General model of hydropneumatic suspension for the grain combine harvester adapter

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Abstract. The paper presents the hydropneumatic suspension mathematical and simulation model development results for the grain combine harvester adapter. The urgency of the work in connection with the hydropneumatic suspensions spread and the need to increase the system efficiency for field relief copying has been substantiated. A computational model of a hydropneumatic suspension including hydraulic cylinders and hydropneumatic shock absorbers connected in parallel to them, including chokes and gas springs has been presented. A system of equations for the hydraulic cylinder piston motion, depending on the hydropneumatic suspension parameters and the number of the connected hydropneumatic shock absorbers has been presented. On the assumption basis, a simplified diagram of the adapter hydropneumatic suspension with back pressure and the corresponding equation system for determining the hydraulic cylinder piston movement have been drawn up. The model includes changing the parameters of external influence on the system, as well as the hydropneumatic accumulator presetting parameters. To assess the developed model’s performance, its simulation model, which was integrated into the well-known simulation model of the grain combine harvester movement, has been compiled. The modeling results of the combine movement on the field with real geometric evenness parameters have been presented. Based on the simulation results, an oscillogram of the hydraulic cylinder piston movement has been shown, depending on the hydropneumatic accumulators’ settings. Conclusions and directions for further research in this area have been presented.

1. Introduction
One of the main problems of agriculture to increase grain production is to increase the grain combine harvesters’ (GCH) efficiency on agricultural areas with complex macro- and micro-relief, which account for more than 16 million hectares of grain crops in the CIS alone [1, 2]. When carrying out harvesting operations in such areas in order to reduce losses behind the adapter, the field relief copying systems (FRCS) have become of the greatest use. The hydropneumatic suspension adapter introduction made it possible to somewhat simplify the FRCS, to improve its manufacturability, however, at the same time, a number of tasks, which boil down to the development of a methodology for calculating such systems have appeared. This methodology is an obligatory component in modern methods of designing machines and equipment and determines the relevance of this work.
2. Main part
Earlier combines with FRCS models worked in the longitudinal (figure 1a) and transverse direction (figure 1b) by ensuring the adapter 1 mobility relative to the combine body 2, the kinematic connection between which was provided by means of a spring drive mechanism.

![Figure 1](image1.png)

**Figure 1.** Scheme for ensuring the longitudinal and transverse mobility of the adapter relative to the combine body

However, at present, FRCS of modern harvesters primarily works parallel connection with hydraulic cylinder 3 and hydropneumatic shock absorbers (HPSA) 4 (figure 2a), including chokes 5 and gas springs 6 (figure 2) due to the technological feasibility. The main FRCS element is an HPSA with variable structure. Variable structure means various HPSA configurations by changing the number of constituent modules of their parameters. Available 5 controlled chokes make it possible to change the HPSA parameters by changing the flow section, as well as the pressure in the HPSA gas cavity.

![Figure 2](image2.png)

**Figure 2.** GCH calculation scheme with hydropneumatic suspension and its HPSA block.

Thus, to ensure the relief copying process, on the one hand, it is required to provide high sensitivity, the system speed, but also the practical complete elimination of harmonic oscillations after removing the load. The solution of such a problem in modern understanding requires the appropriate mathematical and simulation models’ preparation.

It is obvious that to control the output links’ movement in the executive hydropneumatic drive, it is possible to use choke control of the flow rate [3]. For the GCH modern suspension systems with a working medium in liquid form, the formula for calculating the liquid flow rate has the form:

$$Q = \mu \cdot f \cdot \sqrt{\frac{2\Delta p}{\rho}}$$

where $\mu$ and $f$ denote the flow coefficient and flow area of the choke 5 (figure 2); $\Delta p$ defines the pressure drop in the hydraulic cylinder 3 cavity and the cavity and the gas spring 6; $\rho$ is working fluid density.
When studying and calculating the dynamic characteristics of the system under consideration, the hydromechanical processes in it occur with time-varying pressures. The flow rate of the working medium entering the gas spring 6 with volume \( V \) and with time-varying boundaries, according to the continuity equation, can be found in the form:

\[
Q = \frac{dV}{dt} + V \frac{dp}{dt}
\]

where \( p \) – is the medium pressure contained in the volume \( V \), i.e. in a gas spring; \( B \) – is a bulk medium modulus.

Let us draw up a design diagram of a hydropneumatic spring, including hydraulic cylinders, where HPSA variable structure are connected to the rod and piston cavity (figure 3) and introduce the following notation: \( N \) defines HPSA quantity in the rod end piston; \( p_1, p_2 \) – pressure in the piston and rod cavity of the hydraulic cylinder, respectively; \( G_1, G_2, G_{2m-1} \) – are the gas springs connected to the piston cavity of the hydraulic cylinder; \( m = 1, 2, \ldots, N \); \( G_2, G_4, G_{2m} \) – are the gas springs connected to the rod end of the hydraulic cylinder; \( Ch_1, Ch_3, Ch_{2m-1} \) – are the chokes before HPSA \( G_1, G_3, G_{2m-1} \); \( Ch_2, Ch_4, Ch_{2m} \) – are the chokes before HPSA \( G_2, G_4, G_{2m} \); \( p_1, p_3 \); \( p_{ch_{2m-1}} \) define the choke outlet pressure \( Ch_1, Ch, Ch_{2m-1} \); \( p_2, p_4 \); \( p_{ch_{2m}} \) define choke outlet pressure \( Ch_2, Ch_4, Ch_{2m} ; p_g, p_g, p_g \) – define HPSA gas pressure of the piston circuit \( G_1, G_3, G_{2m-1} ; p_g, p_g, p_g \) – define HPSA gas pressure of the rod circuit \( G_2, G_4, G_{2m} \); \( x \) – is the piston movement in the hydraulic cylinder; \( y_1, y_3, y_{2m-1} \) – is the piston movement in the HPSA gas cavity \( G_1, G_3, G_{2m-1} ; y_2, y_4, y_{2m} \) – define the piston movement in the HPSA gas cavity \( G_2, G_4, G_{2m} ; S_1, S_2 \) – show the piston effective area in the piston and rod cavity of the hydraulic cylinder; \( S_{g1}, S_{g3}, S_{g2m-1} \) – show the effective piston area in the HPSA gas cavity of the piston circuit \( G_1, G_3, G_{2m-1} ; S_{g2}, S_{g4}, S_{g2m} \) – show the effective piston area in the HPSA gas cavity of the rod circuit \( G_2, G_4, G_{2m} \).

Figure 3. Design diagram of GCH adapter hydropneumatic suspension with back pressure

For the hydropneumatic suspension system taking into account the expression continuity condition for the perfect fluid (\( Q \)) flow rates in the hydraulic cylinder and HPSA cavities have the form (figure 3):
\[ \dot{Q}_1 = S_1 \cdot \dot{x} ; \]

\[ \dot{Q}_2 = S_2 \cdot \dot{x} ; \]

\[ \ldots \]

\[ Q_{ch,2m-1} = \mu_{2m-1} \cdot f_{2m-1} \text{sign}(p_1 - p_{ch,2m-1}) \cdot \frac{2|p_1 - p_{ch,2m-1}|}{\rho} ; \]

\[ Q_{ch,2m} = \mu_{2m} \cdot f_{2m} \text{sign}(p_2 - p_{ch,2m}) \cdot \frac{2|p_2 - p_{ch,2m}|}{\rho} ; \]

\[ Q_{G,2m-1} = S_{2m-1} \cdot \dot{y}_{2m-1} ; \]

\[ Q_{G,2m} = S_{2m} \cdot \dot{y}_{2m} . \]

The equations for the pressures in the hydraulic cylinder cavities have the form:

\[ \frac{d p_1}{d t} = \frac{Q_1 - Q_{ch} - Q_{ch} - Q_{ch,2m-1} = Q_1 - \sum_{m=1}^{N} Q_{ch,2m-1}}{K_{elast_1}} ; \]

\[ K_{elast_1} = \frac{V_{m_1} + S_1 |x| \text{sign}(x)}{E_{cyl}} ; \]

\[ \frac{d p_2}{d t} = -\frac{Q_2 + \sum_{m=1}^{N} Q_{ch,2m}}{K_{elast_2}} ; \]

\[ K_{elast_2} = \frac{V_{m_2} + S_2 |x| \text{sign}(x)}{E_{cyl}} , \]

where \( K_{elast_1} \) and \( K_{elast_2} \) show the piston and rod cavity elasticity coefficient of the hydraulic cylinder with fluid; \( E_{cyl} \) is the reduced volumetric elasticity modulus of the working fluid in the gas spring; \( V_{m_1}, V_{m_2} \) show the remaining working fluid volume, respectively, in the piston and rod cavity of the hydraulic cylinder when the piston is in extreme positions.

The equations for pressures in HPSA cavities have the form:

\[ \frac{d p_{ch,2m-1}}{d t} = \frac{Q_{ch,2m-1} - Q_{G,2m-1}}{K_{elast_{G,2m-1}}} ; \]

where

\[ K_{elast_{G,2m-1}} = \frac{V_{m_{2m-1}} + S_{2m-1} |y_{2m-1}| \text{sign}(y_{2m-1})}{E_{G,2m-1}} ; \]

\[ \frac{d p_{ch,2m}}{d t} = \frac{Q_{ch,2m} - Q_{G,2m}}{K_{elast_{G,2m}}} ; \]

where

\[ K_{elast_{G,2m}} = \frac{V_{m_{2m}} + S_{2m} |y_{2m}| \text{sign}(y_{2m})}{E_{G,2m}} ; \]
where \( K_{\text{elast},G_{2m-1}} \) and \( K_{\text{elast},G_{2m}} \) show the elasticity coefficient of the cavity with the piston and gas circuits liquid; \( E_{G_{2m-1}} \) and \( E_{G_{2m}} \) – are the reduced volumetric modulus of elasticity of the working fluid in the piston and gas circuits; \( V_{m1}, V_{m2m-1}, V_{m2m} \) show the remaining working fluid volume, respectively, in the gas springs when their pistons are in extreme positions; \( n \) – is a polytropic process indicator; \( p_{0,G_{2m-1}} \) defines gas pressure at a time moment \( t = 0 \) in the HPSA, respectively, of the piston and rod circuits.

The hydraulic cylinder piston’s movement together with the adapter is described by the expression:

\[
m_n \cdot \frac{d^2x}{dt^2} = -p_1 \cdot S_1 + (p_2 \cdot S_2 + F_k - F_f) .
\]

\( m_n \) is the mass of the moving part and working fluid reduced to the hydraulic cylinder piston; \( F_k \) is the external load on the hydraulic cylinder piston; \( F_f \) denotes friction force [4].

HPSA piston movement is described by the equations:

\[
m_{n-p_{2m-1}} \cdot \frac{d^2y_{2m-1}}{dt^2} = (p_{ch_{2m-1}} - p_{G_{2m-1}}) \cdot S_{G_{2m-1}} - F_{fr,2m-1} ;
\]

\[
m_{n-p_{2m-2}} \cdot \frac{d^2y_{2m}}{dt^2} = (-p_{ch_{2m}} + p_{G_{2m}}) \cdot S_{G_{2m}} - F_{fr,2m} ,
\]

where \( m_{n-p_{2m-1}} \) and \( m_{n-p_{2m-2}} \) define the reduced mass of HPSA pistons in the piston and rod cavity respectively; \( F_{fr,2m-1} \), \( F_{fr,2m} \) define friction force in the piston and rod circuits, respectively.

Thus, the system of equations (1-17) appears to be closed - the number of the unknowns coincides with the number of equations. The system of equations (1-17) describes in principle the working processes taking place in the air-hydraulic suspension of the back pressure adapter. The number of equations and unknowns is determined by the number of the hydropneumatic system modules. The unknowns include:

\[
x, y_{2N-1}, y_{2N}, p_1, p_2, p_{ch_{2N-1}}, p_{ch_{2N}}, p_{G_{2N-1}}, p_{G_{2N}},
\]

\[
q_1, q_2, q_{ch_{2N-1}}, q_{ch_{2N}}, q_{G_{2N-1}}, q_{G_{2N}} .
\]

It is obvious, that for \( N = 1 \) the number of equations is 15, for \( N = 2 \) the number of unknowns is 25, for \( N = 3 \) the number of unknowns is 35, etc. If we exclude the auxiliary unknowns from the system of equations and leave only \( x, y_{2N-1}, y_{2N} \), then we obtain the basic system of differential equations according to the Newton’s second law: for \( N = 1 \) – 3 equations; at \( N = 2 \) – 5 equations; at \( N = 3 \) – 7 equations, etc.

The system of equations (1-17) manages to solve the second dynamic problem for the hydropneumatic system under consideration, i.e. for the case when the external force is defined \( F_e(t) \) as a function of time and the initial conditions are given:

\[
x(0), y_{2N}(0), y_{2N-1}(0), \dot{x}(0), \dot{y}_{2N}(0), \dot{y}_{2N-1}(0) .
\]

Then the problem solution will be reduced to the calculation:

\[
x(t), y_{2N}(t), y_{2N-1}(t) .
\]

Solving equations (1-19) for a hydropneumatic suspension adapter with multiple HPSA is a complex task that requires significant computing resources, as well as taking into account a number of uncertainties. In order to simplify the design model, we will consider a model of a hydropneumatic suspension, which has one HPSA on the rod and piston circuits (figure 4), i.e. HPSA back pressure [5].
Figure 4. The simplified design diagram of hydropneumatic suspension with back pressure

If the resistance force and restrictions in the form of stops and gas back pressure are excluded from the calculation, the flow equations will have the form:

\[ Q_1 = S_1 \cdot \dot{x}; \]  \hspace{2cm} (20)
\[ Q_2 = S_2 \cdot \dot{x}; \]  \hspace{2cm} (21)
\[ Q_1 = \mu_1 \cdot f_1 \text{sign}(p_1 - p_3) \sqrt{\frac{2|p_1 - p_3|}{\rho}}; \]  \hspace{2cm} (22)

or

\[ p_1 = p_3 + \text{sign}(\dot{x}) \left( \frac{s_1 x}{\mu_1 f_1} \right)^2 \frac{\rho}{z}; \]  \hspace{2cm} (23)
\[ Q_2 = \mu_2 \cdot f_2 \text{sign}(p_2 - p_4) \sqrt{\frac{2|p_2 - p_4|}{\rho}}; \]  \hspace{2cm} (24)

or

\[ p_2 = p_4 - \text{sign}(\dot{x}) \left( \frac{s_2 x}{\mu_2 f_2} \right)^2 \frac{\rho}{z}; \]  \hspace{2cm} (25)
\[ p_{G1} = p_{0G1} \cdot \left( \frac{v_{0G1}}{v_{0G1} - y_1 s_{G1}} \right)^n = p_{0G1} \cdot \left( \frac{v_{0G1}}{v_{0G1} - x s_{G1}} \right)^n. \]  \hspace{2cm} (26)

From (26) it is seen that \( y_1 \cdot S_{G1} = x \cdot S_1 \), then the consumption conservation law is:

\[ y_1 = \frac{x S_1}{S_{G1}}; \]  \hspace{2cm} (27)
\[ p_{G2} = p_{0G2} \cdot \left( \frac{v_{0G2}}{v_{0G2} + y_2 S_{G2}} \right)^n = p_{0G2} \cdot \left( \frac{v_{0G2}}{v_{0G2} + x S_{G2}} \right)^n; \]  \hspace{2cm} (28)

and

\[ y_2 \cdot S_{G2} = x \cdot S_2; \]
\[
  y_2 = -\frac{x \cdot S_2}{S_{g_2}}
\]  

(29)

It is seen that at \( x > 0, y_1 > 0, y_2 < 0 \) from the static equilibrium position \( x = 0, y_1 = 0, y_2 = 0 \) under static force \( F_0 \). When force \( F_0 \) is applied on the hydraulic cylinder rod the following condition is satisfied:

\[
  F_0 = -p_1 \cdot S_1 + p_2 \cdot S_2.
\]

In this case \( \dot{x} = 0 \).

\[
  p_{G_1} = p_3 = p_{0G_1};
\]

\[
  p_{G_2} = p_4 = p_{0G_2};
\]

\[
  p_1 = p_3; \quad p_2 = p_4;
\]

\[
  G = -p_{G_1} \cdot S_1 + p_{G_2} \cdot S_2.
\]

(30)

Dynamic equations follow from the Newton’s Second Law:

\[
  \begin{align*}
    m_n \ddot{x} &= p_2 \cdot S_2 + G + F_k(t) - p_1 \cdot S_1 - F_{fr} - F_{elast}; \\
    \frac{S_1}{S_{G_1}} \cdot m_n \ddot{x}_1 &= (p_3 - p_{G_1}) \cdot S_{G_1} - F_{fr}; \\
    -\frac{S_2}{S_{G_2}} \cdot m_n \ddot{x}_2 &= (p_4 - p_{G_2}) \cdot S_{G_2} - F_{fr},
  \end{align*}
\]

(31)

where \( F_k(t) \) is the force acting on HPSA; \( F_{fr} \) is friction force; \( F_{fr} \) denote the forces from limit stops.

If the frictional resistance and constraints in the form of stops and gas backpressure is excluded from the calculation, the flow equations will have the form:

\[
  \begin{align*}
    \left( m_n \right) \ddot{x} &= p_2 \cdot S_2 + G + F_k(t) - p_1 \cdot S_1; \\
    \frac{S_1}{S_{G_1}} \cdot m_n \ddot{x}_1 &= (p_3 - p_{G_1}) \cdot S_{G_1}; \\
    -\frac{S_2}{S_{G_2}} \cdot m_n \ddot{x}_2 &= (p_4 - p_{G_2}) \cdot S_{G_2};
  \end{align*}
\]

\[
  p_4 = p_2 + \text{sign}(\dot{x}) \left( \frac{S_2 \cdot \ddot{x}}{S_{G_2}} \right)^2 \cdot \frac{\rho}{z};
\]

(32)

\[
  p_3 = p_1 - \text{sign}(\dot{x}) \left( \frac{S_1 \cdot \ddot{x}}{S_{G_1}} \right)^2 \cdot \frac{\rho}{z};
\]

(33)

\[
  p_{G_1} = f_1(x); \quad p_{G_2} = f_2(x).
\]

(34)

The system of 3 equations (31) with three unknowns \( x, p_1, p_2 \) has a well-known function of time \( F_k(t) \).

The system (31) can be rewritten as:

\[
  \begin{align*}
    m_n \ddot{x} &= p_2 \cdot S_2 + G + F_k(t) - p_1 \cdot S_1; \\
    \frac{S_1}{S_{G_1}} \cdot m_n \ddot{x}_1 &= \left( p_1 - \text{sign}(\dot{x}) \left( \frac{S_1 \cdot \ddot{x}}{S_{G_1}} \right) \right)^2 \cdot \frac{\rho}{z} - f_1(x); \\
    -\frac{S_2}{S_{G_2}} \cdot m_n \ddot{x}_2 &= \left( p_2 + \text{sign}(\dot{x}) \left( \frac{S_2 \cdot \ddot{x}}{S_{G_2}} \right) \right)^2 \cdot \frac{\rho}{z} - f_2(x).
  \end{align*}
\]

(35)

We exclude the unknowns \( p_1, p_2 \) from this system. To do this, we multiply the second equation by \( S_1 \) and divide by \( S_{G_1} \). The third equation is multiplied by \( S_2 \) and divided by \( S_{G_2} \). So, the following equation can be found:
Next, we add to the first equation both parts of the first equation and both parts of the second equation of the system (14). As a result, we get one equation of the motion:

\[
\ddot{x} \cdot M + m_k \cdot \frac{S_1^2}{S_{G_1}^2} + m_2 \cdot \frac{S_2^2}{S_{G_2}^2} = G + F_k(t) - \text{sign}(\dot{x}) \left[ \frac{S_1}{\mu_1 \cdot f_1} \right]^2 \cdot S_1 \cdot \frac{\rho}{2} - f_1(x) \cdot S_1 - \\
- \text{sign}(\dot{x}) \left[ \frac{S_2}{\mu_2 \cdot f_2} \right]^2 \cdot S_2 \cdot \frac{\rho}{2} + f_2(x) \cdot S_2
\]

This equation can be written in a more compact form:

\[
\ddot{x} \cdot M = G + F_k(t) - \Phi(x) - \dot{x}^2 \text{sign}(\dot{x}) \cdot B,
\]

where

\[
M = m_n + m_1 \cdot \frac{S_1^2}{S_{G_1}^2} + m_2 \cdot \frac{S_2^2}{S_{G_2}^2};
\]

\[
\Phi(x) = -f_2(x) \cdot S_2 + f_1(x) \cdot S_1;
\]

\[
B = \left[ \frac{S_1^3}{\mu_1^2 \cdot f_1^2} + \frac{S_2^3}{\mu_2^2 \cdot f_2^2} \right] \cdot \frac{\rho}{2};
\]

Taking into account the direction of the piston movement up or down, it is possible to write

\[
\ddot{x} + n\dot{x} |\dot{x}| + \Phi^*(x) = \frac{G + F_k(t)}{M},
\]

where \( \Phi^*(x) = \frac{\Phi(x)}{M} \); \( n = \frac{B}{M} \).

The equation (38) should be solved with the initial conditions: \( x = 0; \dot{x} = 0 \) by \( t = 0 \). This, according to the studied literature, is the second problem of the dynamics of a material point, i.e. the problem \( x = x(t) \) definition solution. The system of equations (20-39) or the equation (39) can be solved numerically or analytically. The analytical solution can be split into two problems: about damped free oscillations, or about the forced oscillations of the system.

3. Discussion

Thus, the developed mathematical model makes it possible to determine the movement of the moving parts of the hydropneumatic suspension and, in contrast to the known models [6-10], includes several GPU modules of the piston and rod cavity, and also allows taking into account the individual parameters of each throttle and gas spring. For the hydropneumatic drive FRCS developed model’s performance assessment the theoretical research has been carried out. In the research frameworks the system of equations (20-39) was transformed into a simulation FRCS model in the Matlab Simulink modeling environment and was integrated into the previously developed by the author simulation model of a grain combine harvester with a wheel propeller [11, 12]. This model reproduces the main dynamic loads acting on the body of the combine [13-15], and corresponds to the generally accepted approaches in transport engineering [1029/1/012115].
engineering [16-18]. Modeling was carried out for the cases of a combine harvester movement on winter wheat stubble at a speed 5 km/h. For modeling according to the work [19] the following geometrical parameters of the supporting micro-profile surface are taken: maximum height of unevenness – 0.074169 m; minimum unevenness height – 0.049164 m; standard deviation – 0.024431 m; dispersion – 0.000597 m². The model adopted the geometrical and mass-dimensional parameters of the combine harvester RSM Torum 785. The simulation model of the hydropneumatic suspension with back pressure corresponded to the calculated one in Fig. 4 and had the following parameters: piston diameter – 63 mm, rod diameter – 90 mm. HPSA had the following parameters: volume – 750 cm³; gas cavity pressure \( p_G \) – from 3 till 4 MPa.

As a result of simulation, the hydraulic cylinder piston movements oscillograms for various options for adjusting the hydropneumatic suspension have been built. So, for the case when the pressure in the gas cavity HPSA is 3 mPa, the amplitude of the hydraulic cylinder piston movement is 0.07 m. With magnification \( p_G \) the expected amplitude of the hydraulic cylinder piston movement decreases to 0.04 m (figure 5).

![Figure 5. The oscillogram obtained from the results of the hydraulic cylinder piston displacements simulation modeling for the cases when \( p_G \): 1- 3 mPa; 2 – 5 mPa](image)

In general, the obtained values of the hydraulic cylinder displacement correspond to the experimental measurements results on the similar combine harvesters [20].

4. Conclusion

1. The obtained mathematical model of the HSPA adapter hydropneumatic suspension provides the ability to define the technical appearance at early stages of HPSA design, optimize their parameters based on the layout, mass-dimensional and inertial characteristics of the combines. The proposed mathematical model can be used to design FRCS, as well as for motion stabilization systems based on the inertial damping principle.

2. The developed mathematical and simulation model of the adapter hydropneumatic suspension can be integrated as a subsystem of the well-known simulation models of self-propelled transport and technological machines and equipment. The performed assessment of the model performance shows sufficient convergence of the calculated data and simulation results. The model’s improvement should be carried out in subsequent works in terms of taking into account the friction forces between the joined system elements, hydrodynamic processes in the working fluid, as well as studying the HPSA structure effect on system efficiency.

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