HYDROSTATIC DRIVER FOR TOOL CARRIES MT8

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Abstract
The article introduces the tool carrier and justifies the need to solve its drive by a hydrostatic transmission. Subsequently, a block diagram for the hydrostatic transmission and its control is described in the paper. The article describes the components for hydrostatic transmission which are supplemented by charts of calculated operating characteristics of the hydro generator and the hydraulic motors. In conclusion, the article deals with the calculation of the traction force characteristics on the machine wheels and the calculation of gear ratios and hydrostatic transmission ratios.

Key words: *Hydrostatic transmission, traction force, hydro motor, hydro generator*

INTRODUCTION

Tool carriers are mobile machines which are developed for work mostly in sloping mountain and underground terrain. They can work in slopes up to 40 degrees. They are mainly driven by diesel engines. These machines have a wide track and a position of center of gravity is not too high for better stability on the slopes. Their advantage is the possibility of aggregating of various devices from grasses to low or high grasslands, through pickers and manipulators to sweepers and tanks. Attachments are attached to the front or rear by a three-point hinge and they can be driven by an output shaft.

The reason for the new concept of the drive is to replace the components of the mechanical drive and we will achieve the increase in the moral value, the competitiveness of the product and increasing of the machine variability.
The basic requirements were 35 degrees slope accessibility, two driving modes - road / work, 4x4 switchable 4x2, maximum speed 25 km / h, working 12 km / h and use of hinged hydraulic motors in particular. (Hrček S. & Bucala J., 2014)

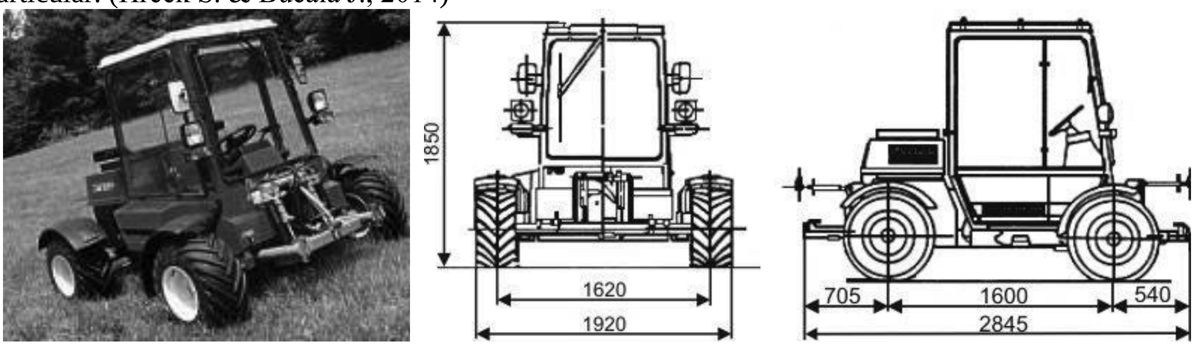

Fig. 1 Tool carries MT 8-222 and its basic dimensions

MATERIALS AND METHODS

Block scheme of hydrostatic drive with control

The Fig. 2 shows block diagram which was prepared from the requirements. All basic mechanical, hydraulic and electronic components of the hydrostatic drive of the wheels and their steering and their interrelationships are shown in this scheme. (Kohár R., Brumerčík F., Lukáč M. & Nieoczym A., 2016) The black continuous line shows the power line of the hydrostatic transmission. The black interrupted line shows hydrostatic overflow line. The blue continuous line is the control circuit. The blue interrupted line is the overhead line of the control circuit and the green continuous line is electric wiring, which connects control and actuator components of the drive. (Kohár R., 2016), (Lukáč M., Brumerčík F. & Krzywonos L., 2016)

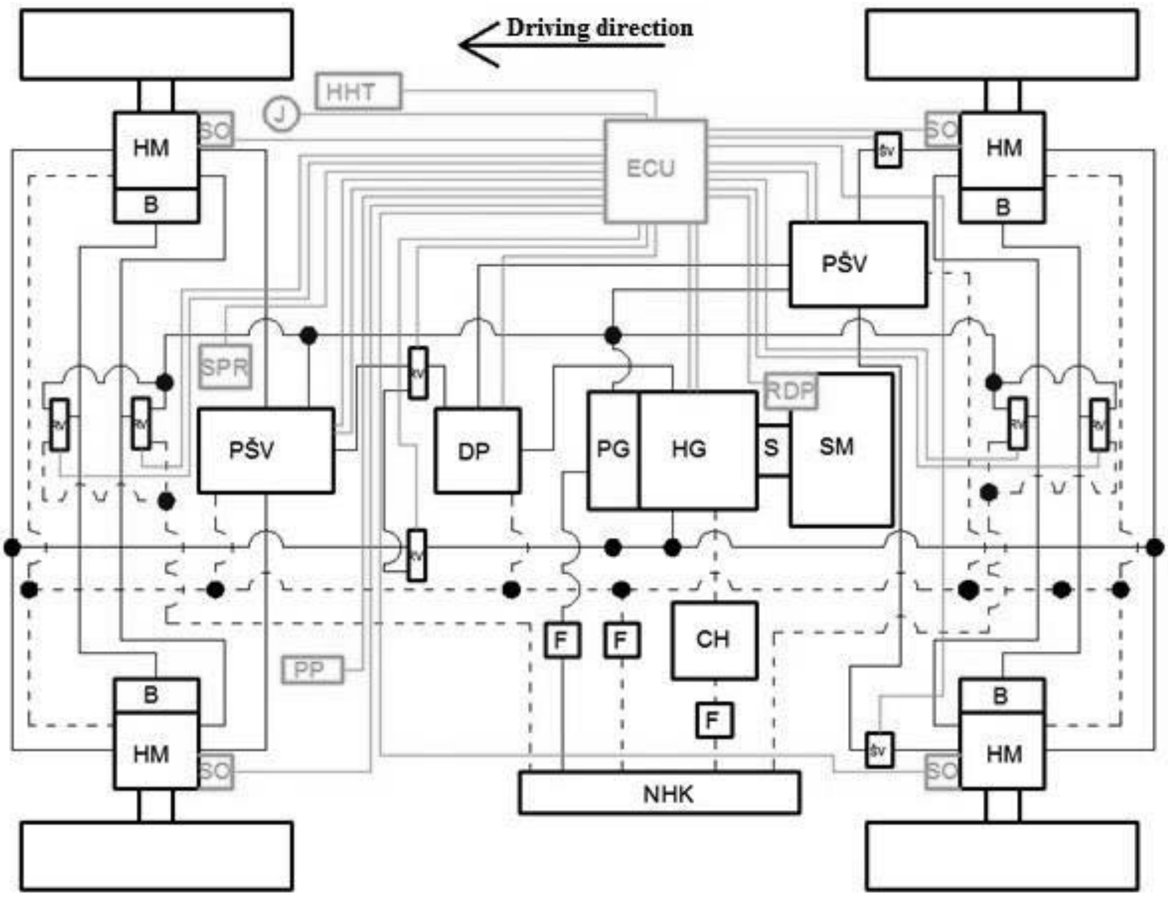

Fig. 2 Block diagram of the hydrostatic drive of the drive wheels with electronic control.

Tab. 1 Explanation of parts, which are used in block scheme

Hydraulic parts		**Mechanical parts**		**Electric parts**	
DP	Flow divider	B	Brake	ECU	Electronic control unit
F	Filter	S	Clutch	HHT	Control terminal
HG	Hydro generator	SM	Engine	J	Joystick
HM	Hydro motor			PP	Acceleration / deceleration pedal
CH	Cooler			RDP	Fuel Dosing Controller of SM
NHK	Hydraulic fluid reservoir			SO	Speed sensor
PŠV	Proportional throttle valve			SPR	Steering position sensor
RV	Control valve				
ŠV	Throttle valve				
PG	Additional hydro generator				

Components and projective parameters of hydrostatic transmission

The power unit is straight-three engine with 1649 cm^3 displacement with maximum power P = 25.4 kW at 3000 rpm and a maximum torque of 94 Nm at 1700 rpm. Tires size- 31x15,5-15 have a static radius 350 mm and the effective circumference 2235 mm. The calculation of the tractive force of the machine at its maximum required weight of 2000 kg, depending on the coefficient of engagement μ on the different surface types, was made before the design of the hydrostatic transmission. (Tropp M., Lukáč M., Nieoczym A. & Brumerčík F.,2016)

The maximum of tractive force of machine is calculated by equation 1:

$$F_{T1max} = m_{max}.g.\mu = 2000.9{,}81.0{,}9 = 17658\ (N) \qquad (1)$$

Subsequently, calculations of rolling resistance, climbing and total loss force at the 35 degrees climbing angle and various types of fieldwork were also done. The maximum traction force required was 17700 N which is based on these calculations. (Kučera Ľ. & Gajdošík T., 2014)
The calculation of the corner power of hydro generator for driving mode – work (2) and driving mode-road (3) was performed in the 4x4 drive was performed from the required tractive force.

$$P_{R1} = \frac{F_{T1max} \cdot v_{1max}}{3600} = \frac{17700.12}{3600} = 59\ (kW) \tag{2}$$

$$P_{R2} = \frac{F_{T2max} \cdot v_{2max}}{3600} = \frac{8500.25}{3600} = 59{,}027\ (kW) \tag{3}$$

The total transmission range with hydrostatic transmission efficiency $\eta_{HSP} = 0{,}85$ will be:

$$R_P = \frac{P_{R1}}{P_{SMmax} \cdot \eta_{HSP}} = \frac{59}{25{,}4.0{,}85} = 2{,}732 \tag{4}$$

On the basis of the corner power calculation, an axial piston axial piston hydro generator was selected and used with a maximum displacement volume of 40 cm3 / rev, a theoretical flow of 144 l / min at 3600 rpm, a theoretical output of 76.8 kW at a pressure difference of 32 MPa, a torque of 63,7 Nm at a pressure difference of 10 MPa. The minimum system pressure is 1.5 MPa and the maximum working pressure is 35 MPa. (Kučera Ľ., Gajdošík T. & Bucala J., 2014)
From the known values of the hydro generator and the combustion engine was made a graph of the hydro generator power which is dependent on the torque of the combustion engine and the angle of the inclined plate of the hydro generator (Fig. 3) and then the graph of the complete characteristic of the hydro generator (Fig.4) was made also.

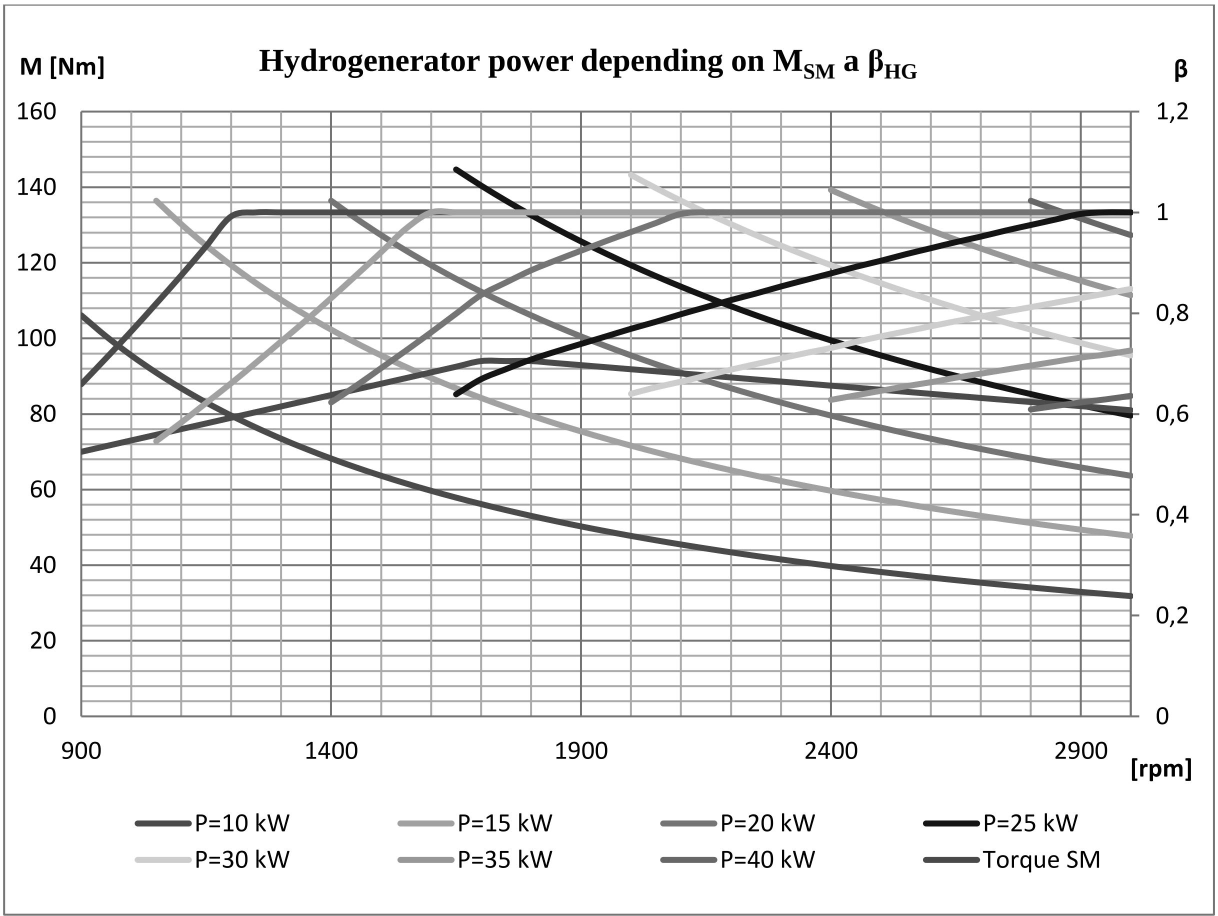

Fig. 3 Hydrogenerator power is depending on M_{SM} a β_{HG}

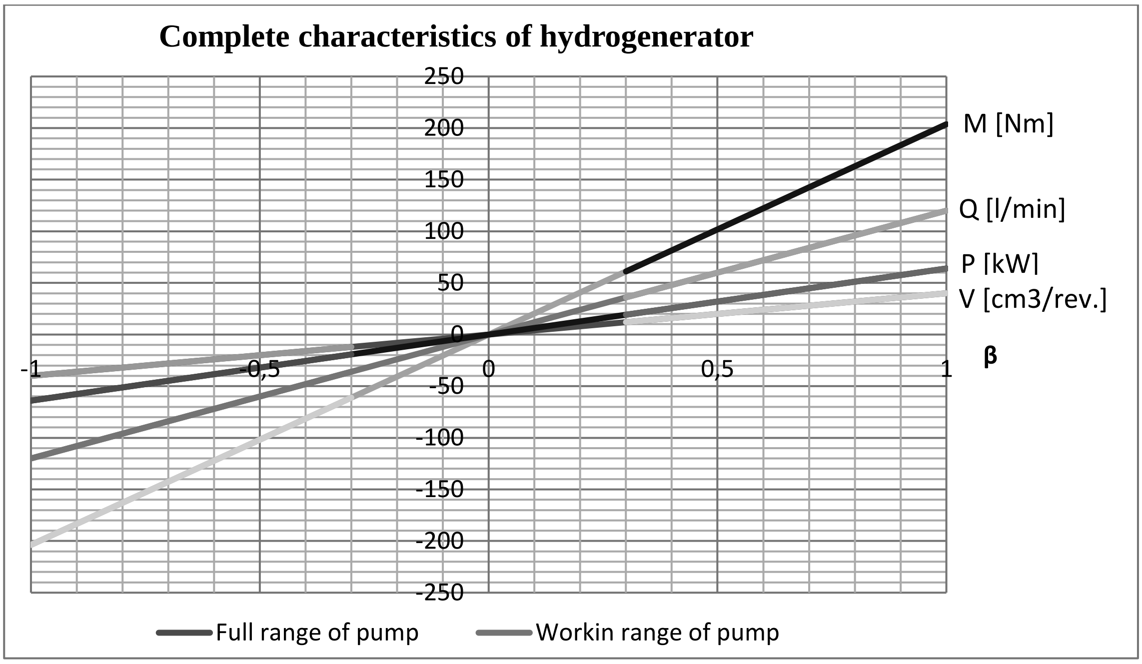

Fig. 4 Complete characteristic of hydro generator

The 2-displacement motors with brake and with placement in swivel joints were used as steerable wheel motors. The displacement is 322/166 cm^3/rev., maximum power is 22 kW, maximum speed id 250/275 rpm and maximum pressure is 40 MPa. The graphs in FIG. 5 and 6 show the dependence of the torque of the hydraulic motor on speed. From the graphs it is also possible to calculate the flow rate at the given speed and the given slope of the inclined plate of the hydro generator and the corresponding pressure in the system.

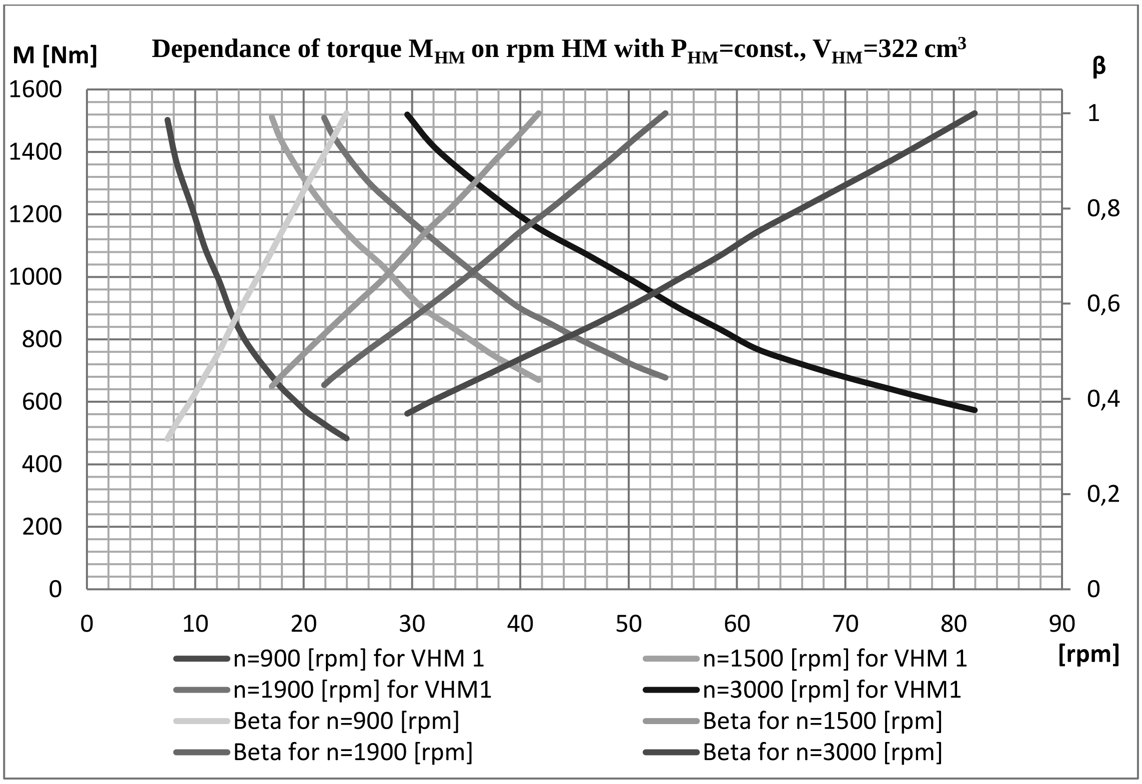

Fig. 5 Dependence of torque M_{HM} on rpm with power= constant, displacement= 322 cm^3

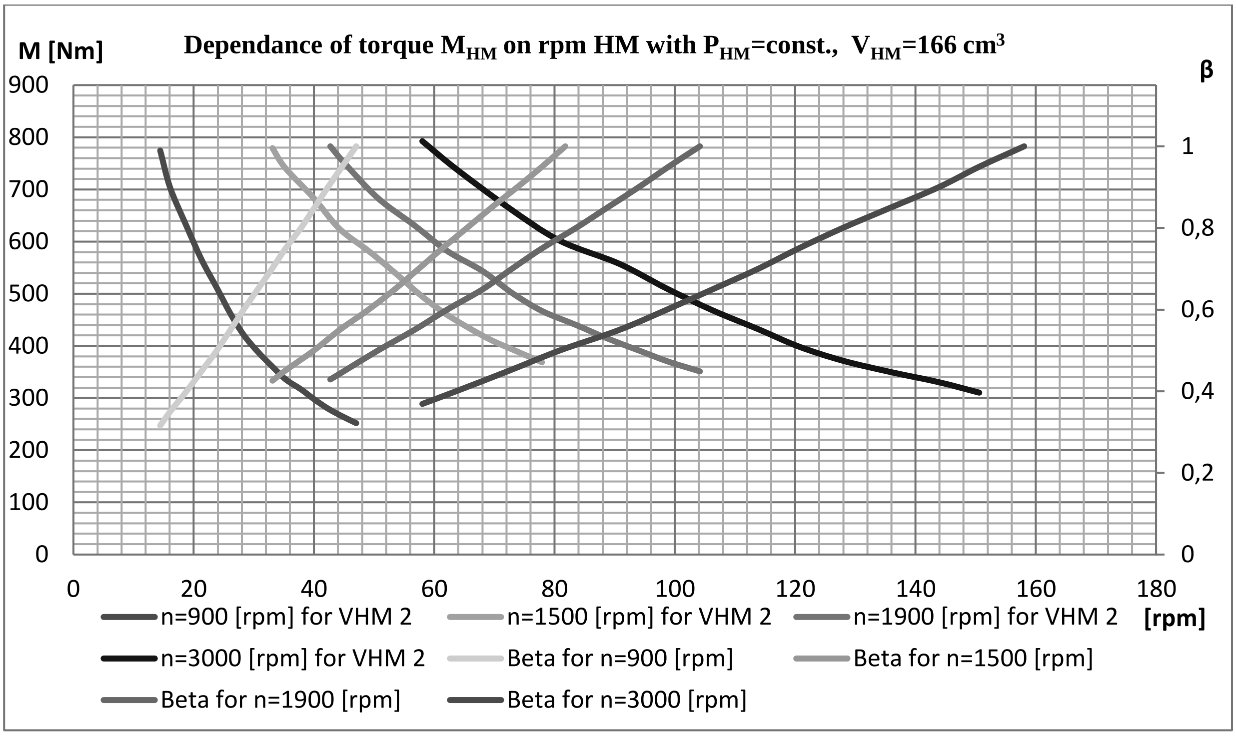

Fig. 6 Dependence of M_{HM} torque on rpm with power= constant, displacement= 166 cm^3

RESULTS AND DISCUSSION

Tractive force characteristics

In this part of paper is description of calculation of traction parameters for driving mode- work, especifically for the A1 point in graph (Fig. 7). (Kučera Ľ. & Gajdošík T., 2013)

- The calculation of the minimum value of regulatory parameter- β_{HGmin} of hydro generator at the maximum possible speeds (n_{HGmax}=3000 rpm and torque M_{sm}=81 Nm) :

$$\beta_{HG} = \frac{M_{SM}.20.\pi.\eta_{mech.HG3000}}{\Delta p.V_{HGmax}} = \frac{81.20.\pi.0,927}{320.40} = 0,369 \qquad (5)$$

- The calculation of displacement of the hydro generator at the maximum engine speed.

$$Q_{HGmaxA3000} = \frac{V_{HGmax}\cdot\beta_{HGmin}\cdot n_{HG}\cdot\eta_{QHG3000}}{1000} = \frac{40.0,369.3000.0,927}{1000} = 41,063\ (l/min) \qquad (6)$$

- Displacement from the hydro generator is divided between four hydro motors. The calculation of hydraulic motor speed at the maximum hydro generator speed and also the first working displacement of the hydraulic motor (V_{HM}=0,322 l/rev.) :

$$n_{HM2maxA23000} = \frac{Q_{HMmaxA3000}\cdot\eta_{QHM3000}}{V_{HM2max}} = \frac{10,265.0,927}{0,322} = 29,565\ (ot/min) \qquad (7)$$

- The calculation of machine speed at the direct ride on plane:

$$v_{1maxA13000} = \frac{RC.n_{HM1maxA13000}\cdot 60}{1000} = \frac{2,253.29,565.60}{1000} = 3,996\ (km/h) \qquad (8)$$

- The calculation of torque and tractive force at the one wheel:

$$M_{HM1maxA13000} = \frac{\Delta p.V_{HM1max}\cdot\eta_{mech.HMV13000}}{20.\pi} = \frac{320.322.0,927}{20.\pi} = 1519,682\ (Nm) \qquad (9)$$

$$F_{T1maxA13000} = \frac{M_{HM1maxA13000}}{SR} = \frac{1519,682}{0,35} = 4341,951\ (N) \qquad (10)$$

- The calculation of total tractive force:

$$F_{T\_total} = 4.F_{T1maxA13000} = 4.4341{,}951 = 17367{,}804\ (N)$$

All the working points of the diagrams in Fig. 7 and Fig. 8 were calculated by the same process.

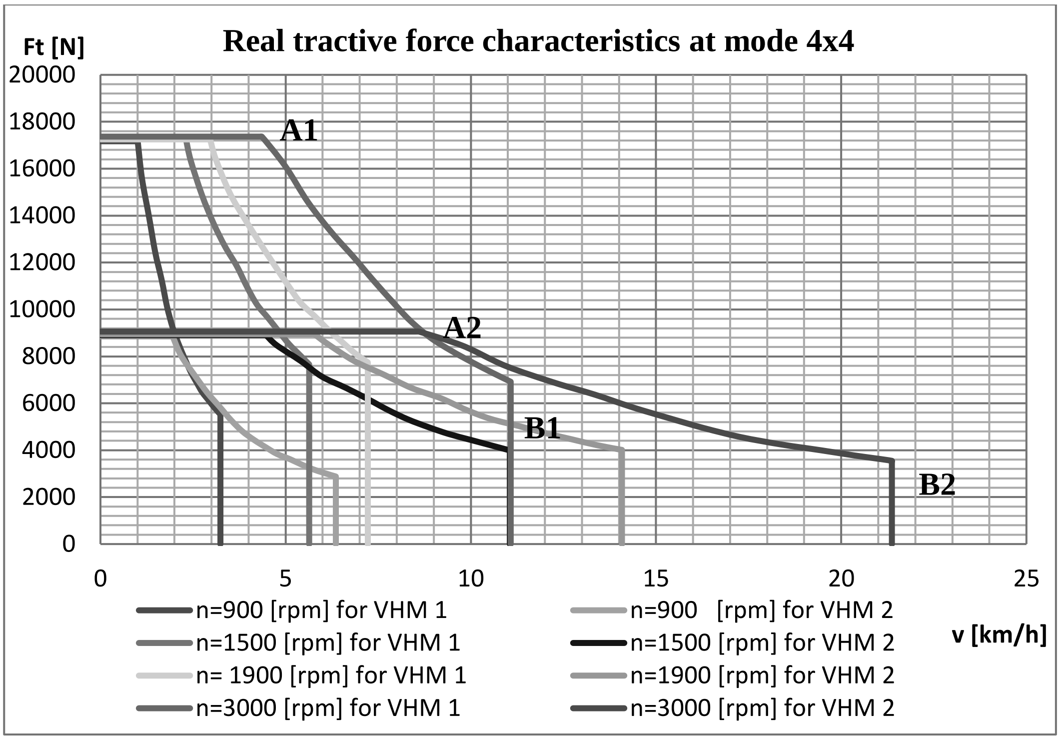

Fig. 7 Real tractive force characteristics at mode- 4x4

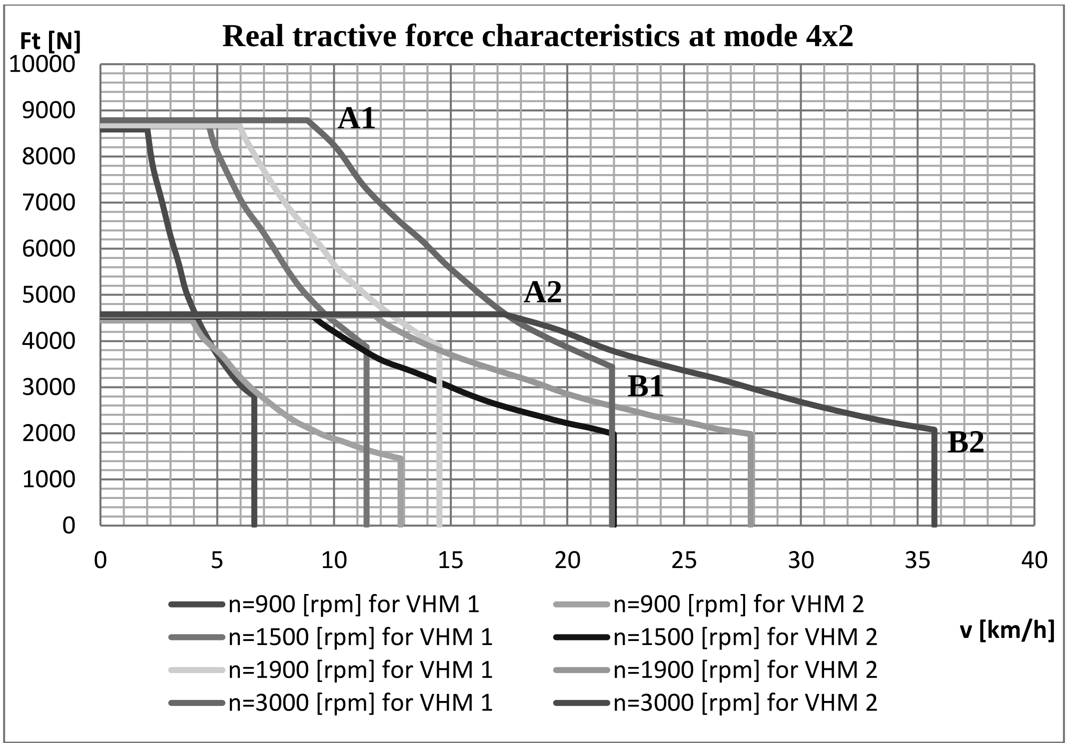

Fig. 8 Real tractive force characteristics at mode- 4x2

Transmission ratios:

- The calculations of the maximum and minimum kinematic ratio at the first displacement of the hydraulic motor (V_{HM1}):

$$i_{h1max} = \frac{4.V_{HM1max}}{V_{HGmax} \cdot \beta_{HGmin} \cdot \eta_{QHM} \cdot \eta_{QHG}} = \frac{4.0,322}{0,04.0,369.0,927.0,927} = 101,547 \quad (11)$$

$$\mathrm{i}_{\mathrm{h1min}} = \frac{4.\mathrm{V}_{\mathrm{HM1max}}}{\mathrm{V}_{\mathrm{HGmax}} \cdot \beta_{\mathrm{HGmax}} \cdot \eta_{\mathrm{QHM82}} \cdot \eta_{\mathrm{QHG3000}}} = \frac{4.0,322}{0,04.1.0,953.0,921} = 36,686 \quad (12)$$

- The kinematic range of the transmission system at the first working displacement of the hydraulic motor is:

$$R_{K1} = \frac{i_{h1max}}{i_{h1min}} = \frac{101,547}{36,686} = 2,768 \quad (13)$$

- The calculation of maximum and minimum torque ratios for the first working displacement of the hydraulic motor:

$$\bar{\iota}_{h1max} = \frac{4.V_{HM1max} \cdot \eta_{mech.HM29} \cdot \eta_{mech.HG3000}}{\beta_{HGmin} \cdot V_{HGmax}} = \frac{4.0,322.0,927.0,927}{0,369.0,04} = 74,987 \quad (14)$$

$$\bar{\iota}_{h1min} = \frac{4.V_{HM1max} \cdot \eta_{mech.HM82} \cdot \eta_{mech.HG3000}}{\beta_{HGmax} \cdot V_{HGmax}} = \frac{4.0,322.0,953.0,921}{1.0,04} = 28,262 \quad (15)$$

- The calculation of torque gear range:

$$R_{M1} = \frac{\bar{\iota}_{h1max}}{\bar{\iota}_{h1min}} = \frac{74,987}{28,262} = 2,653 \quad (16)$$

The same process was used to calculate the maximum and minimum kinematic and torque transmission ratio of the drive with the second working displacement of the hydraulic motors.

Hydrostatic drive control system

The control system provides complete control over wheel speed control and forward / reverse steering. It provides the option of selecting driving modes via the hand-held terminal in the operator's cab. The machine operator selects only the type of driving mode, the travel direction (forward / reverse) and the driving speed using the accelerator pedal. Speed sensors are built into hydraulic motors measure the speed of rotation of each driven wheel, continuously. The control unit compares these speeds and reduces the flow to this wheel (via the control valve) if it is necessary (increasing the speed of one-wheel relative to others - slipping) until the wheel speed is again balanced. The system also checks the position of the sloping HG plate and controls the fuel dose for engine which is depending on the load. (Tomášiková M., Tropp M., Krzysiak Z. & Brumerčík F., 2015)

CONCLUSIONS

The article describes the design calculation of a hydrostatic drive for a special working machine working on slopes. The article also describes the selection of the basic components of the hydrostatic drive. After designing the components of the hydrostatic drive are made a few process: process for calculating of the drive parameters and also the process of calculating the machine stroke parameters which is projected into graphs. The final calculation of kinematic and torque transmission ratios and ranges is described at the end of the article. The benefits of this are the unconventional solution of the wheel drive system, the removal of the morally obsolete mechanical wheel drive, the increase in the possibility of variability, arrangement and axle concepts.

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