Kinematic and force analysis of a bucket frontlift

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Abstract. Theoretical researches have been carried out and the obtained mathematical expressions characterize a periodic cycle loader with a center pivot frame articulation as a moving mechanical system. The type and the nature of movement of the loader operating element have been considered. The design parameters and the operating modes affect crucially on the interaction between the bucket and the cargo. We have obtained the interdependence between the velocity and the digging depth of a bucket frontlift into the bulk cargo, as well as the kinetic energy of a bucket frontlift as a system, in its initial position. During the force analysis it has been determined the impact of the studied factors on the tractive power of the frontlift and the total digging force, which depends on the tangent tractive power, rolling resistance force and frictional sliding force of a frontlift.

Bucket frontlifts designed on the basis of energy-saturated pneumatic-tired tractors of different classes with a rigid and center pivot frame articulation are widely used in various industries for bulk cargo loading. Their efficiency is possible due to the reasonable operating and designed parameters of a frontlift in terms of productivity and energy intensity of the operating process.

Let us consider the impact of the digging velocity on the process of cargo picking up. When the bucket is dug into the cargo mass, the digging force $F_{vn}$, is consumed for the self-propelled movement of a frontlift $F_p$, frictional sliding $F_b$ and overcoming of the resistance forces, developing as a result of the interaction of the bucket and the material $F_c$ (figure 1). The bucket digging into the cargo until the digging force becomes more or equal to the sum of the forces $F_p$, $F_b$ and $F_c$

With the help of such a parameter as the bucket digging depth into the cargo mass $L$, the quality assessment of this process was carried out. To find out the interdependence of the velocity and the digging depth of a bucket, it is necessary to use the theorem of the kinetic energy change of the system of objects [1]:

$$T_1 + T_2 = \sum A_i,$$  \hspace{1cm} (1)

where $T_1$ and $T_2$ – the kinetic energy of the system in initial and final states;

$\sum A_i$ – external force energy, applied to the system on the designed motion.

If we agree that the system is at rest in the final state (figure 1), then the $T_2 = 0$, expression (1) will take the following form [2]:

$$T_1 = \sum A_i.$$ \hspace{1cm} (2)

In the initial state the kinetic energy of the system $T_1$ will be determined as:
\[ T_1 = T_{kov} + 2T_m + 2T_q \]  
\( T_{kov} \) - the kinetic energy of the bucket, J;  
\( T_m \) - the kinetic energy of the motored axle wheel, J;  
\( T_q \) - the kinetic energy of the load axle wheel, J. The kinetic energy of the bucket \( T_{kov} \), moving progressively:

\[ T_{kov} = 0.5m_{kov}v^2, \]  
where \( m_{kov} \) - the bucket mass, kg;  
v - the digging velocity, m/s.

The kinetic energy of the motored axle wheel \( T_m \), that makes a plane-parallel motion \([3, 4]\):

\[ T_m = 0.5m_m v^2 + 0.5I_m \omega_m^2, \]  
where \( m_m \) - the mass, that occurs on one motored axle wheel, kg;  
\( I_m \) - the inertia moment of one motored axle wheel relating to the axle, coming through the centre of its mass, kg\( \cdot \)m\(^2\);  
\( \omega_m \) - the angular velocity of motored axle wheels relating to the instantaneous centre of velocity (point B), rad/s.

The inertia moment \( I_m \) is determined in the following way:

\[ I_m = 0.5m_m R_m^2, \]  
where \( R_m \) - the rolling radius of motored axle wheels, m.

The angular velocity of motored axle wheels relating to the instantaneous centre of velocity:

\[ \omega_m = v/R_m. \]  
Substituting the values of \( I_m, \omega_m \) in equation (5), we determine the kinetic energy of the motored axle wheel \([3, 5]\):

\[ T_m = 0.75m_m v^2. \]  
The kinetic energy of the load axle wheel, that makes a plane-parallel motion:

\[ T_q = 0.5m_q v^2 + 0.5I_q \omega_q^2, \]
where \( m_q \) - the mass that occurs on one load axle wheel, kg;

\( I_q \) - the inertia moment of one load axle wheel relating to the axle, coming through the centre of its mass, kg·m²;

\( \omega_q \) - the angular velocity of load axle wheels relating to the instantaneous centre of velocity (point C), rad/s.

The inertia moment \( I_q \) is determined in the following way:

\[
I_q = 0.5m_qR_q^2, \quad (10)
\]

where \( R_q \) - the rolling radius of load axle wheels, m.

The angular velocity of load axle wheels relating to the instantaneous centre of velocity

\[
\omega_q = v/R_q. \quad (11)
\]

Substituting the values of \( I_q, \omega_q \) in equation (9), we determine the kinetic energy of the load axle wheel:

\[
T_q = 0.75m_qv^2. \quad (12)
\]

As a result, the kinetic energy of a frontlift as a system is determined in its initial state as the following [6]:

\[
T_1 = v^2[0.5m_{kov} + 1.5(m_m + m_q)]. \quad (13)
\]

Let us determine the sum of the external forces applied to the system on the designed movement. External forces, \( H \) influence the system (figure 2): \( F_{kov} = m_{kov}g \) — the gravity force of the frontlift bucket; \( F_m = m_mg \) — the gravity force, acting on the motored axle; \( F_q = m_qg \) — the gravity force, acting on the load axle; \( F_{pm} \) and \( F_{pq} \) — the rolling resistance forces of motored and load axle wheels respectively; \( F_e \) — the sum of resistance forces, acting on the bucket; \( F_{bm} \) and \( F_{bq} \) — the forces, consumed for the overcoming of frictional sliding of motored and load axes respectively; \( F_{km} \) and \( F_{kq} \) — the forces, developing on the motored and load wheels under the influence of operating torque respectively; \( g \) — free fall acceleration, m/s²[4, 7].

The rolling resistance force of the frontlift drive wheels:

\[
F_p = F_{pm} + F_{pq} = f m_mg + f m_qg = f g(m_m + m_q), \quad (14)
\]

where \( f \) — the coefficient of rolling resistance of pneumatic wheels.

\[\text{Figure 2. The schema for the determination of the external force energy, acting on the system.}\]
The forces, consumed for frictional sliding of drive axles:
\[ F_{bm} = \delta F_{km} \]  
\[ F_{bq} = \delta F_{kq} \]  
(15)  
(16)

where \( \delta \) - the coefficient of frictional sliding.

The total force, consumed for the frontlift frictional sliding:
\[ F_b = \delta F_k \]  
(17)

where \( F_k \) – the equivalent force resulting from the torque moment, \( F_k = F_{km} + F_{kq} \).

In this case the work of forces \( F_m, F_q, F_{kov} \) will equal to zero, because their vectors are perpendicular towards the bucket movement [8]. Thus, the total work of external forces, acting on the system is equal to, \( J \):
\[ \sum A_i = A_{Fc} + A_{Fp} + A_{Fk} + A_{Fb} \]  
(18)

The work of the total digging resistance forces:
\[ A_{Fc} = F_c \cdot L. \]  
(19)

The work of rolling resistance forces:
\[ A_{Fp} = F_p \cdot L. \]  
(20)

The work of forces, occurring on the wheels due to torque moments:
\[ A_{Fk} = -F_k \cdot L. \]  
(21)

The work of forces, consumed for the tractor frictional sliding:
\[ A_{Fb} = F_b \cdot L. \]  
(22)

Using the values \( A_{Fc}, A_{Fp}, A_{Fk}, A_{Fb} \) we determine the external force energy, acting on the system in its designed motion [4]:
\[ \sum A_i = L \cdot (F_c + F_p + F_k - F_b). \]  
(23)

As a result, we get the interrelation between the digging velocity \( v \) and the digging depth of the bucket \( L \):
\[ v^2 \left[ 0.5m_{kov} + 1.5(m_m + m_q) \right] = L \cdot (F_c + F_p + F_k - F_b). \]  
(24)

Analyzing the equation (24), we conclude:
\[ F_{vn} = F_k - F_p - F_q + v^2 \left[ 0.5m_{kov} + 1.5(m_m + m_q) \right] / L. \]  
(25)

In other words, the digging force \( F_{vn} \) depends on the value of the frontlift tangent tractive power, the rolling resistance force, the force consumed for frictional sliding and the accumulated kinetic energy during the acceleration.

For the tractor with the wheel arrangement 4x4 and the blocked drive, the tangent tractive power on its axles will be distributed unevenly [5]:
\[ F_{km} = M \eta_{tr} \eta_{tr} / R_m; \]  
(26)
\[ F_{kq} = M \eta_{tr} \eta_{tr} / R_q, \]  
(27)

where \( M \) — torque rating of the engine, Nm;  
\( \eta_{tr} \) — transmission ratio;
\( \eta_{tr} \) — the coefficient of mechanical loss in transmission; 
\( R_m, R_q \) — radii of motored and load axle wheels, m. The rolling resistance force is determined according to equation (14). The force, occurring during the acceleration of the frontlift:

\[
F_v = v^2 \left[ 0.5 m_{kov} + 1.5 (m_m + m_q) \right] / L. 
\]  
(28)

The difference of the tangent tractive power and the forces, consumed for frictional sliding and rolling resistance, results in the tractive power of the frontlift:

\[
F = F_k - F_p - F_b. 
\]  
(29)

Using the obtained equation, it is possible to determine the impact rate of the studied factors on the tractive power of the frontlift and the total digging force [4]. The digging force, developed by the frontlift, depends on the tangent tractive power of the frontlift, rolling resistance force, frictional sliding force and the bucket digging depth into stockpile. The digging depth depends on the digging velocity and mass distribution, acting on the frontlift axles.

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