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RESEARCH ARTICLE

Control Strategy of Shuttle Shifting Process of Agricultural Tractor During Headland Turn

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ABSTRACT Tractor-implement combination generally involve the forward and reverse motion to perform the required task, which causes the tractor to frequently change its direction of movement. Automating the shuttle shifting process can reduce the labor intensity of the driver and improve the field efficiency and comfort. In this study, a power shuttle motion inverter is designed with a planetary gear train and clutch/brake. The dynamics model of the tractor power-train and a simulation model are established. Two types of the shuttle shifting process are designed and analyzed by the lever analogy method. To improve the shifting quality, control strategies for the shuttle shifting processes are formulated by analyzing the changing characteristics of the force exerted on the tractor-implement combination during the shifting processes. The analysis results show that quick and comfortable shuttle shifts are achieved during the process of moving from the forward to reverse directions and vice versa.

INDEX TERMS Agricultural tractor, dynamic analysis, shuttle shift.

I. INTRODUCTION

In modern agriculture, mechanization, automation, and precision are the development trends [1]. As the main power source of farm operation, the automation level of the agricultural tractor is crucial to precision agriculture [2], [3]. Carrying different implements and tools, general purpose agricultural tractors are often used in ploughing, harrowing, sowing, material movement, etc. Many of these applications involve forward and reverse motion of the tractor to perform the required task, which causes the tractor to frequently change its direction of movement [4]. For example, as shown in Figure 1, the tractor needs to change direction twice when turning around on end rows (from forward to reverse at circle A, and from reverse to forward at circle B in Figure 1). Automating the headland turn can reduce the labor intensity of the driver, save operating time and improve the field efficiency. Therefore, it is necessary to study the automatic shuttle shift to perform quick, easy, and safe forward/reverse changes in vehicle movement. This is a nontrivial task since it is difficult to find a good compromise between shift speed and comfort [5].

The shuttle shifting belongs to the clutch-to-clutch shifting, that is, changing the power flow by changing the operating states of the clutches to change the vehicle motion [6]. Many studies have been conducted on the process of clutch-to-clutch shifting [5], [6], [7], [8], [9], [10], [11], [12], [13], [14], [15], [16], [17], [18], [19], [20], [21]. Among them, Goetz et al. developed a gearshift controller for twin clutch transmissions, and integrated powertrain control strategy for gearshifts is developed [7], [8]. Zhang et al. presented an analytical model for the simulation, analysis, and control of shift

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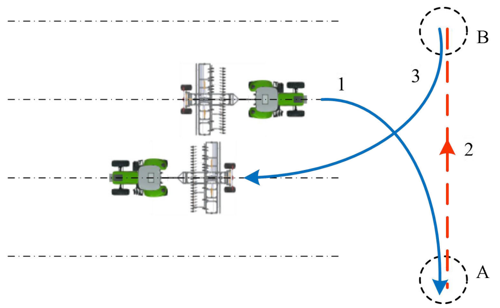

FIGURE 1. Tractor trajectory during headland turn.

dynamics for the DCT vehicles, and models the kinematics, dynamics, and control of the transmission for the analysis of powertrain overall performance and shift transient characteristics [9], [10]. Amendola and Alves established a parallel between the automatic transmission and double clutch transmission during up shifts [11]. Liu et al. focused on the dynamic modeling, analysis, control strategy, and the related experiments for dry dual clutch transmissions (DCTs) during vehicle launch and shifts [12], [13]. Walker and Zhang et al. proposed a method for combined speed and torque control of vehicle powertrains with dual clutch transmissions for both the engine and clutches, and conducted comparison of different strategies for modelling DCT equipped powertrains [14], [15]. Galvagno et al. presented a detailed model for the quantitative analysis of a Dual Clutch Transmissions kinematics and dynamics [16]. Li et al. developed coordinated control of gear shifting process with multiple clutches to improve the gear shifting quality for power-shift transmission [17]. Li et al. proposed a finite-time linear quadratic regulator for the optimal control of the two friction clutches in the torque phase for the upshift process [18]. These studies contributed greatly to vehicle drivetrain modeling, clutch-to-clutch shifting process analysis, and control methods. However, they mainly focused on the gear shifting process between the forward gears and are not fully applicable to gear shifting between the forward and reverse gears because the driving conditions and changes in external forces between them are quite different.

Raikwar et al. developed a model of the individual component of power shuttle transmission system and whole vehicle model, using MATLAB as a simulation platform, and the proportional valve characteristic, vehicle shift time and motion inversion time of modelled tractor were studied by this model [19]. Kim et al. investigated the effects on the shifting performance of the design parameters of a power shuttle tractor by developed models of the hydraulic control system and power shuttle transmission using commercial software EASY5 [20]. Wang et al. studied the matching of the control parameters of clutch hydraulic systems in order to improve the shift quality of the tractors fitted with hydrostatic power split continuously variable transmission during starting [21]. Xia et al. proposed a power shift control strategy based on torque and speed transition, which aims to deliver multiple target and multi-parameter optimization of power shift control. Then the clutch terminal oil pressure of optimal control during power shifting is optimized for shift quality based on minimum optimal control theory [22]. Chen et al. analysed the shift quality of the cotton picker to improve the ride comfort of a continuously variable cotton picker [23]. Cao et al. studied the parameter configuration of the gearbox wet clutch to improve the shift quality of the power reversing tractor [24]. Song et al. proposed a tractor full-power shift transmission shifting coordination section law for the problem of poor transmission shift quality [25]. Savaresi et al. discussed the design of an automatic motion inverter control system for agricultural tractors, and the design of an inner loop for the control of the EVP (electro-hydraulic valve) current, the open-loop switching strategies, and the design of the outer control loop, which regulates the vehicle speed, are considered [5]. Park et al. analyzed the characteristics of the delay time when starting the power shuttle at various oil temperatures based on the developed model using AMESim [26].

These studies provided detailed analysis of the shuttle shift process, mainly focused on the hydraulic system performance and control for shuttle shift, and achieved good results. In order to further improve the shifting performance, it is necessary to design a control strategy according to the evaluation index of the shifting performance (quality).

The shuttle shifting performance is usually evaluated by the shuttle shifting speed and comfort, which can be quantified by the shifting time and impact. However, these two evaluation indicators are mutually restrictive to some extent. For example, reducing the shift time often leads to an increase in impact. This restriction can be reduced by analyzing the cause of the impact and carrying out reasonable control. The impact (jerk) is the rate of change of the vehicle acceleration, and can be regarded as the rate of change of the resultant force on the vehicle. In other words, the change in the driving force and resistance of the tractor determines the impact (jerk). The changing characteristics of resistance are objective. By properly controlling the clutch/brake to reduce the rate of change of the tractor resultant force at the critical moment (e.g., the moment when the rolling resistance changes direction owing to the vehicle changing its travel direction), the impact can be reduced. In addition, the shifting speed can be increased at other times. Thus, a quick and comfortable shuttle shifting process can be achieved. The dynamic analysis of the shuttle shifting process and analysis of the driving conditions are the prerequisites for realizing this control.

Therefore, in this study, a power shuttle motion inverter is designed for an agricultural tractor, and the shuttle shifting processes and driving conditions are analyzed. Then, a dynamic model of the power-train and a simulation model of the entire tractor are established. Furthermore, automatic control strategies for the shuttle shifting processes are developed based on the operating conditions and driver's intent to meet a quick and comfortable shifting process. Finally, the effectiveness and advantages of the automatic control strategies are verified by simulation analysis.

II. MODEL DEVELOPMENT

A. DESCRIPTION OF POWER SHUTTLE SYSTEM

A power shuttle system is a type of transmission system used in heavy-duty vehicles such as tractors, backhoes, and loaders. It provides two modes of operation: forward and reverse, allowing the operator to quickly and easily change the direction of travel without having to stop the vehicle completely. A motion inverter usually alternates the direction of the vehicle by switching the operating states of its clutches/brakes [27], [28]. It is widely used in PST (power-shift transmission) and CVT (continuously variable transmission) of tractors, engineering vehicle, and automobiles. Vehicles with a power shuttle motion inverter can offer forward-reverse shuttling to perform quick forward-reverse functions. Major advantages of power shuttle tractors include effortless operation, a reduction in operating time, operator safety, and power-train safety [29]. As Figure 2 shows, a power-shift transmission for an agricultural tractor is designed in our work. The motion inverter consists of a clutch CF, brake B, and planetary gear train with a dual planet. The multi-disc clutch and brake allow the operator (or controller) to engage or disengage the transmission without stopping the vehicle. Compared with the parallel axis arrangement, the planetary gear train can reduce the installation size, and the double planetary structure is advantageous for the speed ratio design. The power splitting characteristic of the planetary gear train facilitates the soft starting and shifting of the tractor.

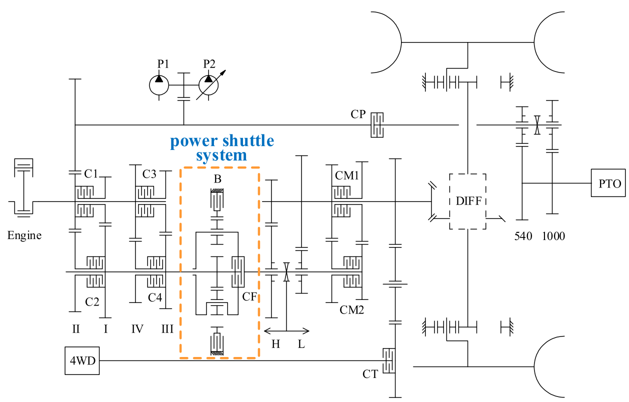

FIGURE 2. Stick diagram of power-train with shuttle motion inverter.

This motion inverter has three major working positions (neutral, forward, and reverse), which are described as follows:

(a) Neutral: In this position, both clutch CF and brake B are disengaged. There is no power transmitting from the engine to the rear gearbox, and the tractor is in neutral gear.

(b) Forward position: In the forward mode, the power shuttle system operates like a conventional transmission. Clutch CF and brake B are in engaged and disengaged conditions, respectively, which allows the tractor to move forward. Because the sun gear and planet carrier of the planetary gear train are held together by the engaged clutch CF in this condition, the transmission ratio of this motion inverter system in the forward position is 1.

(c) Reverse position: In this position, Clutch CF and brake B are in disengaged and engaged conditions, respectively. This condition allows the tractor to move in reverse. Because the ring gear of this planetary gear train is fixed to the case, the transmission ratio of this system in the reverse position depends on the characteristic parameter of the planetary gear train, as Equation (1) indicates. The lever analogy method can be used to analyze the speed relations of the tractor components during the shuttle shifting process between forward mode with reverse mode.

$$\begin{cases} i_f = 1 \\ i_r = k - 1 \end{cases} \tag{1}$$

where i_f and i_r are the transmission ratio of the motion inverter in forward and reverse positions, respectively; k is the characteristic parameter of the planetary gear train, which is equal to the ratio of number of ring gear teeth to that of sun gear (or radius), and $k = 2$ in this study.

In addition, as shown in Figure 2, clutches C1, C2, C3 and C4 are used to select the gear of the front power-shift gearbox. Clutch CM1, clutch CM2 and the H/L gearshift are used to select the gear of the rear gearbox. Clutch CP is used to supply or cut off the power of the PTO (power take off) shaft. Clutch CT is used to select whether to use front-wheel drive.

B. DYNAMICS

The dynamics analysis of the powertrain with the motion inverter is the premise of shifting process analysis and automatic control for shuttle shifting. Since the gears of the tractor's front and rear gearboxes are fixed during the shuttle shifting process, the front and rear gearboxes can be further simplified when establishing a dynamics model for the analysis of the shifting process. The dynamic model for the powertrain designed in this study is shown in Figure 3. Gear shafts are modeled as lumped masses.

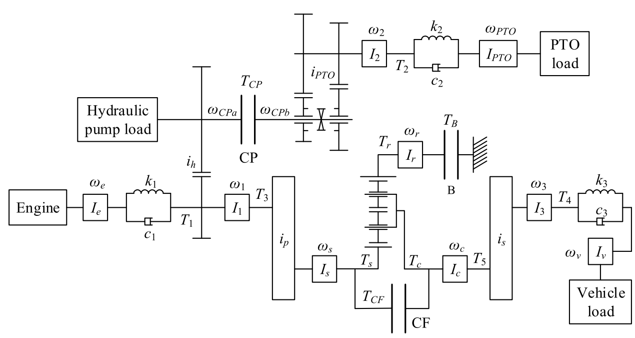

FIGURE 3. Dynamic model of powertrain.

The mathematical model can be expressed by Equations (2) to (15).

$$T_e - T_1 = I_e \dot{\omega}_e \tag{2}$$

$$T_1 - (T_h + T_{CP})/i_h - T_3 = I_1\dot{\omega}_1 \tag{3}$$
$$T_{CP}i_{PTO} - T_2 = I_2\dot{\omega}_2 \tag{4}$$
$$T_2 - T_{PTO} = I_{PTO}\dot{\omega}_{PTO} \tag{5}$$
$$T_3 i_p - T_s - T_{CF} = I_s\dot{\omega}_s \tag{6}$$
$$T_c + T_{CF} - T_5 = I_c\dot{\omega}_c \tag{7}$$
$$T_r - T_B = I_r\dot{\omega}_r \tag{8}$$
$$T_5 i_s - T_4 = I_3\dot{\omega}_3 \tag{9}$$
$$T_4 - T_v = I_v\dot{\omega}_v \tag{10}$$
$$T_1 = k_1(\theta_e - \theta_1) + c_1(\omega_e - \omega_1) \tag{11}$$
$$T_2 = k_2(\theta_2 - \theta_{PTO}) + c_2(\omega_2 - \omega_{PTO}) \tag{12}$$
$$T_4 = k_3(\theta_3 - \theta_v) + c_3(\omega_3 - \omega_v) \tag{13}$$
$$T_s{:}T_r{:}T_c = 1{:} - k{:}k - 1 \tag{14}$$
$$\omega_s - k\omega_r + (k-1)\omega_c = 0 \tag{15}$$

where T_e, T_h, T_{PTO}, and T_v are the torques of engine, hydraulic pump, PTO tool, and vehicle load, respectively; T_1, T_2, T_3, T_4, and T_5 are the torques on the respective shafts shown in the Figure 3; T_{CP}, T_{CF}, and T_B are the torques of clutch CP, CF, and brake B, respectively; I_e is the rotational inertia of the engine flywheel; I_1, I_2, I_3, I_s, I_r, and I_c are the equivalent mass moments of inertia for the lumped masses including the connective parts as shown in Figure 3, respectively; I_v denotes the vehicle equivalent mass moment of inertia on the output shaft; ω_s, ω_r, and ω_c are the angular velocities of the sun gear, ring gear, and planet carrier, respectively; ω_e, ω_1, ω_2, ω_3, ω_{PTO}, and ω_v are the angular velocities of corresponding shafts, respectively; In a similar fashion, θ_e, θ_1, θ_2, θ_3, $\theta_{PTO,}$ and θ_v are the rotation angles of corresponding shafts, respectively; i_h, i_p, and i_s are the speed ratios of corresponding gear pairs, respectively; k_1, k_2, and k_3 are the equivalent stiffness of each part shown in Figure 3, respectively; c_1, c_2, and c_3 are equivalent damping coefficients of each part, respectively; k is the characteristic parameter of the planetary gear train.

According to the structure of the drivetrain and the principle of mechanical transmission, the following equations can be obtained.

$$\omega_1 = \omega_s i_p \tag{16}$$
$$\omega_c = \omega_3 i_s \tag{17}$$
$$\theta_1 = \theta_s i_p \tag{18}$$
$$\theta_c = \theta_3 i_s \tag{19}$$
$$k = R_r/R_s \tag{20}$$

where R_r and R_s are the pitch circle radii of the ring gear and sun gear, respectively.

The running state of clutch CP depends on the type of farm implements and work condition. The clutch CP is engaged when the PTO shaft is required to operate. This dynamic model is suitable for 4WD (4 wheel drive) and 2WD (2 wheel drive). It should be noted that the losses in the bearings, losses of churning, etc. are ignored in this model.

C. SIMULATION MODELS

In addition to the drivetrain dynamics model described above, models of other key components are also critical to simulation analysis.

In the normal operation of a tractor-implement combination, the power of the tractor is mainly used to drive the tractor-implement combination to move and drive the implements connected via the PTO shaft for operation. Because the shuttle shifting process in a headland turn is mainly studied in this paper, and the agricultural implements do not participate in work during this process, only the driving load is considered in the load model. The external resistance during the tractor's travel mainly includes the rolling resistance, air resistance, slope resistance and draft resistance of the implements. Among them, the air resistance and draft resistance of the implements can be ignored owing to the low travel speed of the tractor and the fact that the agricultural implements do not enter the soil for work. Therefore, as the force hindering the movement of the tractor, the resistance model of this paper is expressed by Equation(21).

$$F_r = F_f + F_i = mgf\ cos\theta + mg\ sin\theta \tag{21}$$

where F_r, F_f and F_i are the total external resistance, rolling resistance, and slope resistance, respectively; m is the total mass of the tractor-implement combination; g is the gravitational acceleration; f is the coefficient of rolling resistance; θ is the slope of the farmland.

It should be pointed out that rolling resistance F_f always hinders the movement of the tractor-implement combination. Therefore, when the tractor changes its direction of motion, the direction of the rolling resistance also changes. This poses a challenge to ensure the comfort of the shuttle shifting process. The values of parameters are $m = 10000$kg, $f = 0.008$, and $\theta = 0$ rad in this study.

The working states of clutch CF and brake B play a decisive role in the shuttle shifting process. Brake B in the motion inverter is basically the same as clutch CF in principle, so clutch CF and brake B adopt the same model in this paper. The torque that the clutch/brake can transmit is calculated by Equations (22) to (23). When the clutch is fully engaged, the torque transferred by the clutch depends on the actual torque of the corresponding shaft under the condition that the maximum static friction torque is not exceeded [29], [30].

$$T_C = \mu_c N_p A_p P_{eff} R_{eff} \tag{22}$$
$$P_{eff} = max\left[(P_{hc} - P_{ks}), 0\right] \tag{23}$$

where T_C is the torque transferred by the clutch/brake; μ_c is the friction coefficient; N_p is the number of friction surfaces; A_p is the engagement surface area of the piston; P_{eff} is effective pressure; and R_{eff} is the effective torque radius; P_{hc} is the actual oil pressure; P_{ks} is the kiss-point pressure of the clutch/brake [30].

The values of some parameters of above model in this study are as follows: $N_p = 6$, $A_p = 0.012$ m^2, $R_{eff} = 0.065$ m. The kiss-point pressure is 0.4 MPa in this study, beyond which the clutch/brake torque increases proportionally. The friction coefficient is a function of the speed difference between the main and driven discs of the clutch/brake, and the model in existing literature is used [31]. The changing conditions and processes of the clutch/brake state are variable. During simulation, the clutch/brake transitions to the appropriate state according to the real-time conditions based on the state-transition diagram [17]. Specific hydraulic values are used directly and the adjustment process of the hydraulic actuator is neglected in this paper. The specific control and simulation of clutch hydraulic actuator can refer to a lot of existing research [19].

In this study, a high-pressure common-rail diesel engine of 175 kW power with a rated torque of 765 N· m@2200 rpm is used [32]. This engine has a full-range speed-governing characteristic whose full-load characteristics are shown in Figure 4.

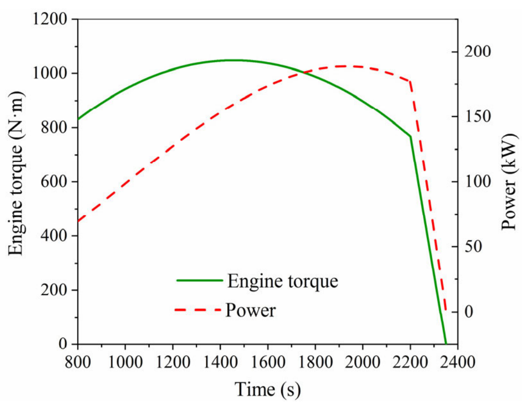

FIGURE 4. Full-load characteristics of diesel engine.

The mean engine torque model is widely used for simulation study of gear shifting, and the high-frequency vibrations of the engine can be neglected for investigations on gear shift comfort [15], [33]. In this study, a mean diesel engine torque model is used for simulation. The engine torque depends on the engine speed and cycle fuel injection quantity, and the cycle fuel injection quantity is adjusted by the PID (proportional-integral-derivative) controller based on the accelerator pedal opening to meet the full-range speed-governing characteristic of the tractor diesel engine. Therefore, the engine torque T_e can be obtained by inputting the engine speed n_e and accelerator pedal opening. The speed-governing characteristic can ensure that the engine speed remains relatively constant in the event of a sudden load change, thus ensuring a relatively stable vehicle speed.

An entire simulation model of the tractor for shuttle shifting is built with the Matlab/Simulink and Matlab/Stateflow in accordance with the above-mentioned mathematical models. A fixed-step solver is selected to compute the states of the model during simulation.

III. SHUTTLE SHIFTING PROCESS ANALYSIS

A. PROCESS OF FORWARD TO REVERSE

In order to be intuitive and easy to understand, the lever analogy method [34] is used in this paper to analyze the speed relations of the tractor components during the shuttle shifting process. As Figure 5 shows, the entire shifting process of forward to reverse direction in this study can be described in five stages. The sun gear as the motion inverter input can reflect the engine speed to some extent, and the planet carrier as the output can reflect the tractor speed to some extent according to the transmission ratios.

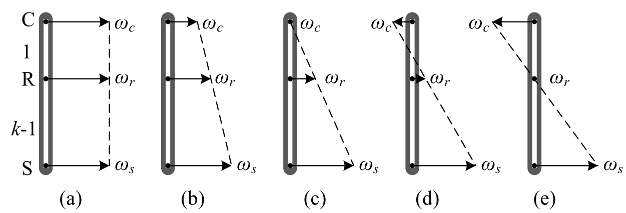

FIGURE 5. Speed analysis of forward to reverse.

As Figure 5(a) shows, at the beginning of the shifting process, clutch CF is still in a completely engaged state. The sun gear, planet carrier, and ring gear are running at the same speed, at which point the tractor is moving at normal speed. At this stage, the torque is transferred through the sun gear and clutch CF, and its power flow is indicated by the blue dotted line in Figure 6. Next, when CF begins to disengage, the reduction in the positive driving force causes the speed of the planet carrier and tractor to drop. Owing to the governing characteristic of the tractor diesel engine, the sun gear can maintain a relatively stable speed. For this stage, the relationship between the speeds of the three components is shown in Figure 5(b). Under the action of the resistance, the speed of the carrier continues to drop until the tractor reaches the zero point and stops.

During the above two stages, the clutch and brake operate according to different control strategies that depend on the operating conditions or driver's intention. Therefore, the power flow may be transmitted along the blue dashed line, solid yellow line, or a combination of both, as shown in Figure 6. In the next stage, as Figure 5(d) shows, as brake B is gradually engaged, the ring gear speed is further reduced, and the planet carrier and tractor begin to reverse direction. When brake B is completely engaged, the ring gear is fixed to the case. The planet carrier and tractor reach respective normal reverse speeds, and the shuttle shifting process ends. In the last two stages, clutch CF does not transmit torque, and the power flow is shown by the yellow curve in Figure 6.

B. PROCESS OF REVERSE TO FORWARD

The shuttle shifting process of reverse to forward is basically the opposite of the above shifting process. Figure 7 shows the changes in the speeds of the three components (sun gear, ring gear, and planet carrier) and their relationship.

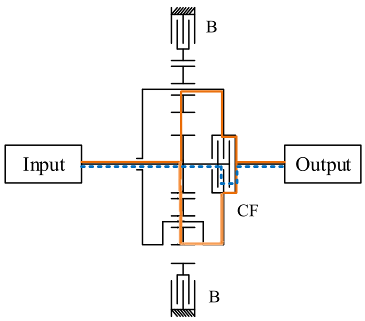

FIGURE 6. Power flows of motion inverter.

At the beginning, the tractor travels in reverse gear. During the shifting process, owing to the governing characteristic of the diesel engine, with a fixed accelerator pedal opening, the speed of the sun gear can maintain a relatively steady speed. Under the coordinated control of clutch CF and brake B, the ring gear speed will gradually increase from 0, and in the meantime, the carrier speed will gradually change from negative to positive. The power flow varies according to the states of clutch CF and brake B, from the route indicated by the initial yellow curve to the route indicated by the final blue curve in Figure 6. Correspondingly, the tractor speed successfully turns from reverse to forward. The control method of clutch CF and brake B will be discussed later, which depends on shifting performance and driver's intention.

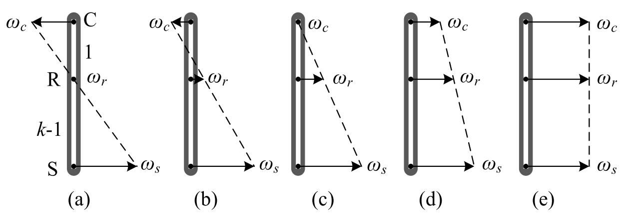

FIGURE 7. Speed analysis of reverse to forward.

IV. CONTROL STRATEGY

In general, comfort during shifting, shifting time, and sliding work are the main indicators used to measure power-shift quality. These indicators can be used for reference to measure the shuttle shifting quality. Among them, the criterion of comfort in shifting process is the jerk (impact), that is, the derivative of tractor acceleration, as shown in Equation (24).

$$\mathrm{Jerk} = \frac{\mathrm{d}a}{dt} = \frac{dv^2}{\mathrm{d}t^2} \tag{24}$$

where a and v are the acceleration and travel speed of tractor, respectively.

The key to improving the quality of the shuttle shifting is to reduce the impact (improve the comfort) and reduce the shifting time, on the premise of keeping the sliding grinding work basically not increased. However, there is usually a mutually restrictive relationship between the reducing impact and saving the shifting time. Therefore, it is necessary to achieve comprehensive optimization by formulating a reasonable control strategy. According to the above analysis of the shuttle shifting process, the change rate of the resultant force acting on the vehicle determines the impact to a certain extent. At the critical moment when the resultant force is prone to sudden change, proper control of the shifting speed can effectively reduce the impact. At other times, the shifting time can be reduced by appropriately increasing the speed of shift, thereby maintaining a faster and more comfortable commutation throughout the shifting process.

The control of the clutch and brake is key to realizing the shuttle shifting process, and the coordinated control of the engine can be carried out when necessary. The control strategies developed in this paper are as follows.

A. CONTROL OF FORWARD TO REVERSE

Clutch and brake torques of the shuttle shift system determine the driving torque of the agricultural tractors. During power shuttle shifting, parasitic power and power loss should be avoided, which results in reduced clutch life or increased shift time and shocks. The torques of clutch/brake in the shuttle shift system change according to the following designed rules can effectively improve the shifting quality.

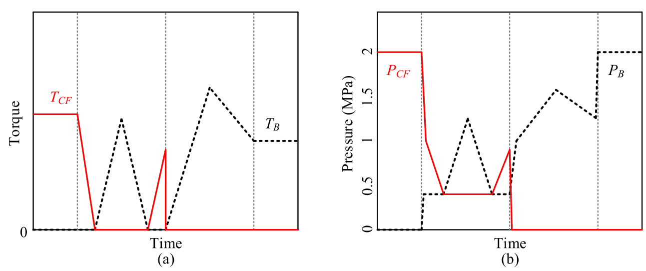

FIGURE 8. Control schematic for shift from forward to reverse: (a) torque changing rules; (b) oil pressure.

The core to reduce impact is to use a smaller torque change rate when the speed crosses 0. The specific change process is as follows: As shown in Figure 8(a), at the beginning of the shift from forward to reverse, the torque of clutch CF should be slowly lowered to avoid a large impact owing to a sudden drop in the driving force. Then, the torque of brake B should be gradually increased to accelerate the tractor speed drop. The brake torque is gradually reduced back to 0 at the right moment. When the vehicle speed approaches 0, the clutch torque is slightly increased to avoid the impact caused by the reversal of the rolling resistance. Next, the torque of brake B gradually rises to cause the vehicle to start in reverse. Finally, the torque of brake B before the brake is engaged and should be kept at a relatively low level to reduce the impact at the end of the acceleration. Coupled with the speed-governing characteristic of the tractor diesel engine itself, given the above rules, the tractor speed can maintain a stable and rapid change.

To achieve the above torque variation, the change processes of oil pressures (P_{CF}, P_B) are designed as shown in

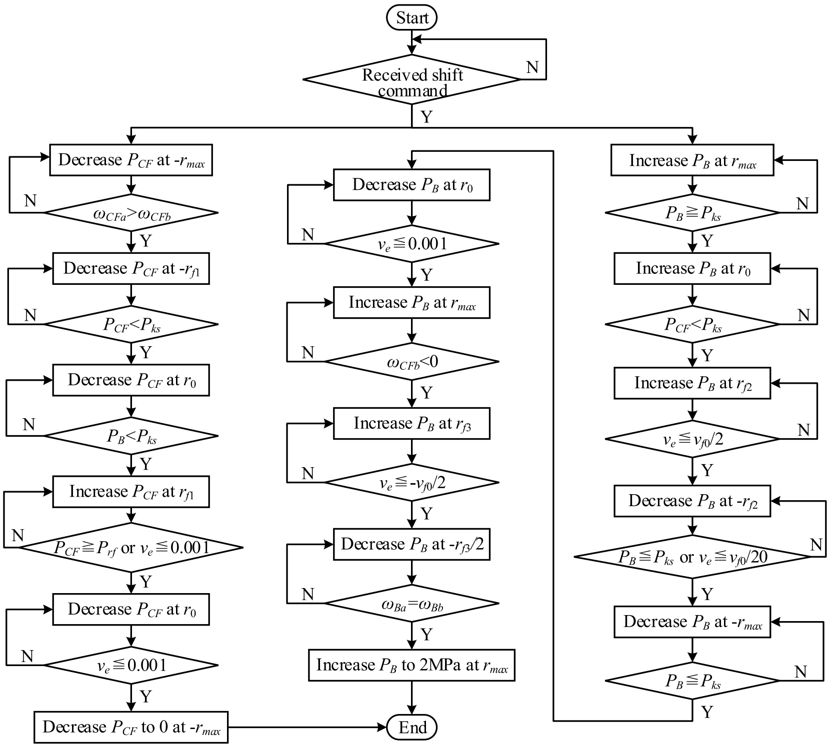

FIGURE 9. Control flow for shifting process of forward to reverse.

Figure 8(b). The pressure interval of the clutch/brake hydraulic cylinder ranges from 0–2 MPa. The oil pressures are maintained at 0 MPa, 2 MPa, and 0.4 MPa (kiss-point pressure) in the brake/clutch states of completely disengaged, engaged, and prepared to engage, respectively. The pressure changes rapidly during the period which does not affect the torque (e.g., between 0 and the kiss-point pressure, between 2 MPa and the pressure at which the clutch main and driven discs start to have a difference in rotational speed). Figure 9 shows the specific control flow based on above control strategy for the shifting process of forward to reverse.

In Figure 9, ω_{CFa}, ω_{CFb}, ω_{Ba}, and ω_{Bb} are the speeds of clutch CF main disc, CF driven disc, brake B main disc, and B fixed disc, respectively; r_{max}, r_0, r_{f1}, r_{f2}, and r_{f3} are the oil pressure change rates at different times during the shuttle shifting process, respectively; P_{ks} and P_{rf} are the kiss-point pressure and threshold pressure (which can provide a torque slightly lower than that needed to overcome road resistance), respectively; ve and v_{f0} are the tractor's real-time speed and the speed at the beginning of the shifting, respectively.

The values of these parameters determine the speed of the shuttle shift. Among them, the values of ω_{CFa}, ω_{CFb}, ω_{Ba}, ω_{Bb}, v_e, and v_{f0} can be obtained in real time during travel of the tractor-implement combination. The maximum and minimum oil pressure change rates (r_{max}, r_0) are set as 1×10^{7} Pa/s and 0 Pa/s, respectively; and the kiss-point pressure P_{ks} is 0.4 MPa in this study. The values of other parameters including the pressure change rates (r_{f1}, r_{f2}, and r_{f3}) and threshold pressure P_{rf} need to be determined according to the working conditions and the driver's intention to better adapt to the actual operation.

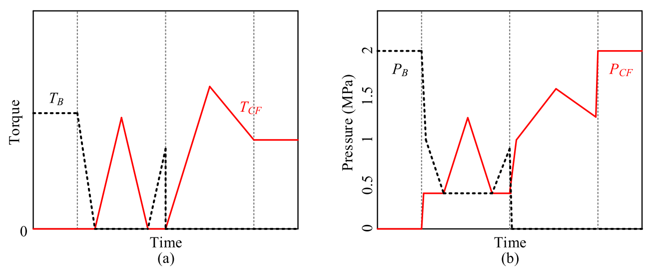

FIGURE 10. Control schematic for shift from reverse to forward: (a) torque changing rules; (b) oil pressure.

B. CONTROL OF REVERSE TO FORWARD

The control rule for the shifting process of reverse to forward is similar to the process of forward to reverse. Compared with the above process, the torque (as well as the pressure) changes of the clutch and brake are exchanged in this shifting process. Another major difference is the setting of the parameter values. Figure 10 shows the control schematic for this shuttle shifting process.

Figure 11 shows the specific control flow based on the control strategy from Figure 10 for the shifting process of reverse to forward. The values of major parameters for shuttle shifting speed including the pressure change rates (r_{r1}, r_{r2}, and r_{r3}) and threshold pressure P_{rr} need to be determined

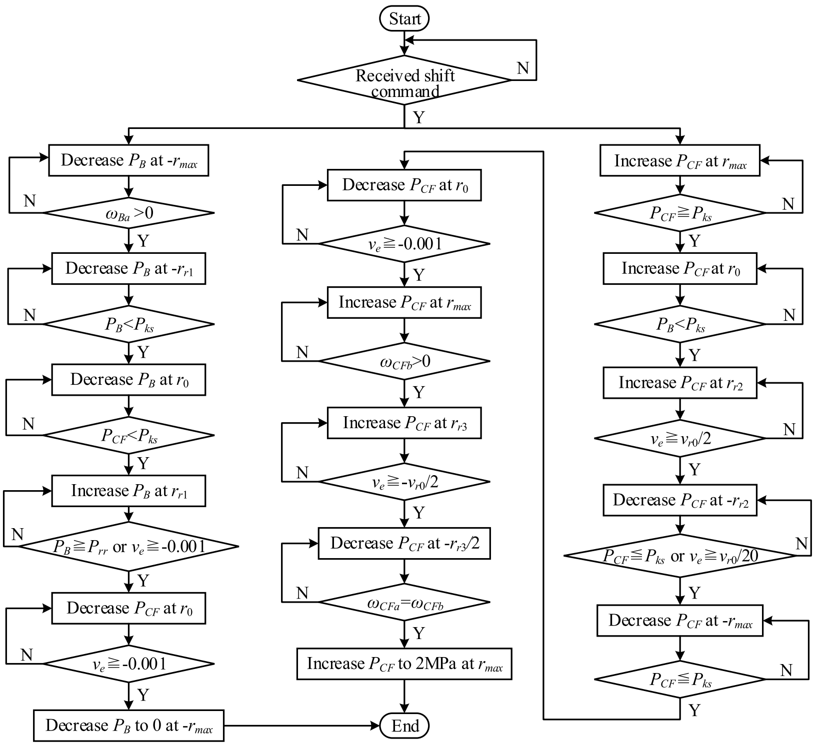

FIGURE 11. **Control flow for shifting process of reverse to forward.**

according to the working conditions and the driver's intention to better adapt to the actual work. This will be discussed in Section IV-C. v_{r0} is the reverse speed at the beginning of the shifting. The values of other parameters are consistent with that of the above shifting control for shift from forward to reverse.

C. CONTROL PARAMETER DESIGN

The values of the control parameters have an important influence on the shifting process. However, different drivers have different requirements for the shuttle shifting speed and different tolerability for impact, and different working conditions have different effects on the shift. No uniform values of the control parameters are available for all drivers and conditions. Therefore, the ideal control parameter design needs to be based on the intentions of different drivers and various working conditions. For example, the greater accelerator pedal opening and the higher gear mean the driver has a more aggressive driving style, and a quicker shift response is needed; and higher vehicle speeds and greater driving resistance require smaller shifting speeds to avoid excessive impact. It is a complicated task to take all these factors into account to design the control parameters, and it is difficult to implement it all in one article. In this study, only a specific condition is used as a case to design control parameters and conduct shuttle shifting control studies.

The condition of a 30% accelerator pedal opening, tenth gear, and medium level rolling resistance coefficient (0.08) are used for the shuttle shifting analysis, which belongs to relatively poor condition for shuttle shifting. The parameter values during shifting process of forward to reverse are set as follows: $r_{f1} = 5 \times 10^5$ Pa/s, $r_{f2} = 1 \times 10^6$ Pa/s, $r_{f3} = 1 \times 10^6$ Pa/s, $P_{rf} = 5.8 \times 10^5$ Pa. Since the state change processes of the clutch and brake are exchanged, the parameter values during shifting process of reverse to forward are designed as: $r_{r1} = 1 \times 10^6$ Pa/s, $r_{r2} = 5 \times 10^5$ Pa/s, $r_{r3} = 5 \times 10^5$ Pa/s, $P_{rr} = 7.2 \times 10^5$ Pa. In practical engineering applications, the change rate of the clutch pressure can be adjusted according to different driver types, comfort requirements, and working conditions. Specific studies on intentions of drivers and operating conditions are not included in this paper.

V. RESULTS AND DISCUSSION

According to the above control strategy and simulation method, the shuttle shifting processes are simulated and analyzed in this study. Figure 12 shows the simulation results of the shifting process of forward direction to reverse. In Figure 12, graph (a) shows the history of pressure changes in clutch CF and brake B that are in line with the control strategy proposed above. Graphs (b) and (d) show the changes in rotation speeds and torques of key components in the motion inverter under pressure changes. Graphs (c), (e), and (f) show the tractor longitudinal acceleration, tractor speed, and impact (jerk) during the process of shuttle shifting from the forward to reverse direction of the tractor, respectively.

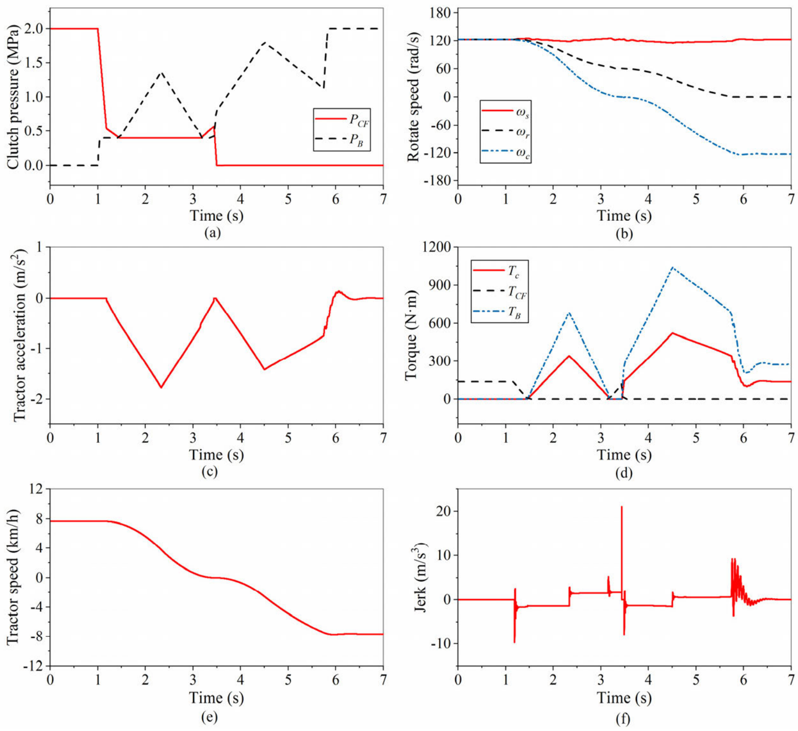

FIGURE 12. Analysis results of shuttle shifting process from forward to reverse direction.

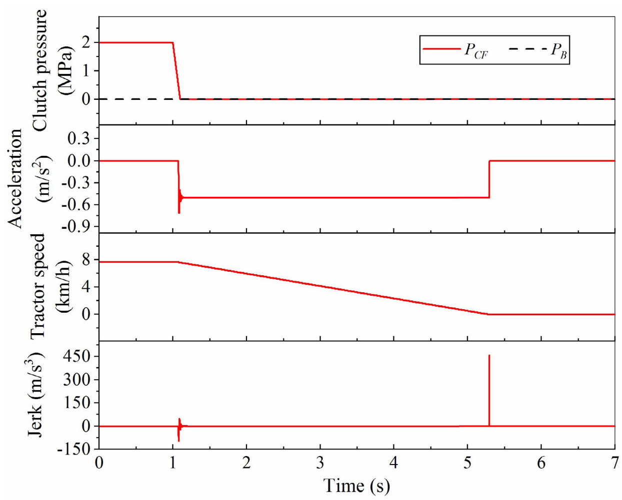

FIGURE 13. Simulation results of free parking without interference.

As can be seen from Figure 12, before shuttle shifting, the oil pressure of clutch CF is kept at the highest value of 2 MPa, and oil pressure of brake B is maintained at 0 MPa. During this time period (0 s to 1 s), the engine power is completely transmitted by the clutch, and the clutch torque is about 145 N·m. The sun gear, planet carrier, and ring gear operate at the same speed (~120 rad/s). The tractor travels smoothly at a speed of about 7.8 km/h. As the shifting command is issued at the first second, the brake oil pressure is rapidly increased to the kiss-point pressure and maintained. The clutch oil pressure drops rapidly until the clutch main and driven discs have a difference in rotational speeds, and then the oil pressure slowly drops. During this time period, the clutch torque and speeds of the ring gear, planet carrier, and tractor begin to decline. At ~1.4 s, the clutch reaches the kiss-point pressure. The brake oil pressure rises to speed up the vehicle speed, and then returns to the kiss-point pressure. In the meantime, the ring gear, planet carrier, and tractor speed drop rapidly. When the vehicle speed is close to 0, the clutch pressure rises slightly to avoid excessive parking shock caused by the large rolling resistance. At ~3.4 s, the speed of the tractor drops to 0, and the clutch completely releases the pressure to complete its mission. The brake raises the pressure quickly and then slowly to start the tractor in reverse. Then, the brake pressure changes according to the control strategy. The speeds of the planet carrier and tractor change from 0 to the reverse target values and finally reach their respective target values at ~5.7 s. The shuttle shifting process is completed. Then, the tractor is stably running at a speed of ~−7.8 km/h in the reverse tenth gear.

During the entire shifting process, the longitudinal acceleration curve of the tractor presents a 'W' shape, which ensures that the change rate of tractor acceleration at the beginning of reversing, at the point of 0-speed crossing, and at the end of shifting process is small. This avoids an excessive impact.

As can be seen from Figure 12(f), except for the moment when the tractor speed reaches 0, the impact (jerk) during

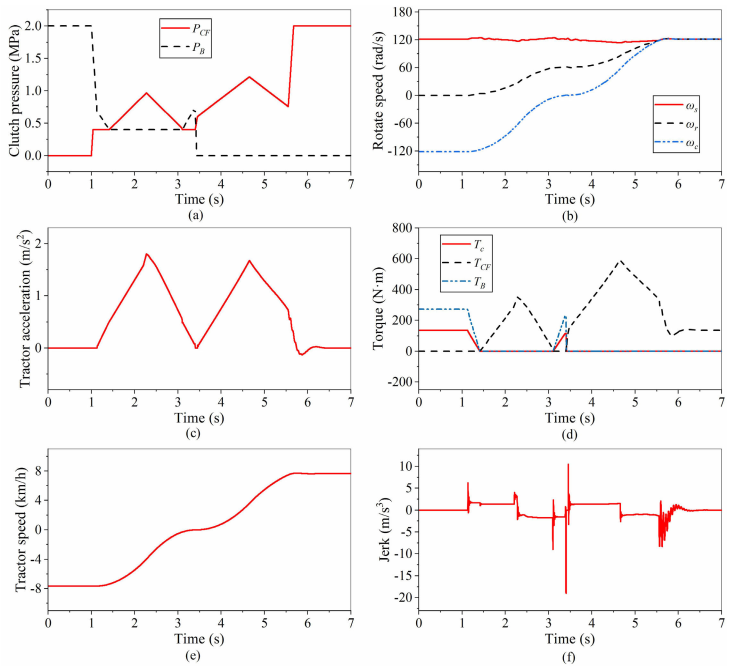

FIGURE 14. **Analysis results of shuttle shifting process from reverse to forward direction.**

the entire process is less than 10 m/s^3, which indicates good comfort in the shuttle shifting process. The impact is ∼20 m/s^3 when the tractor reaches 0 at ∼3.4 s, which is caused by the fact that the clutch pressure is not further increased at that time in order to save the shifting time. The impact here still has room for improvement if a slight increase in shift time can be accepted. The entire shuttle shifting process takes less than 5 s and makes the speed of the tractor change from ∼7.8 km/h to ∼−7.8 km/h, showing the rapidity of the shifting process.

Due to the complex operating conditions of tractors, the current evaluation indicators of power shuttle shifting quality are not perfect enough, and few literature have given specific index values at present. In order to reflect the advantages of the proposed strategy, a simple contrasting simulation is conducted. In the contrasting simulation, the clutch pressure is released directly according to the traditional parking method in the deceleration stage of the shifting process. Then, no further interference is given in this deceleration process so that the tractor free-taxies until it stops. After the vehicle has come to a complete stop, the vehicle will start in reverse. The difference between the two methods can be analyzed comparatively.

As can be seen from the simulation results shown in Figure 13, it takes about 4.3 s for the tractor to travel from 7.8 km/h to a complete stop, which is far beyond the 2.4 s consumed in the control strategy proposed in this study (as shown in Figure 12(e)). Moreover, at the moments when the tractor begins to slow down and stop, the impacts exceed 100 m/s^3 and 400 m/s^3, respectively, with very poor comfort. If the braking force is applied to accelerate the reverse start when the speed is close to zero, then the impact will be higher. Thus, the necessity and advantages of the control method designed in this paper are highlighted by this contrast. Limited by the experimental conditions, this paper mainly conducts research based on simulation. In future research, we will try to obtain more enterprise platform support to improve the applicability of our control strategy and verify its advantages based on experiments.

Figure 14 shows the simulation results of the shifting process from the forward to reverse direction. Compared with the above control process of forward to reverse, in this shifting process, the change rule of the clutch and the brake is exchanged. The brake oil pressure changes from the initial 2 MPa to the final 0 MPa, and the clutch pressure changes from 0 MPa to 2 MPa. Under this action, the rotational speeds of the ring gear and planetary carrier gradually synchronize with the speed of the sun gear (∼120 rad/s) from the initial different rotation speeds (0 and ∼−120 rad/s). Accordingly, the tractor speed changes from ∼−7.8 km/h at the beginning to ∼7.8 km/h at the end.

During the shuttle shifting process, the longitudinal acceleration curve of the tractor presents an M shape, thus avoiding a large acceleration change rate. The entire shuttle shifting process takes less than 5 s, and the impact (jerk) during the entire process (except for the moment when the tractor speed reaches 0) is less than 10 m/s^3. The impact is ∼−20 m/s^3 (which can be improved if necessary, as mentioned above) when the tractor reaches 0 at ∼3.3 s. Therefore, the control strategy designed in this study can realize a quick and comfortable shuttle shifting process from the reverse to forward direction.

In summary, the control strategies designed in this study can realize a fast and comfortable shuttle shift under high gear (including the shifting processes of forward to reverse and reverse to forward). It should be noticed that the shuttle shifts are performed directly under the conditions of a high gear (the tenth gear) and 30% accelerator pedal opening in this study, which are extreme conditions for shuttle shifting. Thus, the time consumption of the entire shifting processes and the sliding time of the hydraulic components (brake B and clutch CF, respectively) that act as brakes appears to be slightly long. Since the clutch/brake friction material, heat dissipation and wear are not studied in this paper, the clutch slip time and the sliding friction are not specifically limited. If it is necessary to further reduce the reversing time and clutch/brake slip time (to reduce the possibility of clutch/brake wear), then the following measures which will not deteriorate the shifting quality can be taken: (a) downshifting the tractor gear before the shuttle shift, (b) reducing the accelerator pedal opening by coordinated control of the engine before the shuttle shift, (c) using the brakes of the tractor itself instead of the hydraulic components (brake B and clutch CF) of the motion inverter to brake the tractor during the shuttle shifting process.

VI. CONCLUSION

In order to automate the shuttle shifting process and headland turn, a power shuttle motion inverter was designed with a planetary gear train and clutch/brake in this study. The dynamics model of the tractor powertrain and a simulation model are established. Two types of shuttle shifting process were designed and analyzed by the lever analogy method. To improve the shifting quality, control strategies for the shuttle shifting processes were formulated by analyzing the changing characteristics of the force exerted on the tractor-implement combination during the shifting processes.

The analysis results showed that quick and comfortable automatic shuttle shifts are achieved during the process of moving from the forward to reverse direction and vice versa. The necessity and advantages of the control method designed in this paper are proved. Research plans for further improving the quality of shifting were proposed.

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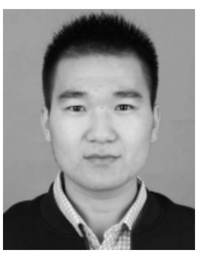

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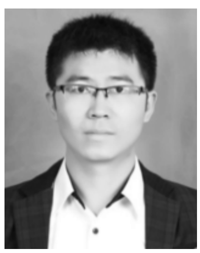

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