CONTROL OF HYDROSTATIC TRAVEL DRIVE SYSTEM IN EARTH MOVING MACHINES

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ABSTRACT

Application of hydrostatic travel drive system in tractor earth-moving machines enables a precise control of digging process according to given requirements and limits. A paper presents some problems connected with this subject which have been chosen by the authors as the most important ones.

1. INTRODUCTION

Most of the tractor earth-movers such as bulldozers, loaders, scrapers etc. operate in cycling and make during a working cycle (which lasts from a few to a dozen or so seconds) a few relatively short distances. Such a cycling work requires almost continuous and adequately correlated control of direction and parameters of power applied to wheels or tracks and piston rods of hydraulic cylinder enabling the changes of attachment position in respect of chassis.

Nowadays used travel drives which contain a single or double-stage torque converters of non-controllable characteristic and a power shift transmission are difficult to control. Hydrostatic transmissions (Fig. 1) are much advantageous as regards the control possibility. They make it possible to automatically control machine operation according to various programs and using various techniques, especially micro-processor ones.

The main purpose of automation is increase in machine efficiency which is understood as energy-consumption, hourly production, and accuracy of their working
processes. The basic methods and conditions for such purpose attainment are described in our paper.

2. OPTIMUM ENGINE LOADING

The basic condition which makes it possible to reduce a fuel consumption is such engine loading, during the working process, is such that probability of engine operation with minimum specific fuel consumption would be the highest (Fig. 2).

![Fuel consumption characteristic of SW 680 engine](image)

**Fig. 2.** Fuel consumption characteristic of SW 680 engine

Such a condition can be taken as the following control task:

$$\left|M_{s,\text{econ}} - \left(\Sigma M_p + \Sigma M_{po} + M_d + J_1 \frac{d\omega_s}{dt}\right)\right| \to 0$$

(1)

where $M_{s,\text{econ}}$ is the optimum engine load, $\Sigma M_p$, $\Sigma M_{po}$ are the total engine loads by the pumps of travel and attachment drives, respectively, $M_d$ is the engine load with additional mechanisms, $\omega_s$ is the angular velocity of engine shaft, $J_1$ is the mass moment of inertia of the mobile elements of the system drive.

The task (1) can be accomplished by the control of fuel charge or by the control of pumps loading as a result of adequate change of the pump delivery according to the dependence:

$$M_p = \frac{1}{2\pi} \frac{\varepsilon_p \cdot q_p \cdot \Delta p_p}{\eta_{\text{mp}}^{\text{mp}}}$$

(2)

where $\varepsilon_p$ is the setting parameter of pump, $q_p$ is the pump displacement, $\Delta p_p$ is the pressure increase, $\eta_{\text{mp}}$ is the pump mechanic efficiency.

The fuel charge is controlled according to $\omega_s = \text{const}$ in order to reduce the inertia losses. The basic method to accomplish this task is the delivery pumps control. The flow controller can respond directly to the change of total pump pressure of the
travel and attachment systems or can respond to the change of engine shaft speed, caused by such pressure changes. In practice the latter method can be insufficiently accurate as a direct method because the changes of the engine speed, operating according to the control characteristic, are relatively small even in case of significant load changes.

3. EFFECTIVE FORMATION OF TRAVEL DRIVE LOAD

The power balance of travel drive with the \( j=1\ldots n \) drive wheels or tracks with hydrostatic transmission (Fig. 1) may be expressed as:

\[
\omega_s \sum_j M_{plj} = \sum_j \frac{1}{\eta_{ij}} T_j v_j
\]

where

\[
T_j = P_{njj} v_j R_j = (\phi_j v_j) R_j \leq (\phi_{max} v_j) R_j
\]

where \( P_{njj} \) is the gross traction, \( T_j \) is the net traction, \( R_j \) is the load on wheel or track, \( \eta_{ij} \) is the efficiency of travel drive, \( f_j \) is the rolling resistance factor, \( \phi_j \) is the traction factor, \( v_j \) is the traction speed, where:

\[
v_j = v_{th} (1 - \delta_j) = \frac{r_j \omega_j}{i_m} (1 - \delta_j)
\]

where \( v_{th} \) is the theoretical traction speed, \( r_j \) is the radius of drive wheel or sprocket, \( i_k \) is the kinematic transmission ratio of hydrostatic drive, \( i_m \) is the transmission ratio of mechanical drive part, \( \delta \) is the wheel or track slip, \( \eta_{ij} \) is the travel drive efficiency.

The travel drive efficiency can be described as:

\[
\eta_{ij} = i_k i_d \eta_m \eta_{rj}
\]

where \( i_d \) is the dynamic transmission ratio of hydrostatic drive, \( \eta_m \) is the efficiency of gearbox, \( \eta_{rj} \) is the efficiency of co-operation wheel or track with soil.

The efficiency of co-operation wheel or track with soil is:

\[
\eta_{rj} = \frac{T_j v_j}{P_{njj} v_j} = \frac{T_j}{P_{njj} (1 - \delta_j)} = \frac{(\phi_j v_j) R_j (1 - \delta_j)}{\phi_j}
\]

Considering inertia of drive system and neglecting its elasticity and damping, the driving force \( P_n \) can be expressed as the following relationship:

\[
P_n = \frac{M_{plj}}{r_j} \cdot i_d \cdot i_m \cdot \eta_m - \frac{1}{r_j} \cdot \frac{d\omega_{hy}}{dt} \cdot \left[ i_m \cdot \left( i_d \cdot J_1 + \frac{d\omega_j}{d\omega_2} + J_2 \right) + J_j \right]
\]

where \( \omega_{hy} \) is the angular velocity of hydraulic motor, \( J_1 \) is the mass moment of inertia of mobile elements reduced on hydraulic motors shaft, \( J_j \) is the mass moment of inertia of running gear reduced on sprocket axis.

It results from the analysis of the relationships (6), (7), and (8), assuming \( \eta_m = \text{const} \), that the smallest losses of the transmitting drowbar-pull power in case when:

1) the engine operates with the fixed speed - the losses of power for acceleration of running gear and drive system are minimum;
2) the setting parameters of the pumps \( \varepsilon_p \) and motor \( \varepsilon_m \) of the hydrostatic transmission are controlled according to the hyperbolic characteristic of transmission ratios \( i_{ij} i_{dj} = \text{const} \).
Fig. 4. Horizontal components of load resistances of SG-15 bulldozer during ground mining for a) flat manner of filling, b) filling with continuous lift of blade, $W_a$ is the cut resistance, $W_{hx}$ is the resistance of the ground displacement up the bulldozer blade, $W_p$ is the resistance of the pile pushing, $W_i$ is the rolling resistance, $l$ is the cut profile, $l$ is the length of machine travel during filling process.

A productivity of the tractor machine depends mainly on the operating travel speed which is limited not only by the engine power but also by abilities of an operator.
3) the running gears operate in the slip condition and dependence $\phi(1-\delta) = \max$ (Fig.3).

![Graph showing the relationship between $\phi$, $\phi(1-\delta)$, and slip $\delta$.]

Fig.3. Tyre traction factor $\phi$ for firm (1) and soft (2) soil vs. slip $\delta$

The above conditions can be relatively easy satisfied if there is a possibility to accomplish the machine working process for the fixed attachment loads (Fig.4). Then, the travelling speed can be calculated from the equation:

$$\sum_j P_{nj} - \sum W_x - W_j - m \cdot \frac{dv}{dt} = 0 \quad (9)$$

where $\Sigma W_x$ is the total horizontal working resistance component, $m$ is the mass of machine.

The rolling resistances of the machine travelling on the surface inclined at the angle $\alpha = 0$, assuming no air resistances, are equal to:

$$W_j = \sum f_i \cdot R_j = f \cdot (m \cdot g + W_y) \quad (10)$$

where $W_y$ is the vertical component of operate resistance.

4. SYNCHRONIZATION AND CONTROL CORRELATION

The wheels or tracks driving by means of separate hydrostatic transmissions of adequately low power significantly increases the machine mobility [8]. However, it causes a necessity to control a synchronization of the settings $\varepsilon_i$ and $\varepsilon_j$ in order to obtain the same or adequately diversified travelling speeds of each track (5) which ensure the fixed travelling path. Therefore, one question appears: should the synchronization system react only to the speed of hydraulic motors $\omega_i$ or to the slip effect $\delta$ too.
who must follow up the control of attachment, during the excavation process according
to technical conditions of the performed works. The control of travel drive and piston
rod of attachment in the general case should be correlated in respect of optimum
loading of the engine, according to the dependence (1) as well as in respect of the
required absolute speed \( v_o \) of the attachment:

\[
\vec{v}_o = \vec{v} + i_o \cdot \vec{v}_c
\]  

(11)

where \( v \) is the real travel speed of tractor, \( v_o \) is the speed of attachment cylinder, \( i_o \)
is the kinematic ratio of attachment.

For the effective performance of the correlation task, the volumetric or
trotting regulation in the power system of attachment is necessary

5. CONTROL FOLLOW UP

The hydrostatic transmissions are significantly stiffer than the transmissions
with torque converter. Considering this fact, an adequate follow up the settings of
hydraulic unit is necessary. Insufficient follow up causes high overloads in the
transmission (fig.5). It also influences disadvantageously on the accuracy of machine
operation. Therefore, one of the most important problems in design of hydrostatic
travel drive is a selection of dynamic parameters of the main hydraulic unit and their
controllers.

![Graph](image)

Fig.5. Theoretical (Q) and real (Q) delivery for linear pressure increase (p) for
controller witch time constant \( \tau = 0.25 \) s, \( k_p \) is the overload ratio

The required control of the hydraulic transmission during the working process
can be approximately estimated on the basis of the traction ratio of hydrokinetic drive
(Fig.6) which automatically change under the load condition. However, the stepless
transmission ratios \( i_k \) and \( i_q \) of hydrostatic drive are:
Fig. 6. Diagram of torque converter parameters $i_k$, $i_d$ and slip $\delta$ for work cycle of loader

where $\eta_{vp}$, $\eta_{vs}$, $\eta_{mhp}$, $\eta_{mhs}$ are the volumetric and mechanic efficiencies of pump and motor, respectively. It should be noticed that the chart for hydrostatic transmission relation $i_6(i_k)$ is of hyperbolic shape while for the hydrokinetic drive the chart is linear. Therefore, the ranges of transmission ratio changes in these both cases will be different.

6. CONCLUSION

The effective control of the hydrostatic travel transmission of the earthmovers is complicated task of dynamic multi-criterion optimization which requires a continuous or enough repeated sampling and processing from the point of view of searching for the optimum decisions.

An accomplishment of this task, only in respect of one criterion, is too difficult for operator and additional automation-aided system using micro-processor is necessary. However, a character of the working processes of the earth-movers is such complicated that full automation of their control is impossible in the nearest time. It is not possible to applied the mechatronic systems instead a man because of the necessity of taking decision in the case of incomplete information. Thus, the main research task is an optimum selection of the control activities for operator and automatic systems in dependence on technology and operating conditions of earth-movers.
Serching for the best solutions of this task requires to work out the new method of analysis and synthesis of hydrostatic drive control systems for tractor earth-movers.

REFERENCES


