STATIC ANALYSIS OF THE HYDRAULIC FRONT-WHEEL DRIVE SYSTEM OF A UNIVERSAL WHEELED TRACTOR

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Abstract
This article presents the results of a static analysis of the hydraulic front-wheel drive system of a universal wheeled tractor. The study focuses on evaluating the balance of forces and moments acting on the front drive assembly under stationary conditions. The methodology includes determining the required hydraulic pressure and flow rate to ensure the stable operation of the front drive mechanism. Key structural parameters of the system, such as cylinder diameter, piston stroke, and mounting geometry, are considered in the calculation. The outcomes of the analysis provide essential data for improving the operational efficiency, reliability, and load-bearing capacity of front-wheel drive systems in agricultural machinery.

Keywords: Hydraulic system, front-wheel drive, universal tractor, static analysis, cylinder force, agricultural machinery, drive mechanism, load distribution, design parameters.

Introduction

In modern agricultural machinery, the use of front-wheel drive systems in universal tractors plays a vital role in enhancing traction, maneuverability, and overall operational efficiency. Hydraulic front-drive mechanisms, in particular, offer flexibility and power transmission capabilities that are well-suited for various field conditions and load scenarios.

Understanding the static behavior of these systems is critical during the design and engineering stages. Static analysis allows for accurate estimation of the forces and torques acting on the front drive components when the tractor is stationary or moving at low speeds. This analysis helps engineers optimize hydraulic parameters such as pressure and flow rate, ensuring efficient energy use and structural integrity.

This paper focuses on the static analysis of the hydraulic front-wheel drive unit of a universal tractor, aiming to identify optimal design parameters and operating conditions. The results serve as a foundation for further dynamic analysis, modeling, and practical design improvements in agricultural equipment engineering.

Materials and Methods

In modern agricultural practices, efficient use of high-powered universal row-crop tractors and reduction in the number of passes across the field have become essential requirements for increasing productivity and minimizing soil compaction. To achieve these objectives, tractors are increasingly being equipped with additional functional units, enabling them to perform multiple operations in a single pass.

One of the key technical solutions for enhancing the performance of such tractors is the integration of hydraulic front-wheel drive systems. These systems significantly improve traction, reduce slippage, and enhance the tractor's maneuverability in challenging field conditions, especially on loose or wet soil.

In this study, a static analysis was carried out to evaluate the working conditions of the hydraulic front-wheel drive system installed on a universal row-crop tractor. The methodology included:

- Selection of the tractor model and its geometric configuration.
- Identification of the hydraulic drive components, including the front hydraulic motor, cylinder dimensions, connecting rods, and mounting brackets.
- Calculation of axial forces, radial loads, and moments acting on the hydraulic unit during standstill and low-speed operation.
- Determination of required hydraulic pressure (P) and flow rate (Q) to maintain balanced and stable operation.
- Analysis of the load distribution between the rear and front axles with and without the hydraulic front-drive system.

The calculation was performed under the assumption of steady-state conditions, considering the tractor's total weight, center of gravity, front axle load, and external resistive forces. The basic structural and hydraulic parameters were derived from technical manuals and experimental field data [1].

A schematic representation of the front-wheel hydraulic drive system is shown in Figure 1, which illustrates the key mechanical components and their interaction within the drive configuration.

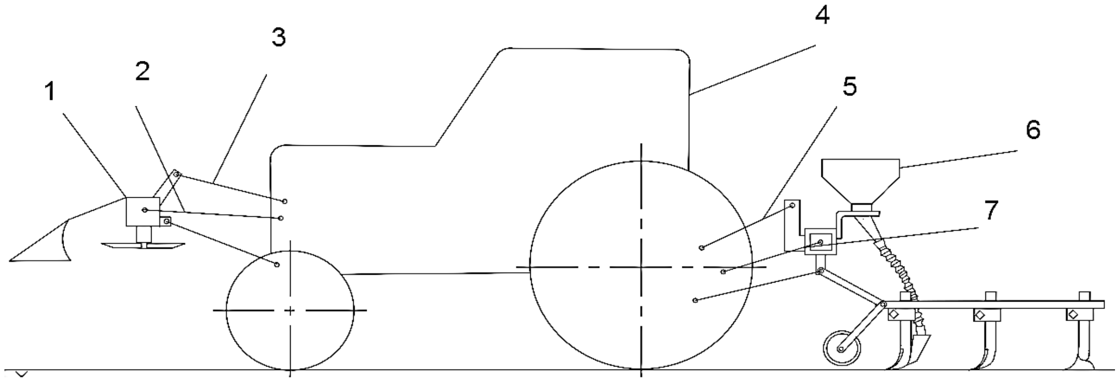

Figure 1. Schematic diagram of a tractor equipped with front and rear hydraulic hitch systems aggregated with agricultural implements.

1 – Front hydraulic hitch with active implement (rotary tiller); 2 – Front Power Take-Off (FPTO); 3 – Front hydraulic hitch system; 4 – Tractor; 5 – Rear hydraulic hitch system; 6 – Rear hydraulic hitch with active implement (cultivator); 7 – Rear Power Take-Off (PTO).

The main component of the additional front-wheel hydraulic drive system in a universal row-crop tractor is the hydraulic system, which is composed of several interconnected elements ensuring fluid power transmission and control (see Figure 2). The system includes:

1 – Hydraulic fluid reservoir, 2 – Filter, 3 – Hydraulic pump, 4 – Hydraulic distributor (control valve), 5 – Pressure and return pipelines, 6 – Hydraulic motor, 7 – Hydraulic cylinders.

These components collectively enable the conversion of hydraulic energy into mechanical motion, thus providing additional traction to the front wheels. The working fluid is drawn from the reservoir (1), filtered (2), and pressurized by the hydraulic pump (3). It is then directed by the distributor (4) through the pressure lines (5) to activate the hydraulic motor (6), which drives the front wheels. Simultaneously, hydraulic cylinders (7) assist in positioning or lifting associated front-mounted implements. The return flow of the fluid completes the circuit by returning to the reservoir via return pipelines.

The configuration and sizing of these components are crucial for the system's reliability, efficiency, and adaptability under various field load conditions.

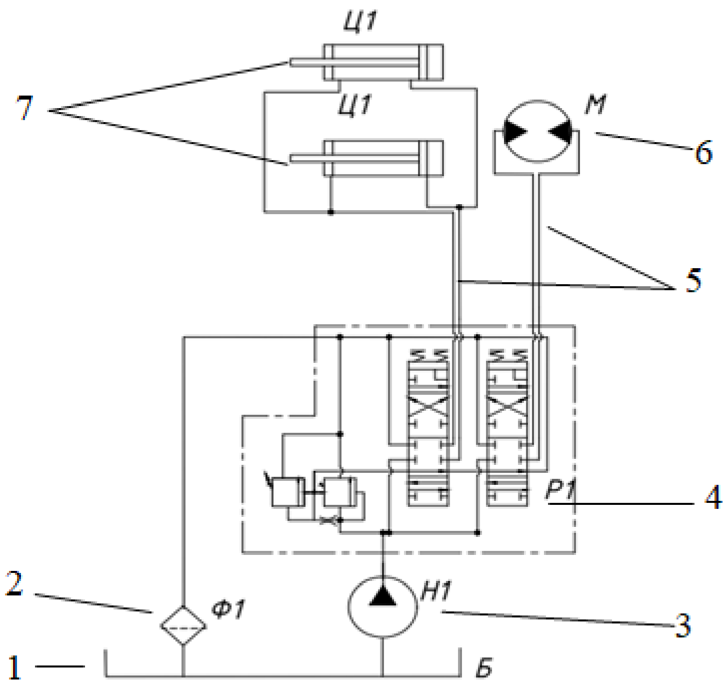

Figure 2. Schematic diagram of the front hydraulic hitch system.

Results and Discussion

Power Requirement Analysis for Active Implements in the Front Hydraulic Hitch System

The analysis of active implement attachments intended for use with the front hitch systemcontrolled via the tractor’s hydraulic drive—under integrated agricultural field operations reveals that the power required to drive the active components of such implements must not be less than 50 horsepower (33.76 kW) [8, 9, 10]. This level of power demand is essential to ensure the effective performance of mechanical operations such as rotary tillage, soil cultivation, and residue management.

To meet this requirement, the active parts of the implements are driven through a Power Take-Off (PTO) shaft connected to a hydraulic motor of the newly designed additional front-wheel hydraulic drive system. According to GOST 3480-2020, the rotational speed of the PTO shaft must be within the range of 540 to 1000 rpm, which is the standardized operating speed for agricultural PTO systems [6].

When using a hydraulic front-mounted PTO, it becomes necessary to determine the hydraulic drive parameters that match the required power output and rotational speed of the active implements. Since both the required power and rotational speed are predefined, the torque resistance necessary to operate the system can be calculated in accordance with GOST 30747-2001, using the following expression [2]:

$$M_{qov} = \frac{N_{qov} \times 9550}{n_{qov}}, \tag{1}$$

Here:

N_{qov} - front power take-off shaft power,

n_{qov} - number of revolutions of the front power take-off shaft.

We determine the fluid working volume of the hydraulic motor using the resistance moment found in expression (1) [7].

$$q_{gm} = \frac{2\pi \cdot M_{qm}}{p} \cdot sm^3 / ob \tag{2}$$

Here:

p- nom Nominal pressure of the working fluid in the hydraulic system MPa (=16 MPa);

q_{gm} - Required resistance torque for the front power take-off shaft;

The working fluid consumption of a hydraulic motor is determined as follows [7].

$$Q_{gm} = \frac{q_{gm} \cdot n_m}{\eta_{gm}} \cdot l / \min \tag{3}$$

Here:

n_{gm} - maximum number of revolutions of the hydraulic motor shaft, rpm.

q_{gm} - working volume of liquid.

Based on the following determined values, we select the hydraulic motor with the closest parameters from the hydraulic motors listed in Table 1.

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Table 1. Technical specifications of selected hydraulic motors for front-wheel drive system

Size	Axial piston hydraulic motors 310.12.01.	Axial piston hydraulic motors 310.28.01.	Axial piston hydraulic motors 310.56.01.	Gerator hydraulic motors MGP8080	Gerator hydraulic motors MGP100	Gerator hydraulic motors MGP125	Geared hydraulic motors ALM2-R-1	Geared hydraulic motors ALM2-R-37-E1	Geared hydraulic motors ALM2-R-40-E1
Working volume q, cm^3/rpm	11,6	28	56	80,5	100	125,7	23.2	25.5	28.7
Number of revolutions n, rpm/min, n_{min}	50			10			650		
n_{nom}	2400	1920	1800	350	270	212			
n_{max}	6000	4750	3750	850	650	512	3000	2800	2500
Fluid consumption Q, l/min, Q_{min}	0,58	1,40	2,80						
Q_{nom}	27,84	53,76	84,00	30			33	36	40
Q_{max}	69,60	133,00	210,00						
Fluid pressure, P, MPa, P_{nom}	20	20	20	21	21	21			
P_{max}	32	32	35	25	25	25	17	17	18
Power N, kVt - N_{min}	9,28	17,92	33,60						
N_{nom}	14,84	28,67	58,80	7.25					
N_{max}	-	44	67,20				25.7	26.7	27.8
Torque M, M_{nom}	35	84,6	169,3	196	250	315			
M_{max}	56,1	135,5	296,3				123	128	133
HMM	0,95			0.78			0.93		

Among the hydraulic motors listed in Table 1, the axial piston motor 310.28.01 was selected as the most appropriate option based on the values calculated using Equations (1)–(3).

If the power output of the hydraulic motor matches the required power input of the front Power Take-Off (PTO) shaft, but the shaft speed and torque of the motor differ from the required PTO specifications (540–1000 rpm and torque of 322.4 Nm), a reduction gearbox is employed to adjust the output to the desired levels.

Specifically, the selected hydraulic motor has a maximum power output of 44 kW, a maximum rotational speed of 4750 rpm, and a torque of 135 Nm. Therefore, in order to adapt this performance to the required PTO parameters, a gear reducer is necessary.

To select an appropriate reducer, it is first essential to determine the gear ratio. Since the rotational speeds of both the input (motor shaft) and output (PTO shaft) are known, the required gear ratio is calculated using the following formula:

$$i_r = \frac{n_{gm}}{n_{qov}},$$

$$i_r = \frac{n_m}{n_v} ay/min \qquad (4)$$

Here:

n_m– Hydraulic motor shaft speed, rpm;
n_v – Front hydraulic PTO shaft speed.

Based on the obtained value of i_r , a suitable gear reducer is selected from the standard series of gear ratios, which includes the following values: 1.25; 1.4; 1.6; 1.8; 2.0; 2.24; 2.5; 2.8; 3.15; 3.55; 4.0; 4.5; 5.0; 6.3; 7.1; 8.0; 9.0; 10.0 [5].

Taking into account the gear ratio of the reducer, the calculated working displacement of the hydraulic motor is determined using the following formula.

$$q_{hm} = \frac{2\pi \times M_{qov}}{p \times \eta_{gm} \times \eta_r \times i_r},$$
$$q_{hm} = \frac{2\pi M_{qov}}{P_n \cdot \eta_{nm} \cdot \eta_r \cdot i_r} \cdot sm^3 \quad (5)$$

Here:

M_{qov} – Hydraulic motor shaft torque (Nm), P_n – nominal pressure of the hydraulic system (MPa), η_{nm}– mechanical (HMM) of the hydraulic motor (0.85- 0.9), η_r– reducer (HMM), i_r– reduction ratio: if there is no reducer in the system, the values of η_r and i_r are ignored. (η_r η_r=1, i=1)

Nominal design pressure in the hydraulic motor.

$$p_{hm} = \frac{2\pi \times M_{qov}}{q_{gm} \times \eta_{gm} \times \eta_r \times i_r}, MPa \quad (6)$$

Here:

M_{qov} - torque of the hydraulic motor shaft (Nm),

q_m- hydraulic motor displacement (cm^3/rpm).

When selecting a hydraulic motor and checking its operating mode, the following conditions must also be observed [4].

$$0.8 p_{\max} \le p_{hm} \le p_{\max} \quad (7)$$

Calculated fluid flow rate in the hydraulic motor. (l/min)

$$Q_{gm} = \frac{q_{gm} \cdot n_m}{\eta_{gm}} \cdot l / \min \quad (8)$$

Here:

q_{hm} - calculated working volume of the hydraulic motor,

n_{gm} n_m - number of revolutions of the hydraulic motor shaft,

η_{gm} -η_{nm} – hydraulic motor mechanic (HMM)

The calculated rotation frequency of the hydraulic motor shaft is determined by the following formula [4].

$$n_{hm} = \frac{1000 \cdot Q_{hm} \cdot \eta_{nm} \cdot \eta_r}{q_{hm} \cdot i_r} \cdot n_{hm} = \frac{1000 \times Q_{hm} \times \eta_r \cdot \eta_{gm}}{q_{hm} \times i_r}, rpm/\min \quad (9)$$

Here:

Q_{hm} Q_{hm}- calculated fluid flow rate (l/min),

η_{gm} η_{nm} – volumetric efficiency of the hydraulic motor,

q_{hm} q_{hm} - calculated working volume of the hydraulic motor (cm^3/rpm);

i_r - the number of gears of the reducer.

The calculated torque of the hydraulic motor shaft is determined by the following formula (10).

$$M_{hm} = \frac{q_{hm} \cdot P_{hm} \cdot \eta_{nm} \cdot i_r \cdot \eta_r}{2\pi} \; M_{hm} = \frac{q_{hm} \times P_{hm} \times \eta_r \times \eta_{gm} \times i_r}{2\pi}, \; N \times m \quad (10)$$

Here:

q_m- (sm^3/ay), P_m- (MPa) these values are taken from the technical classification of the hydraulic motor.

The calculated power of a hydraulic motor is determined by the following formula (11).

$$N_{hm} = M_m \cdot \omega_m = \frac{\pi \cdot n_m \cdot M_m}{30} \; N_{hm} = M_{hm} \times \omega_{gm} = \frac{\pi \times n_{hm} \times M_{hm}}{30}, \quad (11)$$

Here:

M_{hm} M_m- calculated torque of the hydraulic motor shaft (Nm),

ω_m- calculated angular velocity of the hydraulic motor shaft (rad/sec). The computational results are presented in Table 2 below.

Table 2. 310.28.01.56. Calculation values of the hydraulic motor

Parameters and units	Working volume, cm^3/rpm	Pressure, MPa	Fluid flow, l/min	Number of shaft revolutions, rpm/min	Torque, Nm	Power, kW	Reducer
Definition	q_{hm}	P_{hm}	Q_{hm}	n_{hm}	M_{hm}	N_{hm}	i_r
Values	31,5	16	157,7	1187	274	34,1	4

The calculation results show that when calculating the number of transmissions to the tractor's front hydraulic suspension system with a reducer with 4-type gears arranged in a cylindrical shape, the calculated power of the selected axial piston hydraulic motor of the 310.28.01 brand was N_{hm} =34.1 kW, the calculated torque M_{hm} =274.7 Nm, and the calculated rotation frequency of the hydraulic motor shaft was n_{hm} =1187 rpm/min.

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