A Reverse Driving Control Method for Hydrostatic Double-drum Vibratory Rollers

Xin Wang
Key Laboratory for Highway Construction Technology and Equipment of Ministry of Education, Chang'an University, Xi'an, China
Email: wangxin_0731@126.com

Ruiying Zhao
Key Laboratory for Highway Construction Technology and Equipment of Ministry of Education, Chang'an University, Xi'an, China
Email: zhaoruiying@sohu.com

***Abstract*—A reverse driving control method for hydrostatic double-drum vibratory rollers is devised in this paper. Based on the analysis of dynamic characteristics of stopping process, the relationship equation between engine speed and pump displacement is established. To restrict the peak value of engine overspeed within the allowable region, an offline parameter determination approach complied with the requirements of average stopping deceleration is developed. The online control method using the selected parameter is presented. Approaches are derived to determine the two key parameters of the control methods, i.e., the critical deceleration value and the coefficient of relationship equation of the engine friction torque and the engine speed. The control method can also be applied to one type of hydrostatic machinery with their engine throttle position unchangeable in stopping process in addition to the double-drum vibratory roller.**

***Index Terms*—double-drum vibratory roller, reverse driving, closed loop hydrostatic transmission, decelerating and stopping process, engine overspeed, engine friction torque**

I. Introduction

For the mobile machinery driven by closed loop hydrostatic system, the power passing path is engine-pump-motor-wheel in traveling or working situation. However, in the case of decelerating or stopping, if the decreasing of pump displacement is too quick, the motor will act as a pump and the pump will act as a motor under the inertia force. This situation is called reverse driving.

When reverse driving happens the power passing path reverses to wheel-motor-pump-engine. Negative load is brought to the engine and make it overspeed. The kinetic energy is absorbed mainly by the engine in the form of friction torque and dissipated as heat finally.

Reverse driving is an intrinsic property for closed loop system, which enables the system's natural braking ability. Nevertheless, for the energy transmitted inversely in the reverse driving process, the function of pump and motor will exchange with each other, which may lead to overspeed of both pump and engine. As the reverse driving power increases, the enhanced overspeed energy may cause huge damages to both the engine and pump. At the same time, frequent appearance of peak pressure and opening actions of relief valve caused by reverse driving energy will increase the hydraulic oil temperature.

Closed loop hydrostatic driving systems are broadly applied in modern double-drum vibratory rollers. For this machine, reverse driving may appear in some cases of traveling or construction. One common case is that the pump displacement is regulated to zero in a very short time from a big initial value, e.g., the emergency brake button is operated or the engine throttle is relieved suddenly. Another case is that the machine travels downhill continuously.

The possible damages to double-drum vibratory roller are as follows:

(1) The reverse driving energy makes the engine and the pump connected overspeed and shortens their lives.

(2) For the pump displacement is regulated to zero in a short time, the hydraulic system peak pressure appears frequently and the relief valve opens frequently, which will lessen the system efficiency.

(3) In the stopping process, the two steel wheels may be locked by reverse driving torque, and the compacted material will be pushed to form bumps or shapes like fish backbone, which will affect the compacting quality.

Recently, more attentions were drawn to the harmfulness of reverse driving, but the theoretical achievements still lack. A comprehensive study and efficient methods are needed to improve the stopping performance of hydrostatic machines.

In practice, in order to lessen the damages of reverse driving, some simple measures are taken as a temporary

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solution, such as regulating the pump displacement slowly, or setting the motor displacement to a small value in stopping process.

The application of these measures often lack theoretical instructions. For the first measure, the regulation amount of pump displacement is only determined by repeated tests of the machine, while the second one is to sacrifice the stopping distance to reduce the negative influences of reverse driving. Both of these methods lack generality so that the effect is not ideal.

Utilizing the control method to lessen the negative affect brought by reverse driving is one of the possible methods. Satisfactory results will be achieved at minimum cost using such methods. As in [1], a reverse driving control method is devised based on the ideal engine speed curve design, which is a try to solve the reverse driving problem of double-drum vibratory roller by automatic control techniques.

In this paper, the reverse driving control method for double-drum vibratory roller is devised on the basis of parameter regulation of transmission system. This method is also applicable for one type of hydrostatic mobile machinery of which the engine throttle position unchangeable in stopping process, such as a hydraulic asphalt paver.

II. The Appearance of Reverse Driving

Through traveling tests of double-drum vibratory roller, the appearance of reverse driving can be recorded by the parameter changing process.

Fig. 1 to Fig. 5 show the changes of a group of parameters of a double-drum vibratory roller in its stopping process.

The engine speed change is shown in Fig. 1, it can be seen from which that the engine overspeed exceeds by about 15 percent of the allowable value.

Fig. 2 to Fig. 4 show the inlet, outlet and difference pressure of pump, respectively. The negative pressure implies the passing path of reverse driving energy.

Fig. 5 shows the velocity of the machine. The stopping deceleration can be calculated from the velocity.

III. Principle of Reverse Driving Control

In reverse driving process, an inverse torque is built on the motor preventing the machine to move. Although such braking ability is one of the advantages of closed loop system, the negative affect may prevail if reverse driving becomes severe and the engine speed driven exceeds the design limit.

The purpose of reverse driving process control is to ensure the engine speed not to exceed the limit. Due to the requirement of stopping distance, it is impractical only to lengthen the releasing time of reverse driving energy. The energy must be dissipated in specified time while the engine overspeed is controlled within permission.

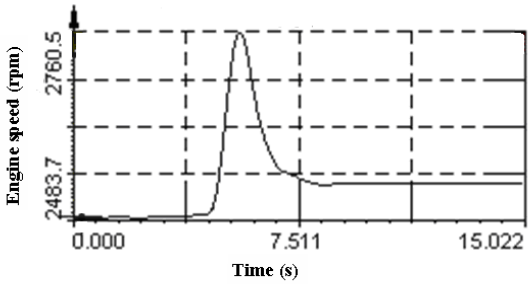

Figure 1. Engine speed plot in reverse driving process.

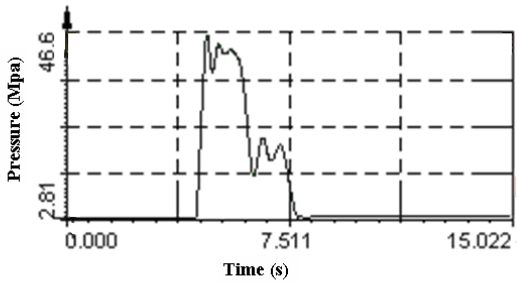

Figure 2. Pump inlet pressure plot in reverse driving process.

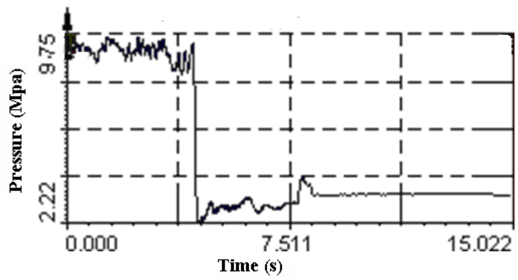

Figure 3. Pump outlet pressure plot in reverse driving process.

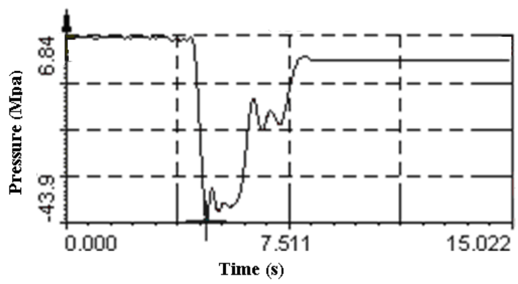

Figure 4. Pump pressure difference plot in reverse driving process.

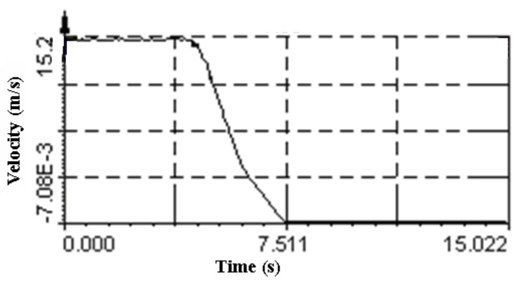

Figure 5. Traveling velocity plot in reverse driving process

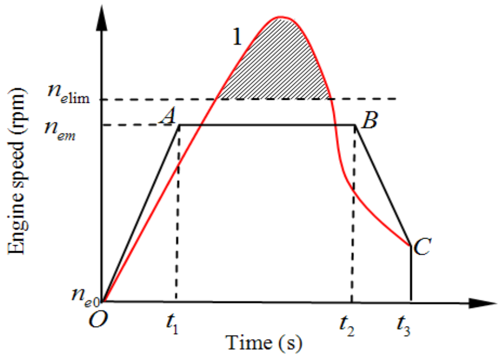

Figure 6. Engine plot designed for reverse driving control

Hence, if the pump displacement control law can be designed to get an ideal engine speed curve, the engine speed will increase to an allowable peak value and then reduce to a steady value. In this control process, the average stopping deceleration needs to be large enough to meet the requirement of stopping distance. By using this method our goal of stopping the machine steadily and quickly can be achieved.

The principle of reverse driving control is shown in Fig. 6. The engine speed plots without control and with control are shown by curve 1 and OABC, respectively. From plot 1 we see that the shadow part above n_{elim} which is the maximum allowable engine speed, is the overspeed region.

Curve OABC is designed as an ideal one by which the engine speed is controlled according to a given process without unallowable overspeed.

In fact, there is no reduction of reverse driving energy. The energy is just absorbed more averagely in the whole stopping process instead of consumed in a short time. If the engine speed can be controlled through the pump displacement regulation, the whole control would be realized. This idea is the essential point of the control method.

IV. Relationship Between Engine Speed and Pump Displacement

A. Assumptions

In order to establish the relationship of engine speed and pump displacement, it is necessary to analyze the kinetic and dynamic characteristics of reverse driving process to set up the model. To simplify the modeling process without loss of generality of the analysis result, the following assumptions are imposed.

A1) The engine throttle is unchangeable in the machine stopping process. The engine speed of reverse driving process is denoted by

$$n_e(t) = n_{e0} + \Delta n_e(t) \tag{1}$$

where $n_e(t)$ is the engine speed(rpm); n_{e0} the initial value of engine speed of reverse driving process(rpm); $\Delta n_e(t)$ the increment of engine speed driven by reverse power(rpm).

A2) The friction torque increment of engine is linear with the increment of engine speed driven by reverse power, which is denoted by

$$\Delta M_e(t) = \lambda \Delta n_e(t) \tag{2}$$

where ΔM_e is the friction torque increment of engine(Nm); λ is a constant coefficient.

A3) The pump is an electro-proportional pump and all the motors are fixed displacement motors.

A4) The double-drum vibratory rollers does NOT slip in reverse driving process.

B. Dynamic Equation of Reverse Driving Process

1) Torque equation for motor shaft output end

By Newton's laws of motion, the torque equation for motor shaft output end may be given by

$$\Sigma M(t) = M_Z(t) + M_R(t) \tag{3}$$

where $\Sigma M(t)$ is the moment of interia of motor(Nm); $M_Z(t)$ the traveling resistance moment(Nm); $M_R(t)$ the reverse driving moment of motor(Nm).

$\Sigma M(t)$ is defined by

$$\Sigma M(t) = mRa(t) = (m_0 + m_J)Ra(t) \tag{4}$$

where $a(t)$ is the traveling deceleration of machine(m/s^2); m_0 the mass of machine(kg); m_J the equivalent translation mass of machine transformed by the moment of inertia (kg); m the equivalent total mass of the machine(kg), and $m = m_0 + m_J$; R the wheel dynamic radius(m).

The moment of resistance is defined by

$$M_Z(t) = \mu m_0 gR \tag{5}$$

where μ is the traveling resistance coefficient, which is not equal to the friction coefficient. In fact, it is a synthetical coefficient including the rolling resistance, frictional resistance, and wind frictional resistance.

The reverse driving moment is given by

$$M_R(t) = \frac{\eta_{mm} q_m \Delta p}{2\pi} \tag{6}$$

where Δp is the pressure of hydraulic system(Mpa); q_m the total motor displacement(ml/r); η_{mm} the mechanical efficiency of motor.

Substitute (4), (5) and (6) into (3), then we have

$$mRa(t) = \mu m_0 gR + \frac{\eta_{mm} q_m \Delta p}{2\pi}. \tag{7}$$

2) Torque equation for engine shaft output end

According to the laws of rigid body in rotational motion, the torque equation for engine shaft output end is described by

$$J_e \frac{2\pi}{60} \frac{dn_e(t)}{dt} = \frac{q_p(t)\Delta p}{2\pi\eta_{pm}} - \Delta M_e(t) \tag{8}$$

where J_e is the moment of inertia of engine flywheel(kgm^2); $q_p(t)$ the pump displacement which acting as motor in this case(ml/r); η_{pm} the mechanical efficiency of pump.

By solving simultaneous (5) and (6), the pressure Δp is eliminated so that

$$J_e \frac{2\pi}{60}\frac{dn_e(t)}{dt} = \frac{q_p(t)R\left[ma(t)-\mu m_0 g\right]}{\eta_{pm}\eta_{mm}q_m} - \Delta M_e(t) \cdot \quad (9)$$

3) Relationship between traveling deceleration and motor revolutions

The traveling speed and the motor revolution have the following relationship

$$v(t) = \frac{2\pi R n_m(t)}{60 i_m} \quad (10)$$

where $v(t)$ is the traveling speed of double-drum vibratory roller(m/s); $n_m(t)$ the motor revolution(rpm); i_m the total reduction ratio from motor to wheel. Hence the traveling deceleration can be defined by

$$a(t) = \frac{dv(t)}{dt} = \frac{2\pi R}{60 i_m}\frac{dn_m(t)}{dt} \cdot \quad (11)$$

4) Flow conservation equation

Supposing that the flow remains constant from pump to motor by ignoring the leak of hydraulic system, we have

$$q_p(t)n_e(t) = q_m n_m(t) \cdot \quad (12)$$

This is equivalent to

$$n_m(t) = \frac{q_p(t)n_e(t)}{q_m} \cdot \quad (13)$$

5) Relationship model of engine speed and pump displacement

Substitute (13) into (11), then (11) can be rewritten as

$$a(t) = \frac{dv(t)}{dt} = C_a\left[q_p(t)\frac{dn_e(t)}{dt} + n_e(t)\frac{dq_p(t)}{dt}\right] \quad (14)$$

where

$$C_a = \frac{2\pi R}{60 i_m q_m} \cdot$$

Substitute(1), (2) and (14) into (9), and it follows that

$$J_e \frac{2\pi}{60}\frac{dn_e(t)}{dt} = \frac{RmC_a q_p(t)}{\eta_{pm}\eta_{mm}q_m}\left[q_p(t)\frac{dn_e(t)}{dt} + n_e(t)\frac{dq_p(t)}{dt}\right] - \frac{\mu m_0 g R q_p(t)}{\eta_{pm}\eta_{mm}q_m} - \left[\lambda n_e(t) - \lambda n_{e0}\right] \quad .(15)$$

Considering that the regulating value of pump displacement is linear in time, it can be denoted by

$$q_p(t) = q_0 - \Delta q_p = q_0 - kt \quad (16)$$

where q_0 is the initial value of pump displacement when the reverse driving process starting(ml/r); k the decreasing slope of pump displacement.

Substitute (16) into (15) and simplify the result, then it can be obtained that

$$\left[\frac{RmC_a\left(q_0-kt\right)^2}{\eta_{pm}\eta_{mm}q_m} - \frac{2\pi J_e}{60}\right]\frac{dn_e(t)}{dt} + \left[\frac{kRmC_a\left(q_0-kt\right)}{\eta_{pm}\eta_{mm}q_m} - \lambda\right]n_e(t) - \frac{\mu m_0 g R\left(q_0-kt\right)}{\eta_{pm}\eta_{mm}q_m} + \lambda n_{e0} = 0 \quad . \quad (17)$$

Let $a_1 = -\frac{2}{k}$, $a_0 = \frac{RmC_a q_0 - J_e\eta_{pm}\eta_{mm}q_m\frac{2\pi}{60}}{k^2 RmC_a}$, $b_0 = -\frac{kRmC_a q_0 + \eta_{pm}\eta_{mm}q_m\lambda}{k^2 RmC_a}$, $c_1 = \frac{\mu m_0 g t}{kmC_a}$ and $c_0 = \frac{-\mu m_0 g R q_0 + \eta_{pm}\eta_{mm}q_m\lambda n_{e0}}{k^2 RmC_a}$, respectively, then (17) can be rewritten as

$$\frac{dn_e(t)}{dt} + \frac{t+b_0}{t^2+a_1 t+a_0}n_e(t) = \frac{c_1 t + c_0}{t^2 + a_1 t + a_0} \cdot \quad (18)$$

Equation (18) is the relationship model of engine speed and pump displacement which is the basis of reverse driving control method.

C. Solution to Engine Speed and Pump Displacement Equation

Equation (18) is an one-order nonlinear homogeneous differential equation. The standard form of such equation is

$$\frac{dx(t)}{dt} + P(t)x(t) = Q(t) \cdot \quad (19)$$

The solution to (19) has the following form

$$x(t) = e^{-\int P(t)dt}\left[C + \int Q(t)e^{\int P(t)dt}dt\right] \quad (20)$$

where we have $x(t) = n_e(t)$, $P(t) = \frac{t+b_0}{t^2+a_1 t+a_0}$ and $Q(t) = \frac{c_1 t + c_0}{t^2+a_1 t+a_0}$. The initial condition $n_e(0) = n_{e0}$.

The analytic solution of (18) can be obtained from (20), which illustrates the relationship between engine speed $n_e(t)$ and pump displacement $q_p(t)$, precisely, the relationship between $n_e(t)$ and the initial displacement q_0 and decreasing slope k. The numerical integration methods may also be used to solve (18) to get the numerical solution.

Through regulating the value of k, an ideal curve of engine speed can be achieved, on which the peak value will not exceed allowable limit. Under this ideal curve, the average stopping deceleration will be controlled no less than its expected value, thus the stopping distance can be guaranteed.

V. THE CRITICAL VALUE OF DECELARATION IN REVERSE DRIVING PROCESS

In the process of reverse driving of the double-drum vibratory roller, besides the problem of engine overspeed, the frequent opening of the relief valve also should be considered. Therefore, it is necessary to differentiate the velocity equation to get the corresponding deceleration equation and determine its maximum value in the stopping process.

If the maximum deceleration remains less than the critical value under the control of engine speed, the system pressure will not exceed the relief valve opening pressure so that valve will not be opened in stopping process. Otherwise, the engine speed curve should be designed again to ensure the relief valve not acting. The approach to determine the critical deceleration is described as follows.

The efficient output torque of motor shaft is given by

$$M_m = M_L - \frac{2\pi J_m}{60}\frac{dn_m(t)}{dt} \tag{21}$$

where M_m is the effective output torque of a single motor(Nm); M_L the load torque of motor(Nm); J_m the moment of inertia converted to motor shaft(kgm^2). Here the load torque is denoted by

$$M_L = J_w \frac{dn_L}{dt} \tag{22}$$

where J_w is the rotating torque converted to the driving steel wheels((kgm^2); n_L the wheel rotating speed(rpm). We have

$$J_w = \frac{1}{2}k_w m_0 R^2 \tag{23}$$

where k_w is the converting coefficient.

From (10) we can see that

$$n_m(t) = i_m n_L(t) = \frac{60 i_m v(t)}{2\pi R}. \tag{24}$$

Substitute (22), (23) and (24) into (21), and the effective output torque of a single motor can be obtained as

$$M_m = \left(\frac{30k_w m_0 R}{2\pi} - \frac{J_m i_m}{R}\right)\frac{dv(t)}{dt}. \tag{25}$$

Note that

$$M_m = q_m \Delta p \tag{26}$$

then

$$q_m \Delta p = \left(\frac{30k_w m_0 R}{2\pi} - \frac{J_m i_m}{R}\right)\frac{dv(t)}{dt}. \tag{27}$$

From (27) it is shown that the pressure is linear with the deceleration.

Assume that the opening pressure of relief valve is $\Delta p_{\max}$. According to (27), the maximum deceleration of the traveling hydraulic system is

$$|a_{\max}| = \frac{30k_w m_0 R}{2\pi q_m \Delta p_{\max}} - \frac{J_m i_m}{R q_m \Delta p_{\max}}. \tag{28}$$

Therefore, in the stopping process, only when $a < a_{\max}$, the relief valve may not be opened. This control will reduce the heating costs of the hydraulic system.

It should be stressed that the maximum value of deceleration should be taken as one of the condition to design the engine plot in the control method.

VI. THE APPROACH TO DETERMINE THE PARAMETER OF FRICTION TORQUE EQUATION

Equation (17) is the basis on which the reverse driving control method is devised. All the parameters of (17) could be obtained directly or computed for an actual machine to be controlled. Comparatively, it seems difficult to get the value of λ in engine speed equation (2) only by theoretical analysis.

For this reason some traveling tests can be made to help establishing the approximate equation of engine friction torque and engine speed, i.e., determining the value of λ.

Combine (1), (2), (7), (8) and (10), we have

$$\frac{B}{2}\frac{d\left[v^2(t)\right]}{dt} = \frac{J_e \pi}{60}\frac{d\left[n_e^2(t)\right]}{dt} + \left[\lambda n_e(t) - \lambda n_{e0}(t)\right] \tag{29}$$

where

$$B = \frac{30k_w m_0 R}{2\pi} - \frac{J_m i_m}{R}.$$

Consider a group of reverse driving test data and separate the data section with overspeed from the whole process. Get the equations of traveling velocity $v(t)$ and engine speed $n_e(t)$ by data fitting, and substitute them into (29), then the value of λ can be determined by solving (29).

Table I. shows the test data of the reverse driving process in one traveling test of a double-drum vibratory roller. Using the test data and the approach above the curve of the engine friction torque with engine speed is obtained as shown in Fig. 7. Ignore the first nonlinear region of the curve, it can be calculated from the linear region that

$$\lambda = 2.2$$

TABLE I.
TEST DATA FOR REVERSE DRIVING PROCESS

No.	Parameters			
	t (s)	$n_e(t)$ (rpm)	$n_m(t)$ (rpm)	Δp (Mpa)
1	1.524	2518	1485	-2.19
2	1.765	2519	1513	-3.14
3	1.853	2521	1190	21.17
4	1.947	2533	1003	25.04
5	2.024	2547	852	25.08
6	2.079	2557	770	23.91
7	2.142	2571	646	21.32
8	2.188	2580	565	19.4
9	2.256	2593	485	15.24
10	2.368	2610	371	6.28

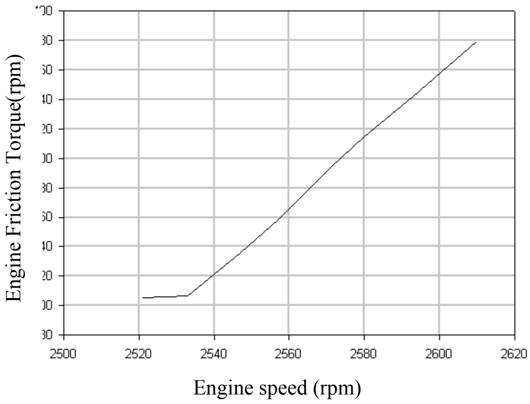

Figure 7. The plot of engine friction torque with engine speed

VII. Control Method of Reverse Driving Process

A. Offline Searching Algorithom for Pump Control Parameter

In section IV, the equation relating the engine speed to pump displacement has been derived as (17). The key control parameter in (17) is the decreasing slope k of pump displacement. The k satisfying the control requirements can be searched by an offline algorithm. The flow diagram of the algorithm is shown in Fig. 8.

It should be noted that if a k meets the overspeed requirement but doesn't achieve the average deceleration requirement, it is pointless to reduce k anymore. In this case, the only way is to make a piecewise linearization of the control for pump displacement, such that an optimal k meets the control requirements of each linear region can be calculated.

Using the selected k as the decreasing slope of pump displacement in the online control process, the engine speed will follow the ideal curve.

It should be noted that sometimes a single k can not meet the requirements so there may be several values of k to compose a piecewise-linearized curve.

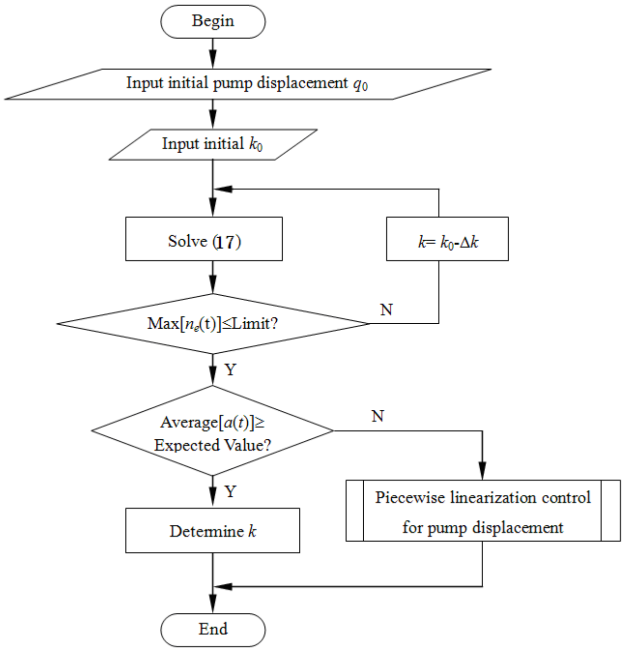

Figure 8. Computing flow diagram for decreasing slope of pump displacement.

B. Online Control Algorithom for Pump Control

In the actual control process, since the initial pump displacement is changeable with the operating joystick, the initial displacement q_0 needs to be computed in real time according to the joystick position value which has been divided into n sections so as to be processed more quickly.

Furthermore, we need to create a table to record the parameters searched by the offline algorithm with each q_0 of different section. This table is shown by table II.

The first column of table II is the joystick position x which determines the initial value of pump in stopping process. The second column represents the initial pump displacement q_0 corresponding to the joystick position. The decreasing slope k is shown in column 3.

In online control, get the sampled value q_0 and find the corresponding slope k or several ks in table II. Then control the pump displacement by the parameters found.

The flow diagram of online control algorithm is shown in Fig. 9.

TABLE II.
PARAMETERS FOR ONLINE CONTRIL METHOD

n	Parameters		
	Joystick position x (%)	Initial pump displacement q_0 (ml/r)	Slope of pump displacement k
1	90~100	q_{01}	$k_{11}, k_{12},\ldots$
2	80~90	q_{02}	$k_{21}, k_{22},\ldots$
3	70~80	q_{03}	$k_{31}, k_{32},\ldots$
4	60~70	q_{04}	$k_{41}, k_{42},\ldots$
5	50~60	q_{05}	$k_{51}, k_{52},\ldots$

n	Parameters		
	Joystick position x (%)	Initial pump displacement q_0 (ml/r)	Slope of pump displacement k
6	40~50	q_{06}	$k_{61}, k_{62},\ldots$
7	30~40	q_{07}	$k_{71}, k_{72},\ldots$
8	0~30	q_{08}	$k_{81}, k_{82},\ldots$

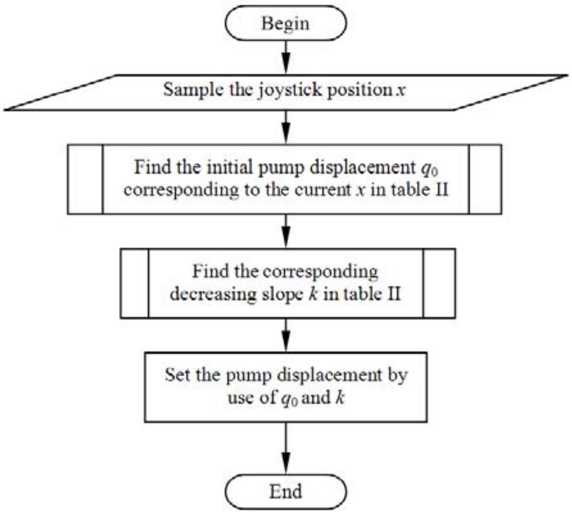

Figure 9. The flow diagram of online control algorithm

C. An Example of the Control Method

An example is given to illustrate the reverse driving control method as shown in Fig. 10 and Fig. 11.

Fig. 10 gives the piecewise-linearized curve of pump displacement, and Fig. 11 gives the engine speed curve meeting the overspeed requirements.

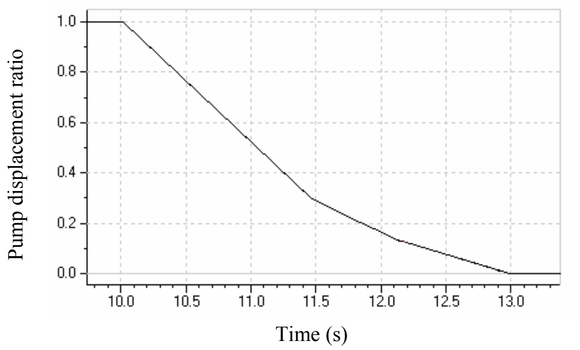

Figure 10. Pump displacement plot designed by the reverse driving control method

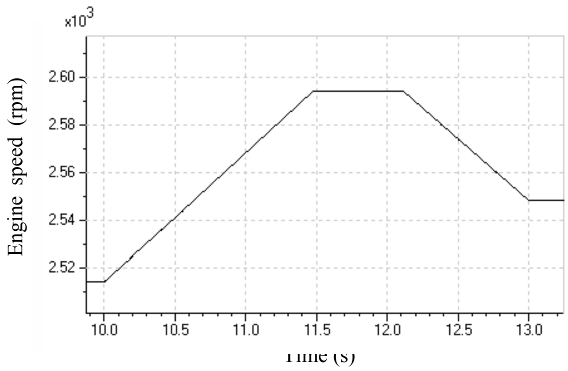

Figure 11. Engine speed plot designed by the reverse driving control method

VII. CONCLUDING REMARKS

Reverse driving is a complicated process of energy transmission, involving transforms and interactions of various parameters of both engine and hydraulic system. Furthermore, the machine itself has specific requirements of dynamic property. Therefore, to solve reverse driving problems comprehensively and systematically, further research and exploration are necessary.

In this paper, the reverse driving control method for one type of hydrostatic mobile machinery is devised under quite general broad assumption conditions. For further research, more assumptions can be relaxed as described in the following.

(1) The relationship between friction torque increment of engine and engine speed could be nonlinear, which could be obtained by testing and curve fitting.
(2) The pump displacement control law might be nonlinear in time.
(3) Electro-proportional motors are used in the system.
(4) If the engine throttle position is changeable, such as the case in a hydrostatic motor grader.

Recently, closed loop hydrostatic circuit are used as traveling driving system in more and more construction machinery such as bulldozers and motor graders, and also in some heavy transportation trucks, for which the complexity of reverse driving problem will increase and the control method should be improved correspondingly.

As one of the bottleneck problems, the problem of reverse driving must to be solved to improve the whole performance of the machines, especially for those with high speed. In this sense, much research is still required to find out better solutions to problems of reverse driving.

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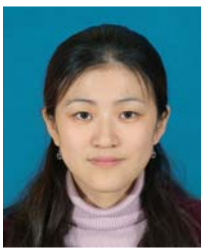

Xin Wang was born in Gaocheng city, Hebei province, China, on May 28, 1974. She received the Ph.D. degree in automatic control area from Northwestern Polytechnical University, Xi’an China, in 2004.

Since May 2004, she’s been an Assistant Professor and master candidate supervisor of Chang’an University, Xi’an China. In September 2007, she finished her two-year research in postdoctoral station of Sany Heavy Industry Co., Ltd, in Changsha city, Hunan province. She has authored and coauthored 15 technical papers in the field of control systems. Some of the previous and recent representative published articles are as follows.

- Xin Wang, Z. Y. Zhang, “Power self-adaptive control for trailing machinery”, *Road Machinery and Construction Mechanization*, vol. 27, pp. 72-74, November 2010, doi: CNKI:SUN:ZLJX.0.2010-11-038
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Her research interests include construction machinery automation, energy saving techniques, robust control, intelligent control, etc..

Dr. Wang received the Award for Science and Technology Advancement of Hunan Province in 2009, for her job in the project of “Research and industrialization on energy saving techniques for Construction machinery”.

Ruiying Zhao was born in Baoji city, Shaanxi province, China, on December 5, 1985. She received the Master degree in mechanical and electronic engineering area from Chang’an University, Xi’an China, in December 2008. Currently, she is studying for the Ph.D. degree in mechanical engineering area.

Since 2006, she has been assisting her supervisors to research on the energy saving techniques and has published two articles. Her research interests include construction machinery automation, energy-saving techniques for construction machinery, mechanical and electronic engineering, etc..