VALVE CONTROL OF DRIVE WITH ROTARY HYDRAULIC MOTOR

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The paper describes the analysis of a valve control of a drive with a rotary hydraulic motor. The valve control is realized by a throttle valve. The individual drive characteristics are measured on the experimental device, i.e. the control and the torque-speed characteristics. The experimentally determined characteristics are compared to the theoretically calculated characteristics. Subsequently, the efficiencies of the tested rotary hydraulic motor are experimentally determined, i.e. the volumetric, mechanical-hydraulic and total efficiencies. The theoretical computational equations are subsequently supplemented by the experimentally determined efficiencies of the hydraulic motor. Theoretically determined characteristics of the drive, including efficiencies of the hydraulic motor, are again compared to the experimentally determined characteristics. The mathematical model of the experimental device is also assembled. This mathematical model is verified in a steady state.

KEYWORDS

drive characteristics, efficiencies of rotary hydraulic motor, valve control, mathematical model of drive, hydraulics.

1 INTRODUCTION

The output member, which is usually controlled at hydrostatic drives, is a hydraulic motor. There are rotary or linear hydraulic motors. Output quantities, which can be controlled at these motors, are position and its derivation over time and force (torque). There are two main control methods in the case of the hydrostatic drives, i.e. volume and valve control. In the case of the volume drive control, the geometric displacement volume of the hydrostatic pump or the rotary hydraulic motor is changed. In the case of the valve drive control, flow rate or pressure are changed by means of a valve. This paper deals with the valve control of the output speed of the hydrostatic drive. There are many possibilities of the valve control of the speed of the rotary hydraulic motor. It is necessary to primarily control the flow rate in order to control speeds. For this reason, this paper is focused on the flow control valves. From a qualitative point of view, these valves are divided into classical valves (conventional), e. g. [Himr 2017], [Kozdera 2014], proportional valves or servo valves, e. g. [Luan 2018]. The speed control problem of the hydrostatic drive can be most clearly described on a classical flow control valve, in this case on the throttle valve.

2 THEORETICAL BACKGROUND

A model example of the hydrostatic drive with the rotary hydraulic motor is shown in Fig. 1. Efficiencies of the hydraulic pump HP and the hydraulic motor HM, pressure losses in pipes and pressure dependence of the pressure relief valve RV are neglected in this basic theoretical analysis. Speeds of the rotary hydraulic motor HM are controlled by the throttle valve TV. The pressure is realized by means of the unidirectional fixed displacement hydraulic pump HP and the pressure relief valve RV. Furthermore, the hydraulic system consists of the tank T.

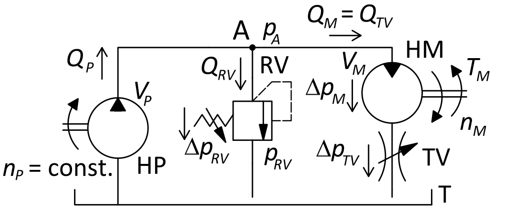

Figure 1. Schematic diagram of hydrostatic drive with rotary hydraulic motor

The speed control method of the hydraulic motor HM by means of the throttle valve is based on the assumption of flow rate dividing. The pump HP delivers the constant flow rate Q_P at the constant speed n_P and the constant geometric displacement volume V_P. If the working liquid doesn´t flow through the pressure relief valve RV, the motor speed n_M is not controlled. The pressure relief valve RV is open at the moment when the working liquid pressure at the point A (see Fig. 1) reaches the pressure value, which is adjusted by the pressure relief valve RV, i.e. $p_A = p_{RV}$ (see Fig. 2). It is possible to increase the pressure at the point A in two ways, i.e. by increasing the load torque T_M on the motor shaft or by closing of the throttle valve TV. The cross-section area A_{TV} is reduced due to the closing of the throttle valve.

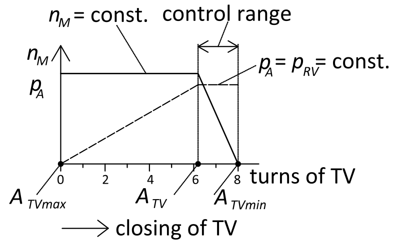

Figure 2. Dependence of motor speed n_M and pressure p_A on cross-section area of throttle valve A_{TV}

Hydrostatic drives are in general considered to be flexible drives. It is caused by the fact that a load change (in this case the motor torque change T_M) leads to the speed change n_M. The dependence of the speed on the load torque for a constant cross-section area of the throttle valve A_{TV} is shown in Fig. 3. This dependence is called as the torque-speed characteristic.

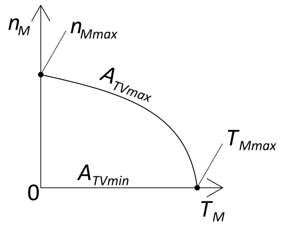

Figure 3. Dependence of motor speed n_M on load torque T_M (torque-speed characteristic)

It is possible to mathematically express the speed dependence on the load torque (see Fig. 3). It is possible to appear from the flow rate equality through the hydraulic motor HM and the throttle valve TV (see Fig. 1):

$$Q_M = Q_{TV}. \quad (1)$$

The motor flow rate Q_M can be defined as the product of the motor geometric displacement volume V_M and the motor speed n_M:

$$Q_M = V_M \cdot n_M. \quad (2)$$

The flow rate through the throttle valve is given by the equation:

$$Q_{TV} = \mu_{TV} \cdot A_{TV} \cdot \sqrt{\frac{2 \cdot \Delta p_{TV}}{\rho}}, \quad (3)$$

where μ_{TV} is the flow coefficient of the throttle valve, A_{TV} is the cross-section area of the throttle valve, Δp_{TV} is the pressure gradient through the throttle valve and ρ is the density of the working liquid.

It is possible to express the pressure gradient Δp_{TV} through the throttle valve by means of the pressure gradient p_{RV} adjusted on the pressure relief valve and the pressure gradient Δp_M on the motor:

$$\Delta p_{TV} = p_{RV} - \Delta p_M. \quad (4)$$

The pressure gradient Δp_M on the motor is given by the equation:

$$\Delta p_M = \frac{2 \cdot \pi \cdot T_M}{V_M}. \quad (5)$$

Substituting the equations 2 ÷ 5 into the equation 1, it is possible to obtain the equation of the dependence of the motor speed on the torque $n_M = f(T_M)$, see Fig. 3, eventually dependencies of the motor speed on the cross-section area of the throttle valve $n_M = f(A_{TV})$:

$$n_M = \frac{\mu_{TV} \cdot A_{TV}}{V_M} \cdot \sqrt{\frac{2}{\rho}} \cdot \sqrt{p_{RV} - \frac{2 \cdot \pi \cdot T_M}{V_M}}. \quad (6)$$

Based on the equation (6), it is possible to plot the drive control characteristic $n_M = f(A_{TV})$, i.e. the dependence of the motor speed n_M on the cross-section area of the throttle valve A_{TV} at the constant motor torque T_M (see Fig. 4).

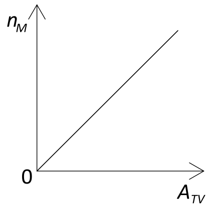

Figure 4. Dependence of motor speed n_M on cross-section area of throttle valve A_{TV} (control characteristic)

3 DESCRIPTION OF HYDRAULIC SYSTEM AND SPECIFICATION OF HYDRAULIC ELEMENTS

The hydraulic pump HP1, the pressure relief valve RV1, and the tank T1 are a pressure liquid source. The pressure relief valve RV1 is used to adjust the working pressure in the hydraulic system. The rotary hydraulic motor HM is driven from this source. The torque sensor TQS, the shaft speed sensor SS and the hydraulic pump HP2 are located on the shaft of the hydraulic motor. The hydraulic pump HP2 with the pressure relief valve RV2 and the tank T2 are used to generate the torque on the shaft of the hydraulic motor HM. The speed control of the hydraulic motor is realized by the throttle valve TV. The hydraulic pump HP1 allows changing the geometric displacement volume by means of an adjusting screw. The pressure gradient through the hydraulic motor HM was measured by means of the pressure sensors PS1 and PS2. The oil flow rate was measured by the flow sensor FS. Time dependencies of the measured quantities for the given adjustment were recorded using the MS 5060+ equipment during measurements. Average values of the quantities for a given measurement were evaluated from a record using Hydrowin software. The measurements were performed at a constant oil temperature, which is measured by the temperature sensor TS. The specification of the used hydraulic elements and sensors is shown in Tab. 1.

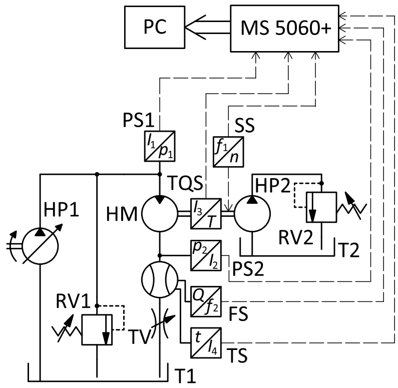

Figure 5. Simplified schematic diagram of hydraulic system

Symbol	Specification
HP1	Axial piston variable pump with manual setting angle of swashplate, PPAR 2–63 10AP (Vrchlabi), V_P = (0 ÷ 51.22) cm³, n_{max} = 2600 min⁻¹, p_n = 250 bar
HP2	Bent axis axial piston fixed pump, HM 28 AV (Glentor), V_P = 28.5 cm³, n_{max} = 4800 min⁻¹, p_n = 250 bar
HM	Gear hydraulic motor, SM 05 / 10 (SM Hydro Group), V_M = 5.8 cm³, n_{max} = 2500 min⁻¹, p_n = 220 bar
RV1	Pressure relief valve, ARAM-20/350/70 (Atos), p_{max} = 350 bar, Q_{max} = 350 dm³·min⁻¹
RV2	Pressure relief valve, VP2-20 (Vrchlabi), p_{max} = 160 bar, Q_{max} = 100 dm³·min⁻¹
TV	Throttle valve, 9N1200S (Parker), p_{max} = 345 bar, Q_{max} = 95 dm³·min⁻¹
T1	Tank, V = 240 dm³
T2	Tank, V = 75 dm³
PS1, PS2	Pressure sensor, PR15 (Hydrotechnik), measuring range (0 ÷ 200) bar, measurement accuracy of 0.5 %
TS	Temperature sensor, Pt 100 (Hydrotechnik), Measuring range (-50 ÷ 200) °C, measurement accuracy of 1 %
TQS	Torque sensor, T22/50NM (HBM), measuring range (± 50) N·m, measurement accuracy of 0.5 %
SS	Speed sensor, DS 03 (Hydrotechnik), signal cycle frequency max. 500 Hz
FS	Flow sensor, Gear flow meter GFM70 (Hydrotechnik), measuring range (0.7 ÷ 70) dm³·min⁻¹
MS 5060+	Measuring system, MS 5060+ (Hydrotechnik)
Oil	Mineral oil VG46, t_O = (40 ± 2) °C, hydraulic power unit is equipped with cooler

Table 1. Specification of used hydraulic elements and sensors

4 THEORETICAL CHARACTERISTICS OF ASSEMBLED HYDRAULIC SYSTEM

Theoretical characteristics of the appropriate hydraulic system, which is described in Chapter 3, will be created according to the theoretical equation in this chapter. It is possible to determine several dependencies of the motor speed on the torque $n_M = f(T_M)$ for different cross-section areas A_{TV} of the throttle valve and different sizes of the torque T_M (see Fig. 6) based on the equation (6), where the flow coefficient of the throttle valve μ_{TV} = 0.73, the geometric displacement volume of the hydraulic motor $V_M = 5.8\ cm^3$, the oil density ρ = 868 kg·m^{-3} and the working pressure p_{RV} = 150 bar.

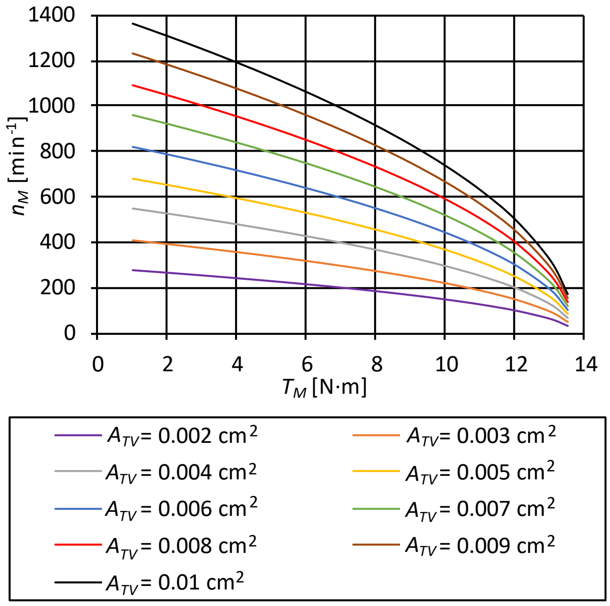

Figure 6. Theoretical torque-speed characteristic $n_M = f(T_M)$ of hydraulic drive

It is evident from the dependencies (see Fig. 6) for single cross-section area A_{TV} of the throttle valve that the motor speed n_M is decreasing with increasing the torque T_M on the hydraulic motor.

In addition, it is possible to determine the theoretical drive control characteristic $n_M = f(A_{TV})$, i.e. the dependence of the speed n_M of the hydraulic motor on the cross-section area A_{TV} of the throttle valve for several constant motor torques T_M, see Fig. 7.

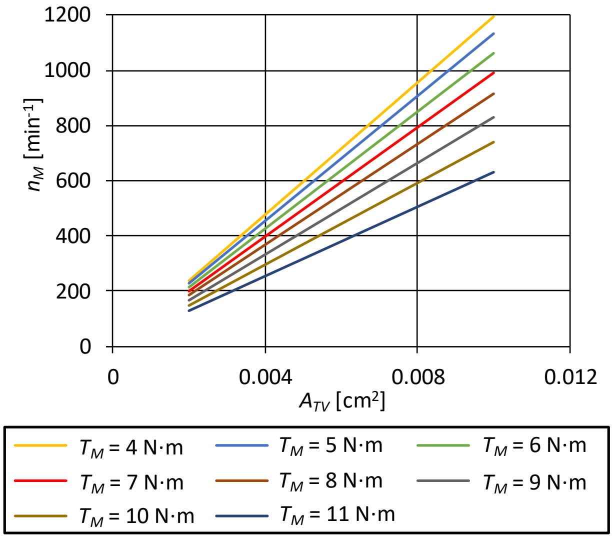

Figure 7. Theoretical control characteristic $n_M=f(A_{TV})$ of hydraulic drive

It is evident from Fig. 7 that the motor speed n_M is increasing with increasing the cross-section area A_{TV} of the throttle valve at a constant motor torque T_M.

5 MEASUREMENT OF CHARACTERISTICS OF HYDRAULIC DRIVE

5.1 *Measurement of torque-speed characteristic of hydrostatic drive*

The pressure p_{RV} = 150 bar is adjusted by the pressure valve RV1. The flow rate Q_P = 8 dm^3·min^{-1} is adjusted by the hydraulic pump HP1. The throttle valve TV is adjusted in such a way that a part of the flow rate is flowing through the pressure relief valve RV1. Subsequently, the speed n_M of the hydraulic motor, the pressure gradient Δp_M through the hydraulic motor, the oil temperature t_O and the flow rate Q_M through the hydraulic motor are measured. These measurements are performed for several load torques T_M on the motor and the cross-section areas A_{TV} of the throttle valve TV. An increasing of the load torque T_M on the shaft of the hydraulic motor is realized by an increasing resistance on the output of the hydraulic pump HP2 by means of the pressure relief valve RV2. This measurement is repeated for several cross-section areas of the throttle valve. Since a specific cross-section area at a given adjustment of the throttle valve is not known, these cross-section areas are marked with the parameters $\varphi_1 \div \varphi_5$. The measured torque-speed characteristic $n_M = f(T_M)$ of the hydraulic drive for several adjustments of the cross-section area A_{TV} of the throttle valve and several load torques T_M is shown in Fig. 8.

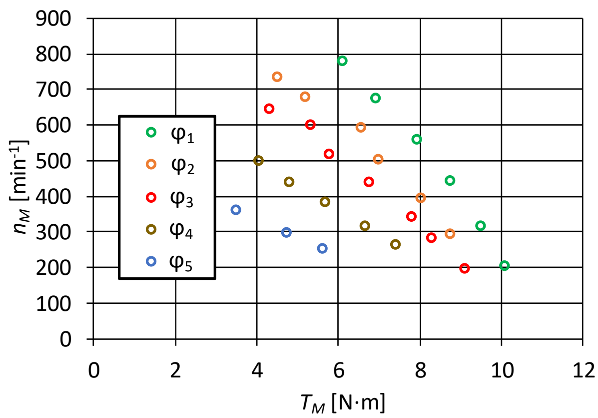

Figure 8. Measured torque-speed characteristic $n_M = f(T_M)$ of hydraulic drive

5.2 *Measurement of control characteristic of hydrostatic drive*

The pressure p_{RV} = 150 bar is adjusted by the pressure relief valve RV1. The flow rate Q_P = 8 dm^3·min^{-1} is adjusted by the hydraulic pump HP1. The flow rate Q_M through the hydraulic motor is controlled by means of the cross-section area A_{TV} of the throttle valve. Subsequently, the speed n_M of the hydraulic motor, the pressure gradient Δp_M through the hydraulic motor, the oil temperature t_O and the flow rate Q_M through the hydraulic motor are measured. These measurements are performed for several cross-section areas A_{TV} of the throttle valve and a constant load torque T_M on the motor. The constant load torque T_M on the motor shaft is maintained by means of the pressure relief valve RV2, which is located on the output of the hydraulic pump HP2. This measurement is repeated for several load motor torques T_M.

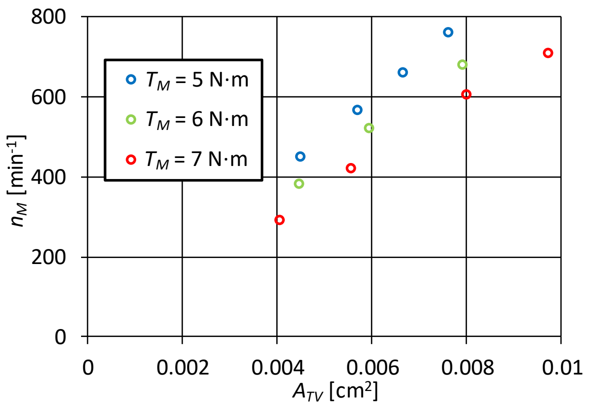

Figure 9. Measured control characteristic $n_M = f(A_{TV})$ of hydraulic drive

The measured control drive characteristic $n_M = f(A_{TV})$, i.e. the dependence of the speed n_M of the hydraulic motor on the cross-section area A_{TV} of the throttle valve for several constant load motor torques T_M, is shown in Fig. 9. It is evident that the motor speed n_M is increasing with increasing the cross-section area A_{TV} of the throttle valve at the constant motor torque T_M.

6 COMPARISON OF THEORETICAL AND MEASURED CHARACTERISTICS OF HYDROSTATIC DRIVE

There are compared measured characteristics with theoretical assumptions in this chapter. The comparison of the theoretical and measured torque-speed characteristics of the hydrostatic drive is shown in Fig. 10. It is visible that the dependence of the measured characteristics does not correspond to the theoretically determined dependencies. It is caused by the fact that the efficiencies of single hydraulic elements were not included in the theoretical analysis.

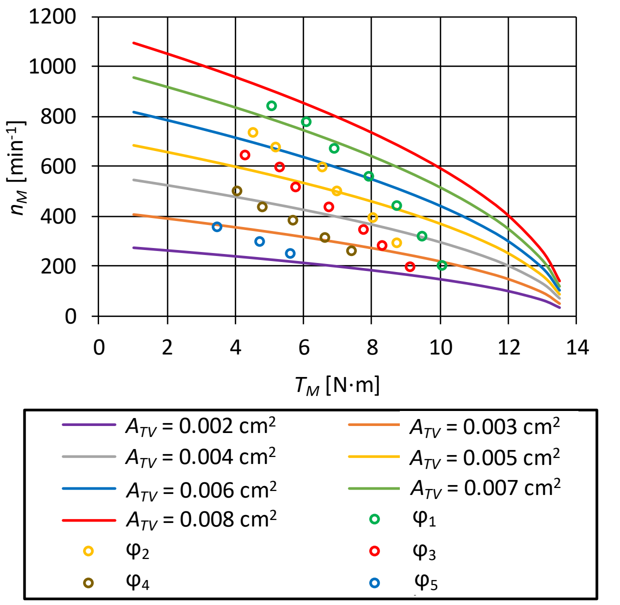

Figure 10. Comparison of theoretical and measured torque-speed characteristics

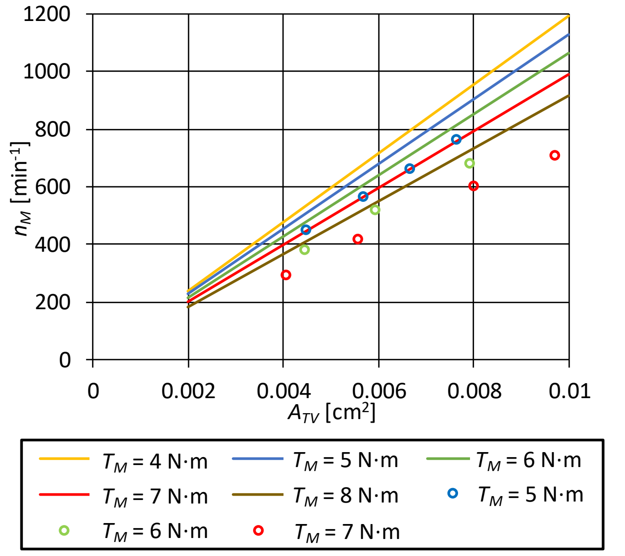

Figure 11. Comparison of theoretical and measured control characteristics

The comparison of the theoretical and measured control characteristics of the hydrostatic drive is shown in Fig. 11. It is visible that the measured control characteristics for single load motor torques do not correspond to the theoretically determined characteristics.

7 EFFICIENCIES OF ROTARY HYDRAULIC MOTOR

There are determined the efficiencies of the rotary hydraulic motor in this chapter, i.e. the volumetric, mechanical-hydraulic and total efficiencies, e. g. [Paszota 2010], [Hruzik 2014].

7.1 Volumetric efficiency

The volumetric efficiency of the rotary hydraulic motor HM is determined from the equation:

$$\eta_{M,V} = \frac{V_M \cdot n_M}{Q_M}. \tag{7}$$

The volumetric efficiencies depending on the pressure gradient Δp_M of the rotary hydraulic motor for several constant speeds are shown in Fig. 12. It is evident that the volumetric efficiency $\eta_{M,V}$ decreases with increasing the pressure gradient Δp_M and increases with increasing the speed n_M.

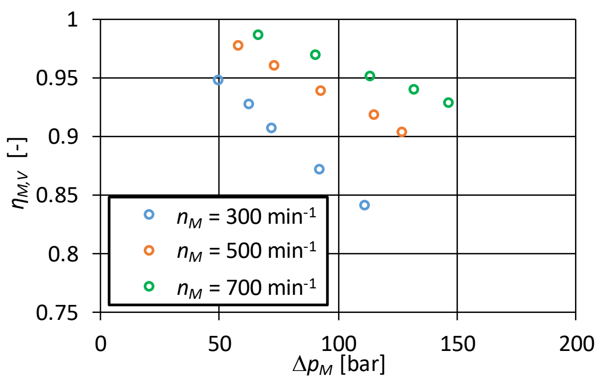

Figure 12. Dependence of volumetric efficiency on pressure gradient for several constant speeds

7.2 Mechanical-hydraulic efficiency

Mechanical-hydraulic efficiency of the rotary hydraulic motor HM is expressed by the equation:

$$\eta_{M,mh} = \frac{2 \cdot \pi \cdot T_M}{\Delta p_M \cdot V_M}. \tag{8}$$

The mechanical-hydraulic efficiencies depending on the pressure gradient Δp_M of the rotary hydraulic motor for several constant speeds are shown in Fig. 13. It is evident that the mechanical-hydraulic efficiency $\eta_{M,mh}$ increases with increasing the pressure gradient Δp_M and decreases with increasing the speed n_M.

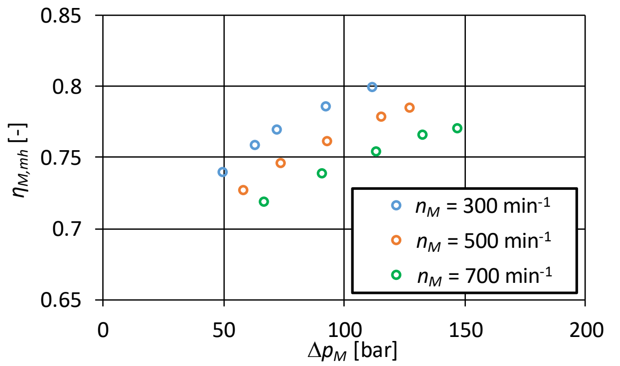

Figure 13. Dependence of mechanical-hydraulic efficiency on pressure gradient for several constant speeds

7.3 *Total efficiency*

The total efficiency is given by the product of the volumetric and mechanical-hydraulic efficiencies:

$$\eta_{M,t} = \eta_{M,V} \cdot \eta_{M,mh}. \quad (9)$$

The total efficiencies depending on the pressure gradient Δp_M through the rotary hydraulic motor for several constant speeds are shown in Fig. 14.

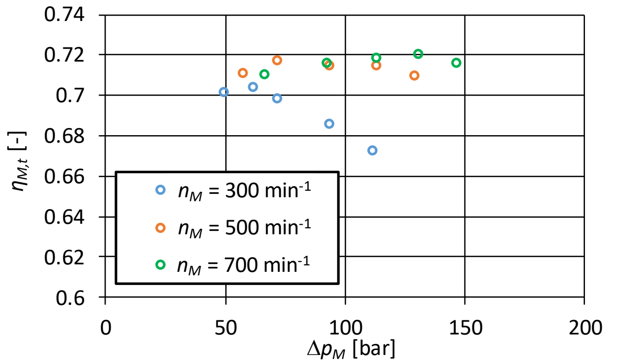

Figure 14. Dependence of total efficiency on pressure gradient for several constant speeds

8 INCLUSION OF EFFICIENCY OF HYDRAULIC MOTOR IN THEORETICAL CHARACTERISTIC

The efficiencies of the hydraulic motor determined in Chapter 7 aren't constant. They have different values depending on the change of the speed n_M (or the flow rate Q_M) or the torque T_M (or the pressure gradient Δp_M). Thus, it is possible to substitute concrete values of the efficiencies for each steady state or to use the average values of these efficiencies in a simplified way. In this chapter, the theoretical characteristics will be extended by the average values of the volumetric and mechanical-hydraulic efficiencies of the hydraulic motor that were determined in Chapter 7.

On the basis of the determined volumetric efficiency, it is possible to modify the equation (2) for the calculation of the flow rate through the hydraulic motor as follows:

$$Q_M = V_M \cdot n_M \cdot \frac{1}{\eta_{M,V}}, \quad (10)$$

where the mean value of the volumetric efficiency of the hydraulic motor from the measured range $\eta_{M,V} = 0.93$.

On the basis of the determined mechanical-hydraulic efficiency, it is possible to modify the equation (5) for the calculation of the pressure gradient through the hydraulic motor as follows:

$$\Delta p_M = \frac{2 \cdot \pi \cdot T_M}{V_M} \cdot \frac{1}{\eta_{M,mh}}, \quad (11)$$

where the mean value of the mechanical-hydraulic efficiency of the hydraulic motor from the measured range $\eta_{M,mh} = 0.76$.

Extending the equation (6) by the efficiencies of the hydraulic motor, it is possible to obtain the theoretical equation for calculation of the torque-speed characteristic or the control characteristic including the effect of the efficiencies of the hydraulic motor:

$$n_M = \frac{\mu_{TV} \cdot A_{TV}}{V_M} \cdot \eta_{M,V} \cdot \sqrt{\frac{2}{\rho}} \cdot \sqrt{p_{RV} - \frac{2 \cdot \pi \cdot T_M}{V_M \cdot \eta_{M,mh}}}. \quad (12)$$

Fig. 15 shows the measured (points in the diagram), theoretical (continuous lines in the diagram) and theoretical (including the efficiency of the hydraulic motor – i.e. dashed lines in the diagram) torque-speed characteristics of the hydraulic motor. It is visible that these characteristics are closer to the measured dependencies when considering the efficiencies of the hydraulic motor into the theoretical equation for the calculation of the torque-speed characteristic.

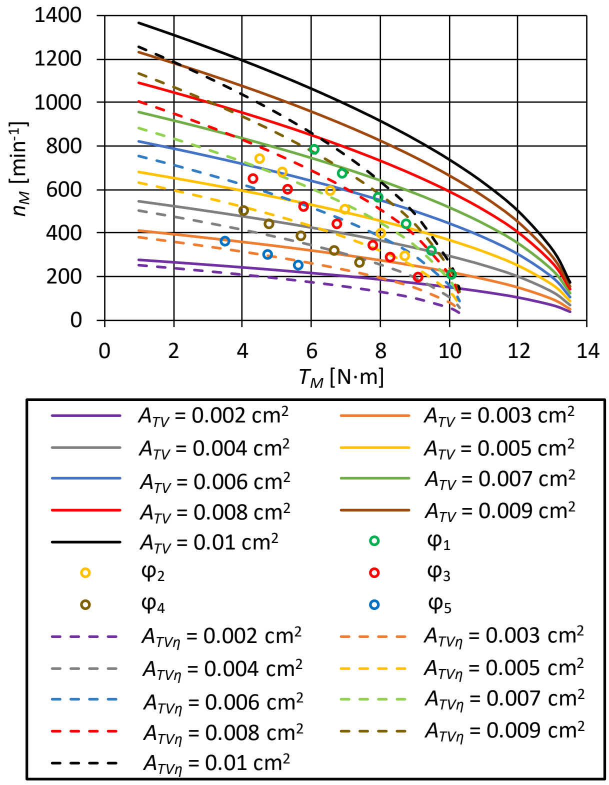

Figure 15. Comparison of measured and theoretical torque-speed characteristics and theoretical torque-speed characteristic including efficiency of hydraulic motor

Fig. 16 shows the measured (points in the diagram), theoretical (continuous lines in the diagram) and theoretical (including the efficiency of the hydraulic motor – i.e. dashed lines in the diagram) control characteristics of the hydraulic motor. It is visible that courses of these dependencies are closer to the measured dependencies when considering the efficiencies of the hydraulic motor into the theoretical equation for the calculation of the control characteristic.

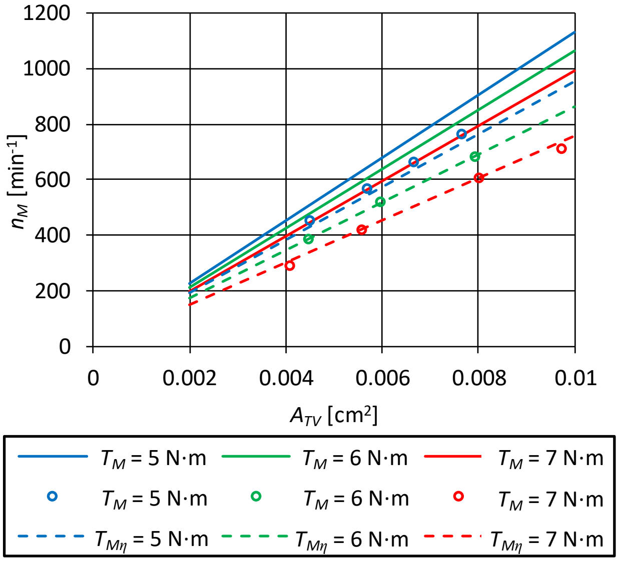

Figure 16. Comparison of measured and theoretical control characteristics and theoretical control characteristic including efficiency of hydraulic motor

It is evident from Fig. 15 that each measured torque-speed characteristic intersects several theoretically determined torque-speed characteristics including the efficiencies of the hydraulic motor. It is evident mainly at lower values of the cross-section areas A_{TV} of the throttle valve. It is given by the fact that the average values of the efficiencies of the hydraulic motor are substituted into the equation for the calculation of the theoretical torque-speed characteristics. For the static calculation, concrete values of the volumetric and mechanical-hydraulic efficiencies for the given steady state can be substituted into the equation (12).

9 MATHEMATICAL MODEL OF MEASURED HYDROSTATIC DRIVE

There are two possibilities for dynamic solving of this drive. Firstly, it is possible to neglect the effect of changes in the efficiencies of the hydraulic motor depending on the speed and the pressure gradient and to enter them as mean constant values. Or it is possible to change these efficiencies for each time depending on the speed n_M (or the flow rate Q_M) or the torque T_M (or the pressure gradient Δp_M). The latter variant can be realized using Matlab Fluids software for 1D mathematical simulation of fluid mechanisms. In this software, the volumetric and mechanical-hydraulic efficiencies are specified in the tabular form.

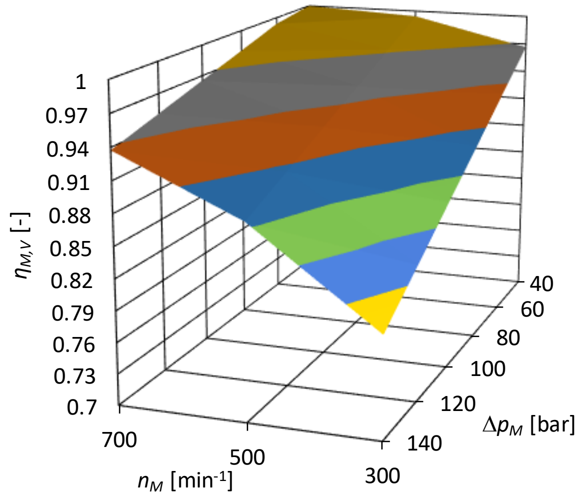

Figure 17. Dependence of volumetric efficiency of hydraulic motor on pressure gradient Δp_M and speed n_M of hydraulic motor

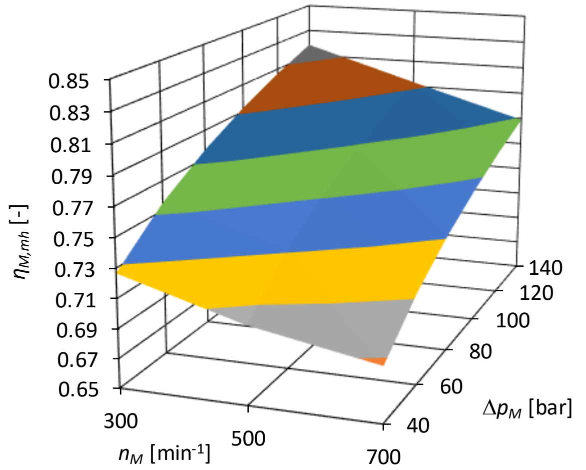

Figure 18. Dependence of mechanical-hydraulic efficiency of hydraulic motor on pressure gradient Δp_M and speed n_M of hydraulic motor

Data entered in this way corresponds to area dependencies of the volumetric and mechanical-hydraulic efficiencies on the motor pressure gradient Δp_M and the motor speed n_M. The dependence of the volumetric efficiency on the pressure gradient Δp_M and the motor speed n_M is shown in Fig. 17. The dependence of the mechanical-hydraulic efficiency on the pressure gradient Δp_M and the motor speed n_M is shown in Fig. 18.

9.1 *Description of mathematical model of experimental equipment*

The mathematical model was created using Matlab Fluids software (see Fig. 19), e. g. [Mehennaoui 2015], [Jablonska 2017]. Steady states of the drive for different cross-section areas A_{TV} of the throttle valve and different torques T_M on the shaft of the hydraulic motor were simulated by means of this model.

The hydraulic pump HP1 is a pressure liquid source and is driven by the motor M. Mineral oil flows to the hydraulic motor HM, further through the flow sensor FS and the throttle valve TV to the tank T1. The hydraulic motor HM is loaded by means of the hydraulic pump HP2 and the pressure relief valve RV2. The system pressure is adjusted by means of the pressure relief valve RV1. Furthermore, the model consists of the block for the size adjustment „φ“ of the cross-section area A_{TV} of the throttle valve, the blocks PS1 and PS2 for pressure measurement, the block SS for speed measurement, the block TQS for torque measurement and the solver block „Solver“. Parameters of the hydraulic oil are adjusted by the block „Oil“.

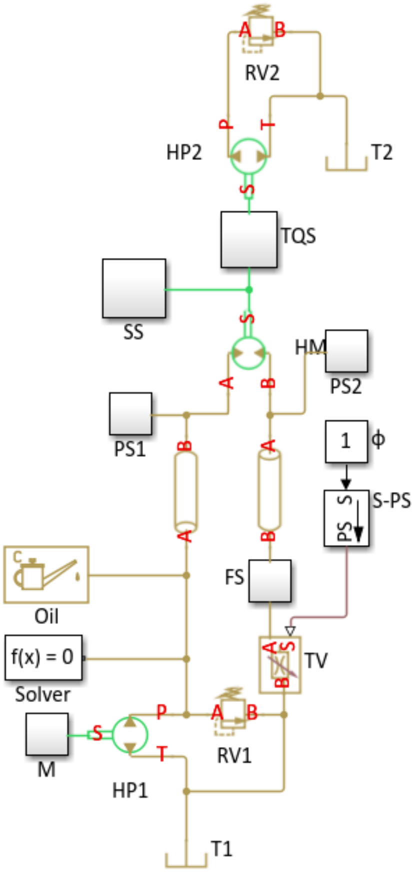

Figure 19. Schematic diagram of mathematical model of experimental equipment

The mathematical model of the throttle valve was defined by means of the Δ*p*-*Q* characteristics (see Fig. 20) that were measured for the single openings $\varphi_1 \div \varphi_5$ of the throttle valve.

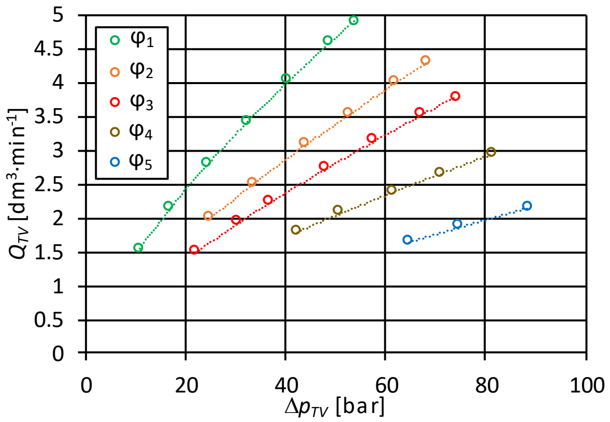

Figure 20. Measured Δ*p*-*Q* characteristics of throttle valve for single openings $\varphi_1 \div \varphi_5$.

9.2 Simulation of torque-speed characteristic

The speed n_M and the torque T_M on the hydraulic motor for the given adjustment range of the pressure relief valve RV2, which loads the hydraulic pump HP2, were simulated for the single adjustments $\varphi_1 \div \varphi_5$ of the throttle valve TV in the mathematical model (see Fig. 19). The result of the simulated torque-speed characteristics for the single openings of the throttle valve TV is compared in Fig. 21 with the experimentally determined characteristics. It is visible that the simulated characteristics almost correspond to the measured characteristics.

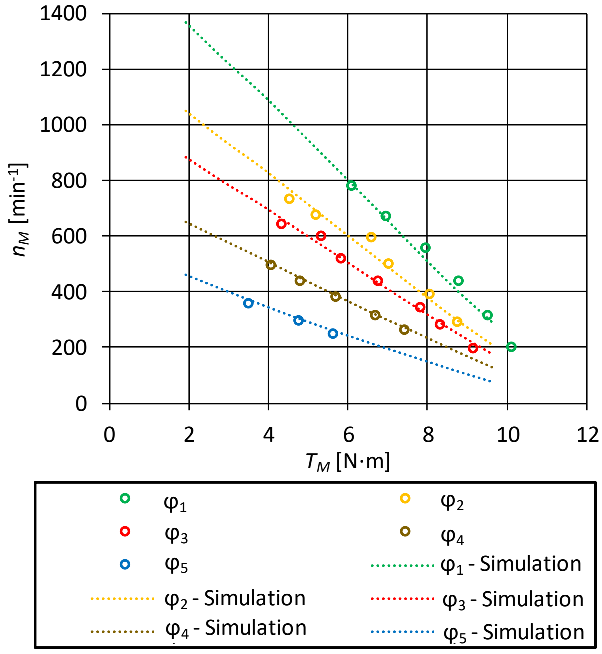

Figure 21. Comparison of measured and mathematically simulated torque-speed characteristics

10 CONCLUSIONS

This paper was focused on valve control of a hydrostatic drive. The theoretical analysis was performed on a system controlled by the throttle valve. The theoretical torque-speed and control characteristics were determined for the drive, where the pump and motor efficiencies, pressure losses and pressure dependence of the pressure relief valve are not taken into account. Subsequently, the torque-speed and control characteristics were experimentally measured on the assembled hydraulic drive. It was found by comparing the theoretical and measured characteristics that these characteristics are significantly different. These deviations are mainly caused by the efficiency of the hydraulic motor. Therefore the volumetric, mechanical-hydraulic and total efficiencies of the rotary hydraulic motor were experimentally determined. Mean values of these efficiencies were subsequently considered at the theoretical determination of the torque-speed and control characteristics. A greater agreement was achieved by comparing these characteristics with the measured characteristics in respect to the original theoretical characteristics without including the efficiencies of the hydraulic motor. The volumetric and mechanical-hydraulic efficiencies of the rotary hydraulic motor depend on the speed and the motor torque. The exact mathematical description of this hydraulic system was performed using Matlab Fluids software, which allows simulations with the efficiencies for instantaneous values of the speed and the torque on the hydraulic motor. The torque-speed characteristics, which were simulated in this manner, already closely correspond to the measured characteristics.

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