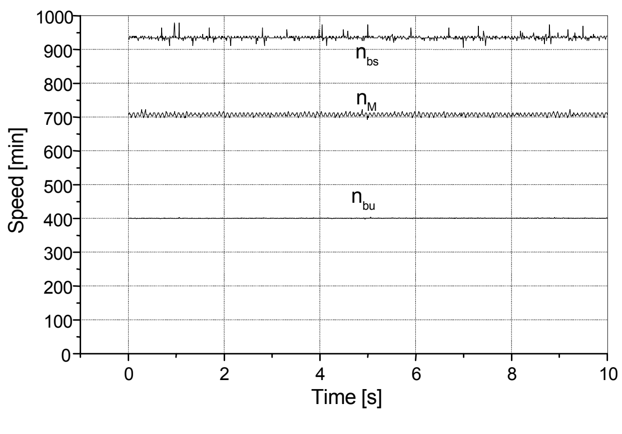


**Figure 3.** Transmission speeds: $n_M$– driving shaft, $n_{bu}$– inside output shaft $n_{bs}$– outside output shaft

Fig.4 shows charts of gear ratio changes on output shafts of the inner ($i_u$) and outer ($i_s$) track, obtained by relations (6a) and (9a), and values of gear ratios calculated on the basis of measured speed values of the input and the output shaft. Appropriate values of hydraulical gear ratios in HDM were calculated by relation (9a).

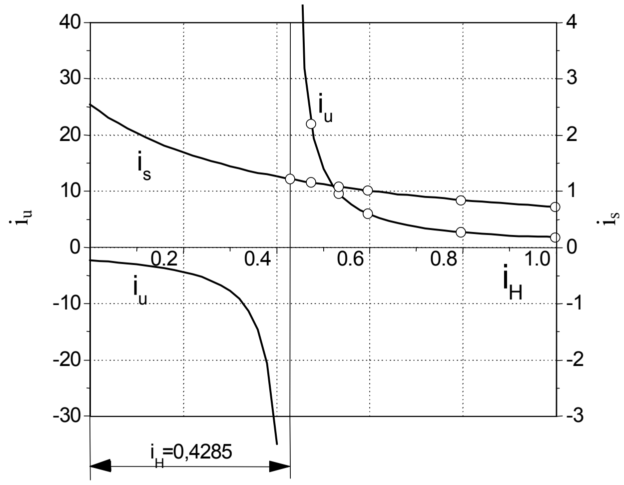


**Figure 4.** Gear ratios: $i_u$ – inside output shaft, $i_s$ – outside output shaft

Figure shows that gear values ratio obtained by measured speed (marked by points on curves) are the same as the values obtained by calculations. The character of their changes is continual.

The stopping of the output shaft from the inner track (turning around the track) was achieved with hydraulical gear ratios higher than zero. This is not convenient because the transformation torque coefficient in HDM is lower (it is the highest for $i_H$=0) and produces lower attainable torque on the output shafts for turning. However, with the appropriate selection of gear ratios in the auxiliary drive, the turning around the track with the maximum value of torque transformation coefficient can be achieved.

Also, it can be concluded that for some values of hydraulic gear ratio (0≤ $i_H$ ≤0.4285) the gear ratio on the inner track is negative, so the output shaft rotates in the opposite direction to the real motion.

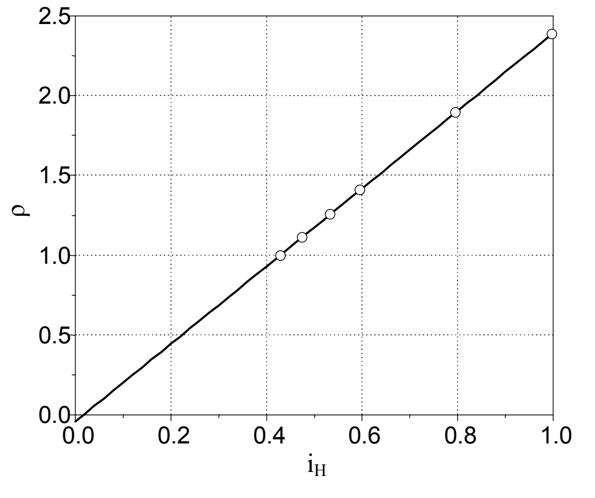


**Figure 5.** Relative turning radius

Fig.5 shows the chart of relative turning radius change obtained from relation (12a). The values of turning radius calculated with relation (11) and speed values measured on the output shafts are presented by points on the same chart. Measured values of relative turning radius are the same as the values obtained by relation (12a). Their change is continual.

The character of turning radius change can be concluded on the basis of charts in Fig.6 where the change of speed during the severe load increase on the output shaft of the inner track are showen, simulating the case when the track comes across an obstacle.

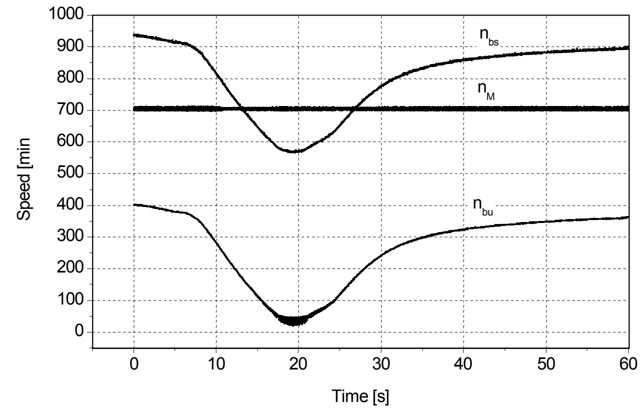


**Figure 6.** Transmisson speed during the load increase on output shafts: $n_M$– driving shaft, $n_{bu}$ – inside output shaft, $n_{bs}$ – outside output shaft

It can also be seen that the speeds of both output shafts decrease if the load of one output shaft increases. With the output shaft unloaded, the speed on both output shafts increases. During the output shaft loading and unloading, the difference in speed of the output shafts is constant and has the same value as during the stationary regime (constant values of the speed on the output shafts). This speed difference is constant due to the elastic connection of the output shafts over HDM and in this case it is defined by relation $\Delta n$=0.8214$n_M$. The speed of the input shaft was not changed during these testing.

As seen, the speed of the output shafts, and thus the relative turning radius, depends on outside resistances and change continually and automatically, so the frequent change of direction is necessary, which is considered as a great disadvantage of these transmissions.

*Simulation of transmission function with constant load*

During this simulation of transmission function during the turn, the change of the drive shaft speed with a constant load on the output shafts was carried out. Fig.7 shows changes of speed on the input and the output shaft during the drive shaft acceleration (situation that simulates the starting of the vehicle).

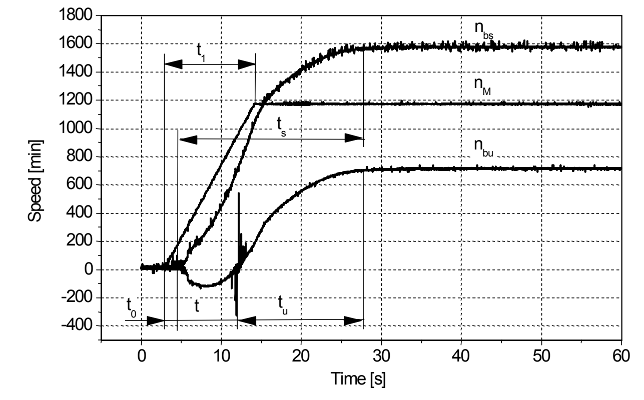


**Figure 7.** Speed during the acceleration of driving shaft: $n_M$– driving shaft, $n_{bu}$ – inside output shaft, $n_{bs}$ – outside output shaft

The transmission testing in such a regime was carried out by switching on the clutches for turning, while the drive shaft was stopped ($n_M$=0). When the drive shaft starts to rotate, since HDM is not filled up with oil, and because of the slipping of HDM working units, the beginning of speed change is delayed for "$t_0$" (Fig.7). The increase of the output shaft speed on the outer track is carried out for time "$t_s$", and the speed increase of the output shaft on the inner track develops in two phases. In the first phase, with the time interval "$t$" (Fig.4), it rotates in the opposite direction to the vehicle movement, which corresponds to the values of hydraulical gear ratio $0 \le i_H \le 0.4285$, presented in Fig.4. The stopping of the output shaft on the inner track occurs when $i_H$=0.4285 and the second phase, during which this shaft accelerates in the direction of vehicle motion $i_H$>0.4285 starts. When the constant speed of the input shaft is achieved, the output shafts, under the influence of rotational mass, still accelerate so their constant speed achieved a little bit later as seen on the presented chart, Fig.4.

The acceleration time of the drive shaft is "$t_1$". The driving shaft acceleration starts from rest position due to the fact that stand design does not contain the clutch between the engine and the transmission. If there were a clutch, the drive could be switched on during the engine functioning. Therefore, the acceleration time of the drive shaft would be shorter and the acceleration times of output shatfs would be shorter too.

The rotation of the output shaft on the inner track in the opposite direction occurs under the influence of chosen gear ratio in the auxiliary drive and it occurs only during testings on the test stand under the influence of drive shaft acceleration, because in that case the value of hydraulic gear ratio changes from $0 \le i_H \le 1$. In the case of transmission testing with a constant speed of drive shaft, and a changeable load, during the output shaft braking on the inner track, it stops when $i_H$ = 0.4285. It is the minimum value of hydraulic gear ratio that can be obtained during the turn, Fig.4.

On the basis of the performed analysis of laboratory test results obtained by the simulation of transmission functioning during the turn it can be concluded that the functional transmission model completely satisfies the function of vehicle turning with the continual change of radius.

## Conclusions

The continual change of transmission ratio during rectilinear motion and turning is obtained with the concept of hydromechanical transmission which contains HDM as a component for rectilinear motion and turning.

The change of relative radius turning is continual and automatic. The automatic change of relative turning radius, depending on external resistances, demands frequent corrections of vehicle direction, which presents one of disadvantages of tis HMT concept.

It is recommended to choose such value of gear ratio in the accessory drive in order to provide the turning around the track when the hydraulic ratio is $i_H \approx 0$. In that case, the torque transformation coefficient in HDM has the maximum value and therefore the available torque on the output shafts of transmission is maximal.

Besides some simplifications which were necessary during the testing of functional transmission model, the results of research allow more complete overview of the problems related to this transmission concept and show the direction of further optimizations, as well as the procedures which are to be performed during the design phase in order to obtain better vehicle performances.

## References


[1] MILIĆ MIODRAG. *Use of hydrodynamical gearbox as component of hydromechanical transmission of tracked vehicles.* Mechanical Faculty, Belgrade, 2002 MSc. thesis (in Serbian)

[2] DIMITRIJE JANKOVIC. *Theory of motor vehicle moving.* Mechanical Faculty, Belgrade (in Serbian).

[3] PROKOFEV,V.N. *Gidravlicheskiye peredachi kolesnyh i gusenichnyh mashin*. Moskva 1960.

[4] PETROV,A.V. *Planetarnye i gidromehanicheskiye peredachi kolesnyh i gusenichnyh mashin*. Mashinostroyeniye, Moskva, 1966.


Received: 30.10.2003

# Identifikacija ponašanja hidromehaničke transmisije u uslovima zaokreta brzohodnog guseničnog vozila

**U radu je analiziran zaokret brzohodnog guseničnog vozila sa hidromehaničkom transmisijom kod koje se hidrodinamički menjač koristi za ostvarivanje pravolinijskog kretanja i za zaokret vozila. Prikazani su rezultati laboratorijskih ispitivanja realizovanog funkcionalnog modela transmisije dobijeni simuliranjem rada transmisije, u procesu zaokreta vozila, na probnom stolu.**

*Ključne reči:* **brzohodno gusenično vozilo, hidromehanička transmisija, hidrodinamički menjač, zaokret, ispitivnaje.**

# Identification du comportement de la transmission hydromécanique pendant le virage du véhicule chenillé à grande vitesse

**On a analysé le virage du véhicule chenillé à grande vitesse à l'aide de la transmission hydromécanique où la boîte à vitesses hydrodynamique est utilisée pour réaliser le mouvement rectiligne et le virage du véhicule. On a présenté les résultats des essais en laboratoire d'un modèle de transmission opérationnel et réalisé par la simulation de la transmission pendant le virage sur le banc d'essai.**

*Mots-clés*: **véhicule chenillé à grande vitesse, transmission hydromécanique, boîte de vitesses hydrodynamique, virage, essai.**