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Research Paper

Automatic gear-shifting strategy for fuel saving by tractors based on real-time identification of draught force characteristics

Baogang Li, Dongye Sun, Minghui Hu, Xingyu Zhou, Dongyang Wang, Yu Xia, Yong You*

State Key Laboratory of Mechanical Transmissions, Chongqing University, Chongqing, 400044, China

ARTICLE INFO

Article history:
Received 31 October 2019
Received in revised form
27 January 2020
Accepted 14 February 2020

Keywords:
Automatic gear shifting
Control strategy
Agricultural tractors
Fuel economy
Real-time identification

Adopting an automatic gear-shifting strategy for agricultural tractors can be an important factor affecting fuel consumption, and a reasonable gear-shifting strategy can effectively improve fuel economy. However, as an important part of the resistance exerted on the tractor–implement combination during tillage, the draught force of the implement strongly influences the driving state of the tractor. The changing characteristics of draught force with work speed largely determine the optimal timing of gear shifting. To realise automatic gear shifting to ensure fuel economy, it is necessary to obtain the varying characteristics of draught force in real time. However, draught force fluctuates is affected by many on-site factors, such as the physical properties of soil, operating speed and the design of the implements. A method is proposed using a recursive least-squares algorithm for the real-time identification of the changing characteristics of draught force. By analysing the forces exerted on the tractor–implement combination and the characteristics of the tractor powertrain, a mathematical model of the tractor–implement combination during field work was established. On this basis, four-parameter gear-shifting schedules were developed for tractor with different types of agricultural implements. The operating conditions were designed based on tractor operating characteristics. The validity and accuracy of the real-time identification method for the changing characteristic of draught force was verified. The designed gear-shifting schedules were proven to realise the automatic gear-shifting function according to the real-time working conditions on the premise of ensuring fuel economy and power performance.

1. Introduction

With the development of precision agriculture (Fountas et al., 2015; McBratney, Whelan, Ancev, & Bouma, 2005), tractors, as the main power machinery for field operations, are rapidly developing towards informatisation and intellectualisation (Li, Sun, Hu, & Liu, 2019a). Realising automatic gear shifting of the tractor can reduce driver work intensity for frequent

* *Corresponding author.*
E-mail address: dysun@cqu.edu.cn (D. Sun).
https://doi.org/10.1016/j.biosystemseng.2020.02.008

Nomenclature

Symbols

A_T	Frontal area of tractor–implement combination (m^2)
A, B, C	Machine-specific parameters
b_e	Brake specific fuel consumption (g kW^{-1} h^{-1})
b, d, δ, h	Unloaded tyre section width, diameter, deflection, section height (m)
C_D	Coefficient of air resistance
CI	Cone index (kPa)
f	Coefficient of rolling resistance
F_r, F_d	External force, draught requirement (N)
F_f, F_i, F_a, F_j	Resistances of rolling, slope, air, and acceleration (N)
F_q, F_{q1}, F_{q2}	Driving forces of tractor, front axle and rear axle (N)
F_x	Dimensionless soil texture adjustment parameter
g	Gravitational acceleration (m s^{-2})
i_h, i_g, i_{t1}, i_{t2}	Transmission ratios for pump, gearbox, front axle, and rear axles
I_e, I_{w1}, I_{w2}	Rotational inertia of flywheel, front wheels, and rear wheels (kg m^2)
k_1, k_2	Condition factors
m, m_1	Tractor-implement combination mass, agricultural implement mass (kg)
n_e, n_{to}, n_{ti}	Engine speed, gearbox output speed, gearbox input speed (rpm)
N	Transmission gear
N_{CI}	Mobility number
p_h	Working pressure of hydraulic system (MPa)
q	Flow of hydraulic pump (m^3 s^{-1})
r_1, r_2	Radii of front wheels and rear wheels (m)
s	Slip (travel reduction ratio)
T_c, T_{sh}	Clutch torque, clutch shaft torque (N m)
T_e, T_l, T_{t1}, T_{t2}	Engine torque, accessory equipment torque, parts of engine torque (N m)
T_{to}, T_{ti}	Output and input torques of gearbox (N m)
v, v_T	Tractor speed, theoretical speed (km h^{-1})
W, W_1, W_2	Loads of tractor, front axle, and rear axle (N)

Greek Symbols

β	Throttle opening (%)
θ	Slope of farmland (rad)
$\eta_1, \eta_2, \eta_t, \eta_g$	Mechanical efficiency
ρ	Air density (kg m^{-3})
$\omega_e, \omega_h, \omega_{w1}, \omega_{w2}$	Angular velocities of engine, pump, front wheels, and rear wheels (rad s^{-1})

Abbreviations

PST	Power-shift transmission
PTO	Power take-off
RLS	Recursive least-squares algorithm

driving operations. The driver can focus more on the operation of the implements, thereby improving the quality and efficiency of field work. In addition, this can also reduce fuel and power loss caused by improper manual gear-shifting operations, especially with unskilled drivers, thereby extending the service life of the powertrain (Li, Sun, Hu, & Liu, 2018). Automatic gear shifting is therefore an important development trend for agricultural tractors.

Fuel economy of a vehicle is an important issue under the background of energy conservation and emission reduction. The automatic gear-shifting strategy of agricultural tractors is an important factor affecting fuel consumption, and reasonable gear-shifting strategy can effectively improve fuel economy. Although the research and application of automatic fuel-saving gear-shifting technology in the field of road vehicles is relatively mature (Lei et al., 2017; Liu, Qin, Jiang, & Zhang, 2014; Zhao, Chen, Li, & Lei, 2019), the existing control strategies cannot be directly applied to the tractor because of differences in the working characteristics and powertrains between road vehicles and agricultural tractors. The main differences are as follows: (a) road vehicles are usually used only for road transport, whilst tractors need to carry and operate a variety of farm implements for field operations, so the changing characteristic of the resistances exerted on tractor–implement combination are more complex than that of road vehicles due to characteristics of agriculture implements and soils. (b) Tractors use diesel engines with full-range speed regulation characteristics as power source, whilst road vehicles mainly use gasoline (petrol) engines or diesel engines without full-range speed regulation characteristics, so the torque characteristics are different. (c) Tractors are mainly aimed at smooth operation on farmland (Gotoh et al., 2010), whilst road vehicles also have to consider improved acceleration performance (Su et al., 2018; Zhou, Qin, & Hu, 2017). All these factors affect the formulation of automatic shifting strategies. Thus, it is necessary to develop a gear-shifting schedule of the tractor based on the characteristics of the power source, resistance, and drivetrain.

Adjusting the engine operating position based on engine characteristics is the main method used by existing technology (Chancellor & Thai, 1984; Schrock & Liu, 1990) for tractors with stepped transmission. The engine state is used as the gear-shifting parameter in existing practical applications. The change in the external resistance acting on the tractor–implement combination, especially the draught force of implement, plays an important role in the engine operating state. However, the external resistance exerted on the tractor–implement combination, and its changing characteristics, are unknown during tillage due to the differences that occur in soil properties and types of agricultural implement used. In order to avoid shift cycle (frequent switching between two adjacent gears) caused by these unknown changes in resistance characteristics, a large safety threshold has to be set when developing an automatic shift strategy. This makes it hard to keep the engine working in the desired region of the fuel consumption contour map by automatic shifting and thus fuel consumption is increased. For tractors working under fixed depth control, once the transmission gear changes, the working speed of the tractor–implement combination changes, and the draught force of implement may change. In order to ensure the fuel economy and power performance of the tractor, the engine operating point should be located in the desired region after gear shifting. Therefore, it is necessary to predict, in real time, the change in the running resistance following gear changes as well as its influence on the running state of the engine, thereby realising automatic gear shifting with better fuel economy. To implement this, a gear-shifting schedule needs to be established that can ensure that the engine operates in the desired region before and after the gear shifting, thereby ensuring not only better fuel economy but also sufficient backup torque. The premise is that the changing characteristics of the draught force should be obtained during the tractor operation.

The draught force of an implement is affected by many factors in field work with a tractor–implement combination. Many experimental studies have shown that soil physical properties, characteristics of farming tools, tillage depth, implement width, and field speed all affect the draught force generated during operation. Kichler, Fulton, Raper, McDonald, and Zech (2011) investigated the effects of ground speed (transmission gear selection) on draught force, fuel costs, and other equipment performance variables using two deep tillage implements. Their results, for two subsoil tillage implements, indicated that the ground speed (transmission gear selection) impacted on tractor fuel usage, and thereby costs, along with the productivity rate. Draught and energy requirement measurements were made Kheiralla, Yahya, Zohadie, and Ishak (2004) using an instrumented tractor for mouldboard ploughing, disc ploughing, disc harrowing, and rotary tilling in Serdang sandy clay loam soil. The effects of travel speed and tillage depth or rotor speed upon the measured draught and power requirements were investigated. Sahu and Raheman (2006) developed a new methodology to estimate the draught requirements of cultivator with disc harrow and mouldboard plough with disc gang combination in a sandy clay loam soil from knowledge of the draught requirements of individual tillage implements in the same soil and the draught utilisation ratio of the rear passive sets of these combination tillage implements. The American Society of Agricultural and Biological Engineers (ASABE) has published draught estimation equation for various agricultural implements (ASABE, 2011). This formula uses a variety of field and machine coefficients to estimate implement draught. Kroulík Chyba & Brant (2015) carried out measurements of tensile force using a tractor's built-in sensor and an external sensor connected between machines.

As a multi-purpose power machine, the tractor needs to carry different agricultural tools to work respectively. Because the soil physical characteristics (cone index, moisture content, electrical conductivity, percentage of clay and sand, etc.) and agricultural implement characteristics are unknown in field work, the draught force of the implement cannot be obtained by existing empirical formulas. In addition, the implement draught force fluctuates greatly during work (Regier, Schrock, Thompson, & Yang, 1986; Sharifi, 2016) and therefore it is difficult to obtain the change regularity of implement draught force with field speed in real time. However, as mentioned above, changes in the characteristics of draught force are an important factor to determine the timing of gear shifting. It is necessary to obtain this change characteristic in real time.

To solve the aforementioned problems, in this study, the factors affecting draught force at a certain speeds are summarised as a characteristic parameter, which is defined as the condition factor. It can reflect change regularity of implement draught force with the working speed. An automatic gear-shifting schedule of tractor based on real-time identification of the condition factor is developed. Because the recursive least-squares algorithm (RLS) has the characteristics of rapid convergence, low calculation time and a low storage demand (Chun, Kim, & Lee, 1998; Guo, Ljung, & Priouret, 1993) and is therefore an excellent method for real-time (online data updating) system identification (Ljung, 2002). Therefore, in this study, a real-time identification method which uses the data from tractor sensors and the RLS algorithm was designed to identify the condition factor during tractor–implement combination work. Thus, the automatic gear shifting strategy based on real-time identification of draught force characteristic is realised, which can ensure fuel economy and power performance for the tractors operating in traction.

2. Development of gear-shifting schedule

To develop a gear-shifting schedule for a tractor with different types of agricultural implements operating under the fixed depth tillage, requires analysis of the forces exerted on the tractor–implement combination by analysing the characteristics of the tractor powertrain and establishing the necessary mathematical models.

2.1. *Mathematical model*

Figure 1 shows the schematic of the forces acting on a four-wheel drive tractor with a farm implement. The longitudinal forces of the tractor–implement combination on the *x* axis need only be considered when the gear-shifting schedule for straight driving is studied.

As Eq. (1) shows, the external resistance of the tractor–implement combination while driving mainly includes the rolling resistance, slope resistance, air resistance, and the draught requirement for the implement (Li et al., 2019b).

$$F_r = F_f + F_i + F_a + F_d, \tag{1}$$

where F_r, F_f, F_i, F_a, and F_d are the resultant external force (N), rolling resistance (N), slope resistance (N), air resistance (N), and draught requirement for implement (N), respectively.

The driving force of the tractor–implement combination is provided by the engine. The mode of all-wheel-drive (AWD) is usually selected in field work. Therefore, the tyres of the front and rear axles all have driving force:

$$F_q = F_{q1} + F_{q2} = \frac{T_{t1} i_{t1} \eta_1 - I_{w1} \dot{\omega}_{w1}}{r_1} + \frac{T_{t2} i_{t2} \eta_2 - I_{w2} \dot{\omega}_{w2}}{r_2} = F_r + F_j, \tag{2}$$

$$T_{t1} + T_{t2} = T_e - I_e \dot{\omega}_e - T_l, \tag{3}$$

$$\omega_e = \omega_{w1} i_{t1} = \omega_{w2} i_{t2}, \tag{4}$$

where F_q, F_{q1}, and F_{q2} are the driving forces (N) of the tractor, front axle and rear axle, respectively; T_{t1} and T_{t2} are the parts

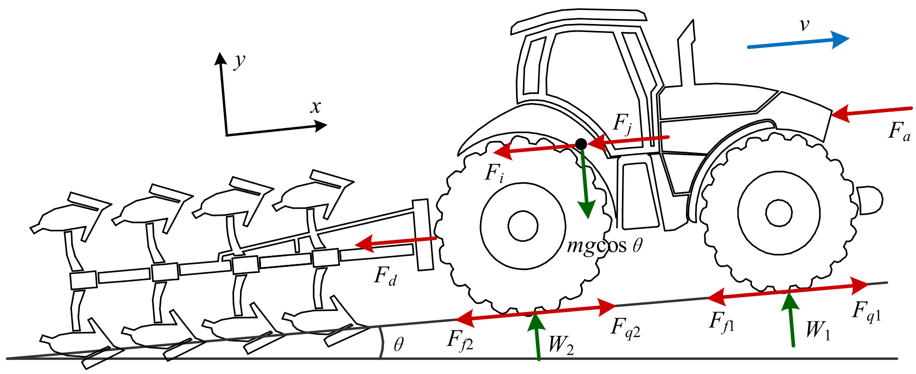

Fig. 1 – Schematic of forces exerted on the tractor–implement combination.

of engine torque (N m) used for front and rear axles, respectively; i_{t1} and i_{t2} are the transmission ratios for front and rear axles, respectively; η_1 and η_2 are the mechanical efficiency; I_{w1} and I_{w2} are the rotational inertia (kg m^2) of the front wheels and rear wheels, respectively; ω_{w1} and ω_{w2} are the angular velocities (rad s^{-1}) of the front wheels and rear wheels, respectively; r_1 and r_2 are the radii (m) of the front wheels and rear wheels, respectively; F_j is the acceleration resistance (N); T_e is the engine torque (N m); I_e is the rotational inertia of the flywheel (kg m^2); ω_e is the angular velocity of the engine (rad s^{-1}); and T_l is the torque of accessory equipment (N m).

The rolling resistance, slope resistance, air resistance and the acceleration resistance can be expressed by Eqs. (5)–(8).

$$F_f = F_{f1} + F_{f2} = W_1 f + W_2 f = mgf\cos\theta, \tag{5}$$

$$F_i = mg\sin\theta, \tag{6}$$

$$F_a = \frac{1}{2}\rho C_D A_T v^2, \tag{7}$$

$$F_j = m\frac{dv}{dt}, \tag{8}$$

where F_{f1} and F_{f2} are the rolling resistances (N) acting on front wheels and rear wheels, respectively; W_1 and W_2 are the loads (N) of front axle and rear axle, respectively; f is the coefficient of rolling resistance; m is the mass of the tractor–implement combination (kg); g is the gravitational acceleration (m s^{-2}); θ is the slope of the farmland (rad); ρ is the air density (kg m^{-3}); C_D is the coefficient of air resistance; A_T is the frontal area (m^2) of the tractor–implement combination; and v is the working speed of the tractor (m s^{-1}).

During field work, air resistance can be neglected because the working speed of the tractor–implement combination is low and the air resistance is not significant. The parameters of the tyre and soil properties have a significant effect on soil–traction and tractive performance (Zoz & Grisso, 2003). Dwyer (1984) reported the equations of coefficient of rolling resistance as follows:

$$f = \frac{F_f}{W} = 0.049 + \frac{0.287}{N_{CI}}, \tag{9}$$

$$N_{CI} = \frac{CIbd}{W}\sqrt{\frac{\delta}{h}}\frac{1}{1+\frac{b}{2d}}, \tag{10}$$

where W is the vertical wheel load (N), N_{CI} is the mobility number, CI is the cone index (kPa), b is the unloaded tyre section width (m), d is the unloaded tyre diameter (m), δ is the tyre deflection (m), and h is the tyre section height (m).

This paper mainly studies the tillage of tractor without operating the power take-off (PTO) shaft. So the equivalent torque of accessory equipment is mainly the torque consumed by hydraulic system. This part of the torque can be expressed by

$$T_l = \frac{qp_h}{\omega_h i_h}, \tag{11}$$

where q is the flow of hydraulic pump (m^3 s^{-1}), p_h is working pressure of the hydraulic system (MPa), ω_h is the hydraulic pump angular velocity (rad s^{-1}), and i_h is the speed ratio of engine to pump.

The draught force F_d is required to pull agricultural implements (N) operated at shallow depths in horizontal direction of running, which is correlated to the width of implement and the speed at which it is pulled. Typical draught requirements can be calculated as Eq. (12) and average draught requirement parameters are summarised for most tillage machines (ASABE, 2011; Li et al., 2019b) by:

$$F_d = F_x(A + Bv + Cv^2)W_M T_S, \tag{12}$$

where F_x is a dimensionless soil texture adjustment parameter; A, B, and C are machine-specific parameters; W_M is the machine width (m); and T_S is the tillage depth (cm).

2.2. *Gear-shifting principle*

Maintaining a safe and smooth operation is the primary goal of the tractor–implement combination during field work. Improving the fuel economy and ensuring the power of the tractor–implement combination is an important pursuit for farmers. According to the driver's intention and working conditions, the control system needs to automatically adjust the transmission gear to make the engine work in the desired operating region, thereby meeting the goals of guaranteeing the relatively stability of the operation, power performance and fuel economy. Therefore, adjusting the engine operating point under different driver's intentions and working conditions is the main basis for formulating gear-shifting strategy.

Figure 2(a) shows the full-load characteristics of the diesel engine. The green line is the engine output torque under full throttle opening (often called full accelerator pedal opening). According to Eqs. (2)–(4), the torque exerted on the front wheels can be equivalently converted to the rear wheels. Therefore, the driving force of the tractor in stable operation can be expressed as:

$$F_q = \frac{(T_e - T_l)i_{t2}\eta_t}{r_2}, \tag{13}$$

where η_t is the mechanical efficiency of driveline.

The theoretical traveling speed of the tractor v_T (m s^{-1}) can be expressed by Eq. (14):

$$v_T = \frac{\omega_e r_2}{i_{t2}}, \tag{14}$$

where ω_e is the engine angular velocities (rad s^{-1}).

Therefore, the relationship between the driving force and the theoretical vehicle speed using different gears at full throttle opening can be obtained. As shown in Fig. 2(b), the three colour curves indicate the driving forces of three adjacent gears at full throttle opening. They are the high gear H with small speed ratio, medium gear M, and low gear L with large ratio, respectively. The black curve is the external resistance acting on the tractor–implement combination.

The full-range speed regulation characteristic of diesel engine can ensure relatively stable engine speed even if tractor operates under varying load, and this can avoid engine stalling. In field work, it is usually expected that a tractor diesel engine works on the governor response curves (sometimes called 'droop' curves), which are approximately straight lines at different throttle openings (Grisso et al., 2009; Schrock & Liu, 1990). According to the engine universal characteristics, when operating near the upper end of the governor response curves, the engine has a higher load rate as well as better fuel economy. Therefore, the automatic gear-shifting schedule is designed according to the following basic principles.

Under any throttle opening, once the engine operating torque is greater than the upper end of the governor response curves, the gear will downshift, thereby reducing the operating torque of the engine, avoiding a drastic drop in engine speed or even engine stall. Therefore, the engine speed (or engine torque) at the upper end of the governor response curves is used as the downshift speed (or torque). By contrast, to increase engine load rate and improve fuel economy, when the engine torque reaches a lower level, the tractor will upshift. Therefore, the key parameter formulating the upshift schedule is to determine this lower-level torque.

At full throttle opening, to determine the lower level torque (point C in Fig. 2(a)), on the one hand, it is necessary to ensure that the engine torque must not exceed the torque at point A after gear upshifts, otherwise it will cause a shift cycle (frequent switching between two adjacent gears). On the other hand, the engine should be kept at a higher load rate as much as possible, thereby ensuring lower fuel consumption. Therefore, the development of the upshift schedule requires analysis of the resistance changing regularity of the tractor operating under various gears (i.e. predicting the resistance with different gear positions).

As Fig. 2(b) illustrates, the tractor works in gear M under full throttle opening first. When the resistance exceeds the driving force indicated by the point a_2, the tractor will downshift to gear L to keep the engine operating point below the upper end of the governor response curve. Then, the speed ratio of driveline will increase and the work speed will decrease. At this moment, the resistance is greater than the force indicated by point c_1. Correspondingly, point c_1 is the upshift point of gear L. Once the resistance is lower than the torque indicated by point c_1, the gear will upshift.

Therefore, gear-shifting conditions under full throttle opening can be expressed by the following formulae:

$$\begin{cases} Geardown, & T_e > T_A(N) \\ Gearup, & T_e < T_C(N) = f(T_A(N+1), v) \end{cases}, \tag{15}$$

where N is the transmission gear, T_A is the torque (N m) indicated by point A in Fig. 2(a), T_C is the torque (N m) indicated by point C in Fig. 2(a).

Therefore, the difference in resistance before and after gear shifting, namely the shape of resistance curve in Fig. 2(b), is an important factor affecting the gear-shifting schedule.

According to Eqs. (1) and (5)–(7) and (9) and (12), the total external resistance exerted on the tractor is:

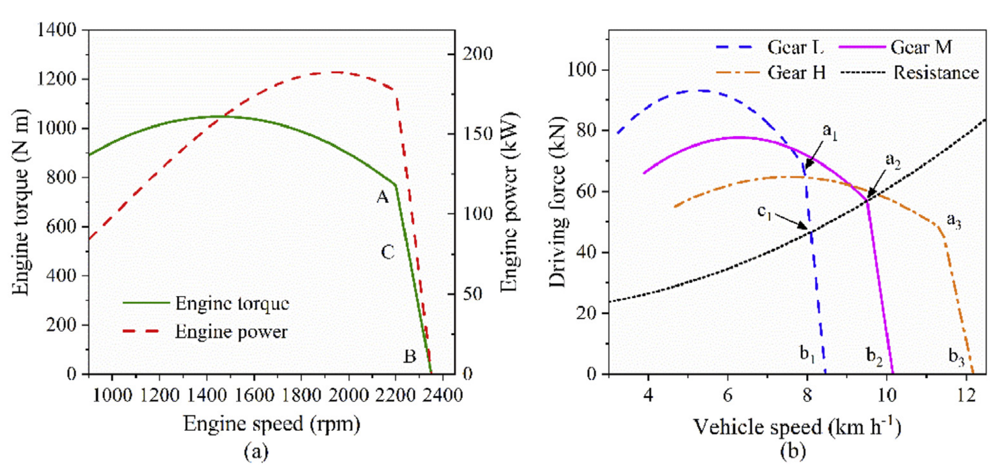

Fig. 2 – Schematic of gear shifting: (a) Full-load characteristics of the engine; (b) Driving forces of gears (L, M, H) and resistance.

$$F_r = F_f + F_i + F_d = mg\sin\theta + mgf\cos\theta + F_x(A + Bv + Cv^2)W_M T_S. \quad (16)$$

In Eq. (16), only the last item (the draught force F_d) is related to the field speed (transmission gear), which determines the shape (variation trend) of the resistance curve in Fig. 2(b).

Based on the model and the average parameters of draught requirements summarised by ASABE (2011), typical agricultural implements were classified into three types in this study.

(1) *Type one* includes mouldboard ploughs, subsoilers etc. where B = 0 and C≠0 in Eq. (12).

$$F_d = F_x(A + Cv^2)W_M T_S = k_1 v^2 + b_1, \quad (17)$$

where $k_1 = CF_x W_M T_S$, $b_1 = AF_x W_M T_S$.

(2) *Type two* include sweep ploughs, disk harrows etc. where B ≠ 0 and C = 0 in Eq. (12).

$$F_d = F_x(A + Bv)W_M T_S = k_2 v + b_2, \quad (18)$$

where $k_2 = BF_x W_M T_S$, $b_2 = AF_x W_M T_S$.

(3) *Type three* includes the land planes, spring tooth harrows, etc. where B = 0 and C = 0 in Eq. (12).

$$F_d = AF_x W_M T_S = b_3 \quad (19)$$

Slip (travel reduction ratio) is an important factor affecting the traction efficiency (Zoz & Grisso, 2003), and it is used as gear-shifting parameter in this study. The slip s is calculated from the actual speed v (km h^{-1}) and the theoretical speed v_T (km h^{-1}):

$$s = \frac{v_T - v}{v_T}. \quad (20)$$

Therefore, the external resistance of the tractor can be expressed as

$$F_r = \begin{cases} k_1[(1-s)v_T]^2 + b_1 + mg\sin\theta + mgf\cos\theta = K_1 v_T^2 + C_1, & \textit{Type one} \\ k_2(1-s)v_T + b_2 + mg\sin\theta + mgf\cos\theta = K_2 v_T + C_2, & \textit{Type two} \\ b_3 + mg\sin\theta + mgf\cos\theta = C_3, & \textit{Type three} \end{cases}, \quad (21)$$

or,

$$F_r = \begin{cases} K_1 v_T^2 + C_1, & \textit{Type one} \\ K_2 v_T + C_2, & \textit{Type two} \\ C_3, & \textit{Type three} \end{cases}, \quad (22)$$

where $K_1 = k_1$ (1-s)^2, $K_2 = k_2$ (1-s), $C_1 = b_1 + mg\sin\theta + mgf\cos\theta$, $C_2 = b_2 + mg\sin\theta + mgf\cos\theta$.

The change in the regulation of resistance is caused by the change in field speed and is the factor that affects gear shifting. For the *Type one* farm implements, the coefficient k_1 determines the change in resistance (variation trend) when the tractor speed (gear) changes. Similarly, k_2 determines variation trend of resistance of the second type farm implements. They are the important parameters for gear-shifting schedule. The coefficients k_1 and k_2 are related to the soil physical properties, implement characteristics, tillage depth, machine width, etc. Therefore, k_1 and k_2 are called condition factors in this paper.

The governor response curve is approximately a straight line at any throttle opening. Thus, under a certain gear and throttle opening, the driving force can be expressed as a function of the theoretical velocity.

$$F_q = c_1 v_T + c_2 \quad (23)$$

The intersection coordinate of driving force (under gear *N*) and the resistance curve (passing the turning point of driving force line of gear *N*+1) is the upshift coordinate of gear *N*. For Eqs. (22) and (23), making $F_r = F_q$, the upshift coordinate (v_{T0}, F_{q0}) can be obtained.

$$v_{T0} = \begin{cases} c_1 + \sqrt{c_1^2 - 4K_1(C_1 - c_2)} \Big/ (2K_1), & \textit{Type one} \\ (c_2 - C_2)/(K_2 - c_1), & \textit{Type two} \\ (C_3 - c_2)/c_1, & \textit{Type three} \end{cases} \quad (24)$$

$$F_{q0} = c_1 v_{T0} + c_2 \quad (25)$$

In Eqs. (24) and (25), coefficients c_1 and c_2 are determined from the engine characteristics and driveline speed ratio, which can be directly derived according engine test data and gearbox parameters. The constants C_1, C_2, and C_3 represent the part of resistance independent of the speed. In a certain gear, with a specified throttle opening and condition factor, they can be calculated by the turning point coordinate of the driving force line and Eq. (22).

Based on the above method, the gear-shifting conditions for each gear with different throttle openings, different working condition factors and different slip ratios were successively calculated. The results were developed into four-dimensional gear-shifting schedules for automatic gear-shifting control with different type of tools. The gear-shifting parameters included throttle opening, theoretical speed (or transmission output torque), slip, and the condition factor. The value of the condition factor should be identified in real time during field work. Before the condition factor is identified in practical application, the throttle opening and engine speed are used as parameters for automatic gear shifting.

2.3. *Gear-shifting schedule*

According to the above-mentioned gear-shifting principle, the four-parameter gear-shifting schedules were formulated. Because the four-parameter shifting schedules are difficult to express directly graphically, the drawing method of fixed parameters was adopted. The gear-shifting schedule for the *Type two* agriculture tools is shown in this paper as a representative. Figure 3(a) shows the gear upshift map when condition factor is 1400 and slip is 10%. It can be converted into a two-dimensional map, as shown in Fig. 3(b), in which the seven curves from right to left represent the upshift conditions of the transmission gears from 5 to 11 under any throttle opening, respectively. Once the transmission output torque is below the torque indicated by these upshift curves, upshifting is performed.

Because the theoretical travel speed can be obtained from a rotation sensor, the theoretical travel speed can be used instead of the transmission output torque as a gear–shift parameter. Fig. 3(c) and Fig. 3(d) show the gear upshift maps using the tractor's theoretical speed as one of the gear–shift parameters. The seven curves in Fig. 3(d) from left to right represent the upshift conditions of transmission gears from 5 to 11 under any throttle opening, respectively. Once the tractor's theoretical speed exceeds the speed indicated by these upshift curves, upshifting is performed.

Figure 4 shows the gear upshift maps when throttle opening is 100% and slip is 10%. The seven curves in Fig. 4(b) from right to left represent the upshift conditions of transmission gears from 5 to 11 under different values of condition factor, respectively. It can be seen that high output torque of transmission and high condition factor require low gear. This agrees with expectations.

The gear downshift maps are shown in Fig. 5, in which the downshift conditions of transmission gears from 6 to 12 under any throttle opening are shown. A three-dimensional map is sufficient to express the downshift schedule because it has nothing to do with the slip and the condition factor.

3. Parameter identification

The agricultural implement characteristic and soil physical properties have important effect on draught force as well as gear-shifting strategy. In the actual work, it is necessary to identify the condition factor in real time to select the optimal transmission gear. Combined with the data from sensor of modern electronic tractors (Kroulík, Chyba, & Brant, 2015; Scarlett, 2001), the method of identifying the condition factor by RLS is described in this section.

A tractor carrying the *Type two* of agricultural implement is selected as an example to describe the identification method. The draught requirement of a second type implement can be expressed as follows:

$$F_d = F_x(A + Bv)W_M T_S + m_1\dot{v} = k_2 v + b_2 + m_1\dot{v}, \tag{26}$$

where m_1 is the mass of agricultural implement (kg).

$$z = k_2 v + b_2 = [v \quad 1]\begin{bmatrix} k_2 \\ b_2 \end{bmatrix} = \mathbf{h}^{\mathrm{T}}\boldsymbol{\theta}, \tag{27}$$

where $\mathbf{h} = [v \quad 1]^{\mathrm{T}}$ is the measured vector which can be obtained by speed sensor of tractor, $\boldsymbol{\theta} = [k_2 \quad b_2]^{\mathrm{T}}$ is the vector to be identified, $z = F_d - m_1\dot{v}$ is the real-time value.

During the field operations, the draught force fluctuates greatly. The measurement result of the kth time can be expressed as

$$z(k) = \mathbf{h}^{\mathrm{T}}(k)\boldsymbol{\theta} + e(k), \tag{28}$$

where $e(k)$ is the error between the measured value and the model.

Assume that a total of L group data is collected. Criterion function is defined as

$$J(\boldsymbol{\theta}) = \sum_{k=1}^{L}[\tilde{z}(k)]^2 = \sum_{k=1}^{L}\left[z(k) - \mathbf{h}^{\mathrm{T}}(k)\boldsymbol{\theta}\right]^2 = (\mathbf{Z}_L - \mathbf{H}_L\boldsymbol{\theta})^{\mathrm{T}}(\mathbf{Z}_L - \mathbf{H}_L\boldsymbol{\theta}), \tag{29}$$

where $\mathbf{Z}_L = [z(1)z(2) \quad \ldots \quad z(L)]^{\mathrm{T}}$ and $\mathbf{H}_L = [\mathbf{h}^{\mathrm{T}}(1)\mathbf{h}^{\mathrm{T}}(2) \quad \ldots \quad \mathbf{h}^{\mathrm{T}}(L)]^{\mathrm{T}}$.

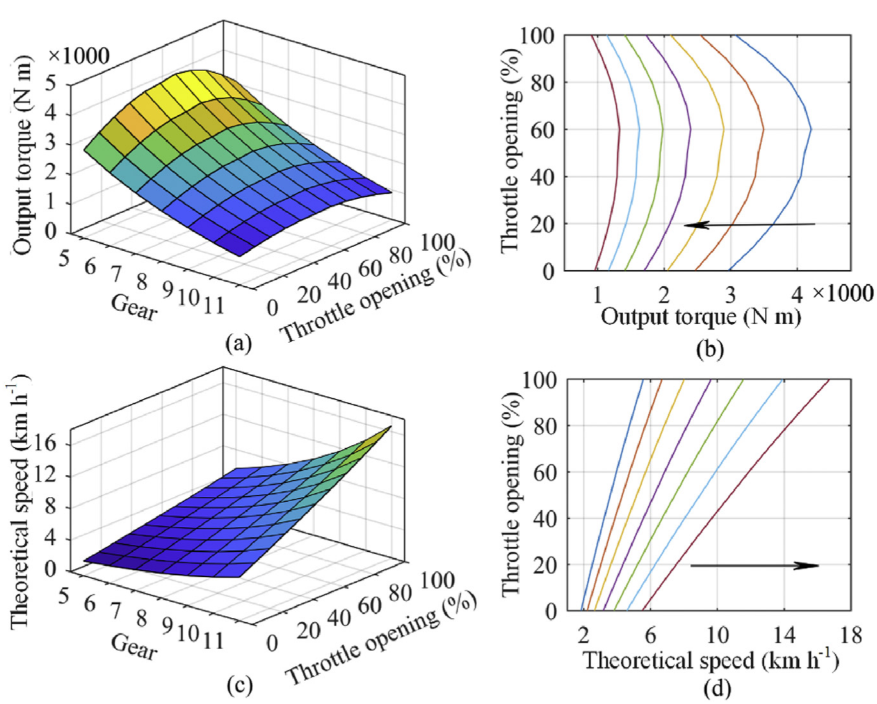

Fig. 3 – The gear upshift maps when condition factor is 1400 and slip is 10%.

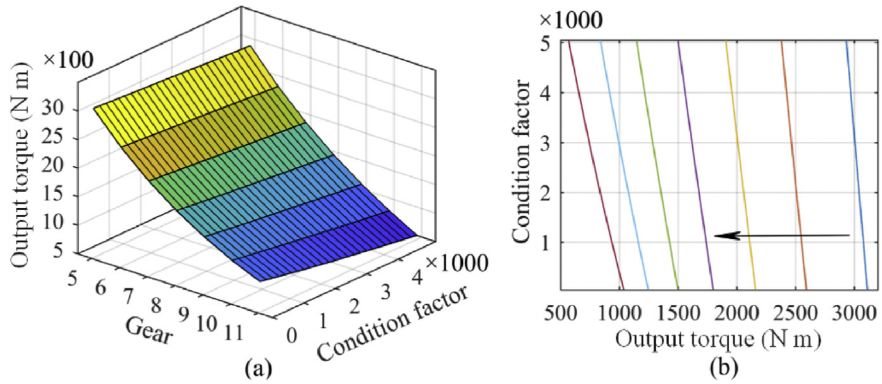

Fig. 4 – The gear upshift maps when throttle opening is 100% and slip is 10%.

The minimum value of the criterion function can be solved by

$$\left.\frac{\partial J(\boldsymbol{\theta})}{\partial \boldsymbol{\theta}}\right|_{\boldsymbol{\theta}=\widehat{\boldsymbol{\theta}}} = \frac{\partial}{\partial \boldsymbol{\theta}}(\mathbf{Z}_L - \mathbf{H}_L\boldsymbol{\theta})^{\mathrm{T}}(\mathbf{Z}_L - \mathbf{H}_L\boldsymbol{\theta}) = 0, \tag{30}$$

where $\widehat{\boldsymbol{\theta}}$ is the estimated value.

$$\widehat{\boldsymbol{\theta}} = \left(\mathbf{H}_L^{\mathrm{T}}\mathbf{H}_L\right)^{-1}\mathbf{H}_L^{\mathrm{T}}\mathbf{Z}_L. \tag{31}$$

In order to identify the parameters in real time and reduce the data storage capacity, a real-time recursion solution is needed. Eq. (31) can be expressed by

$$\widehat{\boldsymbol{\theta}} = \mathbf{P}(L)\mathbf{H}_L^{\mathrm{T}}\mathbf{Z}_L, \tag{32}$$

where square matrix $\mathbf{P}(L) = \left(\mathbf{H}_L^{\mathrm{T}}\mathbf{H}_L\right)^{-1}$.

Assume that the parameter value estimated by the measured values of the previous k group is $\widehat{\boldsymbol{\theta}}(k)$, then

$$\begin{aligned}\widehat{\boldsymbol{\theta}}(k) &= \left(\mathbf{H}_k^{\mathrm{T}}\mathbf{H}_k\right)^{-1}\mathbf{H}_k^{\mathrm{T}}\mathbf{Z}_k = \mathbf{P}(k)\left[\sum_{i=1}^{k}\mathbf{h}(i)z(i)\right] \\ &= \mathbf{P}(k)\left[\mathbf{P}^{-1}(k-1)\widehat{\boldsymbol{\theta}}(k-1) + \mathbf{h}(k)z(k)\right] \\ &= \mathbf{P}(k)\left\{\left[\mathbf{P}^{-1}(k) - \mathbf{h}(k)\mathbf{h}^{\mathrm{T}}(k)\right]\widehat{\boldsymbol{\theta}}(k-1) + \mathbf{h}(k)z(k)\right\} \\ &= \widehat{\boldsymbol{\theta}}(k-1) + \mathbf{P}(k)\mathbf{h}(k)\left[z(k) - \mathbf{h}^{\mathrm{T}}(k)\widehat{\boldsymbol{\theta}}(k-1)\right].\end{aligned} \tag{33}$$

If gain matrix $\mathbf{K}(k) = \mathbf{P}(k)\mathbf{h}(k)$, then

$$\widehat{\boldsymbol{\theta}}(k) = \widehat{\boldsymbol{\theta}}(k-1) + \mathbf{K}(k)\left[z(k) - \mathbf{h}^{\mathrm{T}}(k)\widehat{\boldsymbol{\theta}}(k-1)\right], \tag{34}$$

$$\mathbf{K}(k) = \mathbf{P}(k-1)\mathbf{h}(k)\left[\mathbf{h}^{\mathrm{T}}(k)\mathbf{P}(k-1)\mathbf{h}(k) + 1\right]^{-1}, \tag{35}$$

$$\mathbf{P}(k) = \left[\mathbf{I} - \mathbf{K}(k)\mathbf{h}^{\mathrm{T}}(k)\right]\mathbf{P}(k-1), \tag{36}$$

where $\mathbf{I}$ is the identity matrix.

When the forgetting factor RLS (FFRLS) algorithm is adopted, the criterion function becomes

$$J(\boldsymbol{\theta}) = \sum_{k=1}^{L}[\tilde{z}(k)]^2 = \sum_{k=1}^{L}\left\{\rho^{L-k}\left[z(k) - \mathbf{h}^{\mathrm{T}}(k)\boldsymbol{\theta}\right]^2\right\}, \tag{37}$$

where ρ is the forgetting factor.

Accordingly, the gain matrix $\mathbf{K}(k)$ and the square matrix $\mathbf{P}(k)$ are expressed by

$$\mathbf{K}(k) = \mathbf{P}(k-1)\mathbf{h}(k)\left[\mathbf{h}^{\mathrm{T}}(k)\mathbf{P}(k-1)\mathbf{h}(k) + \rho\right]^{-1}, \tag{38}$$

$$\mathbf{P}(k) = \frac{1}{\rho}\left[\mathbf{I} - \mathbf{K}(k)\mathbf{h}^{\mathrm{T}}(k)\right]\mathbf{P}(k-1). \tag{39}$$

The tractor collects the corresponding data in real time using the sensor when working in the field. According to the above method, $\widehat{\boldsymbol{\theta}}(k)$ can be calculated online in real time. Then the condition factor k_2 is obtained for automatic shift control. In actual operation, once the implement starts working, the program for parameter identification starts. Figure 6 shows the flow chart for parameter identification. The initial values are set according to Eqs. (40) and (41).

$$\mathbf{P}(0) = d^2\mathbf{I}, \tag{40}$$

$$\widehat{\theta}(0) = \left[\,k_2(0) \quad b_2(0)\,\right]^{\mathrm{T}}, \tag{41}$$

where d is a sufficiently large real number, k_2 (0) and b_2 (0) are sufficiently small real numbers (0.1 and 0.1 are used in this study).

4. Analysis and verification

Simulation analysis is widely used in the study of vehicle automatic control strategies (Hu et al., 2018; Kulkarni, Shim, & Zhang, 2007; Zhou et al., 2017). For tractors, Schrock and Liu (1990) compared closed-loop control algorithms and transmission types on the basis of fuel economy by computer simulation. A two-wheel-drive (2WD) tractor simulation model was established (Kolator & Białobrzewski, 2011 and verified by experimental data. The process of tractor motion inversion was analysed by a simulation model of a tractor with a power-shuttle transmission system by Raikwar, Tewari, Mukhopadhyay, Verma, and Sreenivasulu Rao (2015). Therefore it can be seen that, the method of combining experimental data with simulation analysis is well established and was used for this research.

4.1. *Model establishment*

The research object of this paper was a high-power agricultural tractor, whose basic parameters are shown in Table 1.

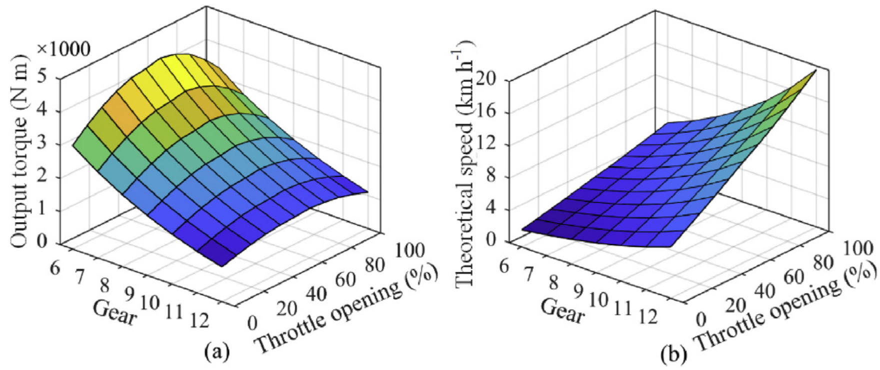

Fig. 5 – The gear downshift maps.

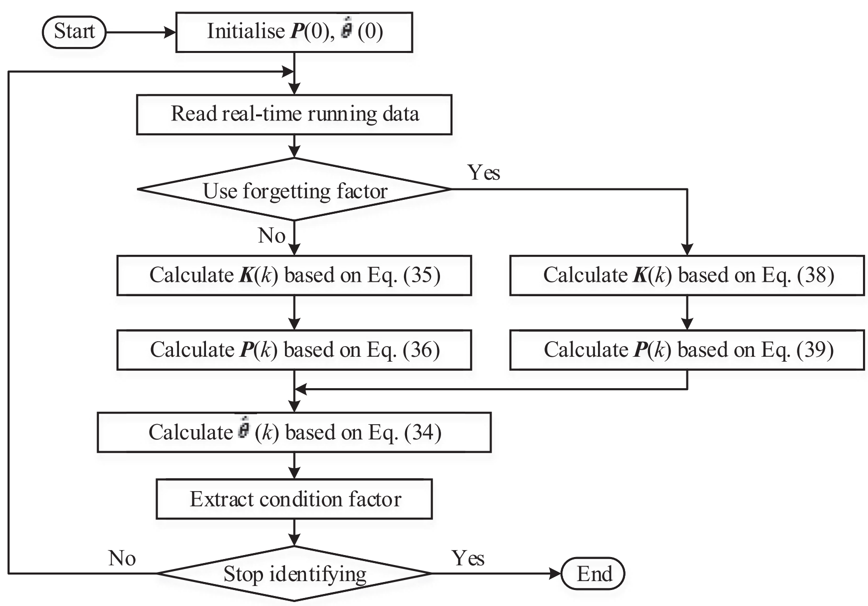

Fig. 6 – Flow chart for parameter identification.

A diesel engine of 175 kW power with a rated torque of 765 N m at a rated engine speed of 2200 rpm is used in this paper, with a full-range speed-governing feature. The mapping characteristics of the engine from test data are shown in Fig. 7. As Eqs. (42) and (43) show, the engine torque T_e can be considered a function of the engine speed and throttle opening, and the brake specific fuel consumption (BFSC) can be considered a function of engine speed and engine torque.

$$T_e = f(n_e, \beta), \tag{42}$$

$$b_e = f(n_e, T_e), \tag{43}$$

where n_e is the engine speed (rpm), β is the engine throttle opening (%), b_e is the brake specific fuel consumption (g kW^{-1} h^{-1}).

A power-shift transmission (PST) was used in this study. The gears for commonly used travel speeds were selected because the control strategy of the tractor in field operations is the main focus of this study. The speed ratios of the commonly used gears for field work, and the corresponding theoretical travel speed information, are shown in Fig. 8. The power-shifted speeds and the input and output relationship of the gearbox can be expressed as

$$T_{to} = T_{ti} i_g \eta_g, \tag{44}$$

$$n_{to} = \frac{n_{ti}}{i_g}, \tag{45}$$

where T_{to} and T_{ti} are the output and input torques (N m) of the gearbox respectively, n_{to} and n_{ti} are the output and input

Table 1 – Tractor parameters.

Parameters	Value
Rated power of engine (kW)	175
Rated speed of engine (rpm)	2200
Radius of rear wheels r_2 (m)	0.95
Mass of tractor–implement combination m (kg)	10,000
Mass of farm implements m_1 (kg)	2000
Mechanical efficiency η_t	0.86

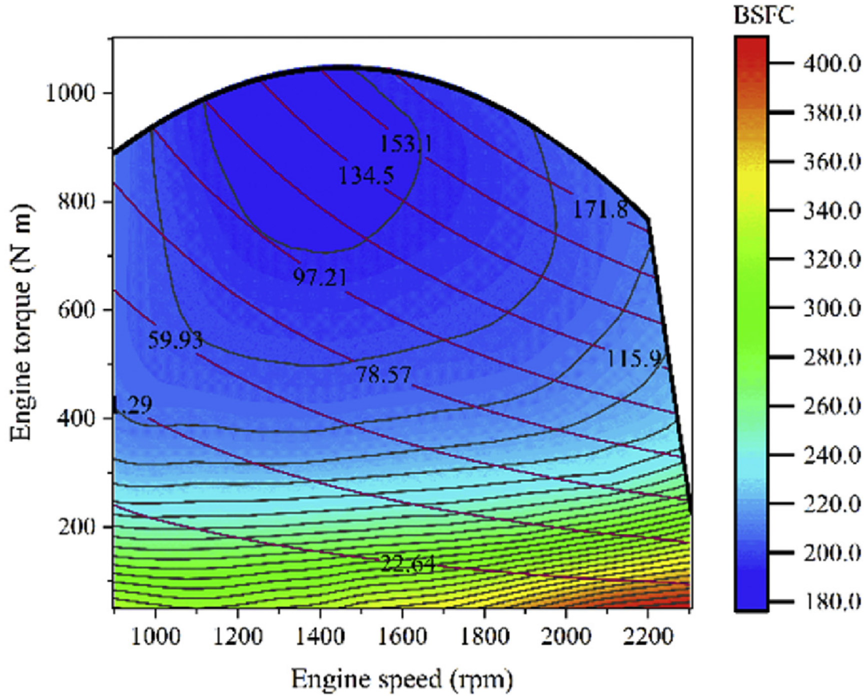

Fig. 7 – Engine mapping characteristics (see Table 1 for tractor parameters).

speeds (rpm) of the gearbox respectively, i_g is the speed ratio of the gearbox, η_g is the efficiency of the gearbox.

In this paper, the gear-shifting process is neglected, so the complex clutch model is simplified as a Boolean switch. Thus the torque transferred by the gear clutch can be expressed by

$$T_c = \begin{cases} 0, & \text{separated} \\ T_{sh}, & \text{engaged} \end{cases}, \tag{46}$$

where T_c and T_{sh} are the torques (N m) of the gear clutch and input shaft, respectively.

The draught force of implement is fluctuant in field work. The fluctuation amplitude and frequency of the draught force are related to soil properties and the characteristics of the implements. Based on the existing research (Kroulík et al., 2015; Regier et al., 1986; Sirjacobs, Hanquet, Lebeau, & Destain, 2002), the draught force fluctuation characteristics of various kind implements were designed in this study. For

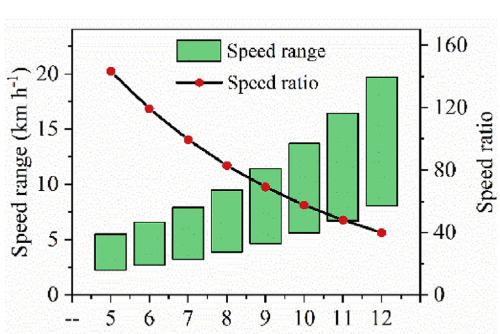

Fig. 8 – Speed ratios and speed ranges of the transmission gears.

instance, Fig. 9 shows the draught force of disk harrow during secondary tillage under different working conditions. In Fig. 9(a), the blue curve shows the draught force of disk harrow with tillage depth of 200 mm and width of 4 m under the working speed of 12 km h^{-1} in farmland with fine soils (high clay content), and the red curve indicates the draught force under the speed of 8 km h^{-1}. At the working speed of 12 km h^{-1}, when the disc harrow (with tillage depth of 200 mm and width of 4 m) works from field with fine soils to field with coarse soils (sandy soils), the changing of draught force is indicated by the blue curve in Fig. 9(b). Although draught force fluctuations will vary under different test conditions, the identification method is universal.

A simulation model of the tractor was built with the MATLAB 2017a (MathWorks, Natick, MA, USA) in accordance with the all above-mentioned mathematical models. This model mainly includes modules for the driver, gear clutch, engine, transmission, resistance, control system, and display. The module of the control system was built using MATLAB and its tool Stateflow to control various modules for the process of tractor operation.

4.2. *Automatic control strategy*

The control flow chart for the automatic control strategy is shown in Fig. 10. In practical operations, the real-time operational data is collected by the controller after the tractor is started. Once the mode of automatic gearshift is selected, the controller selects gearshift schedule based on identification data and performs automatic gear shifting based on gearshift conditions and tractor running status. The tractor will remain in automatic gear-shifting mode until the exit order is issued, including pressing the exit button,

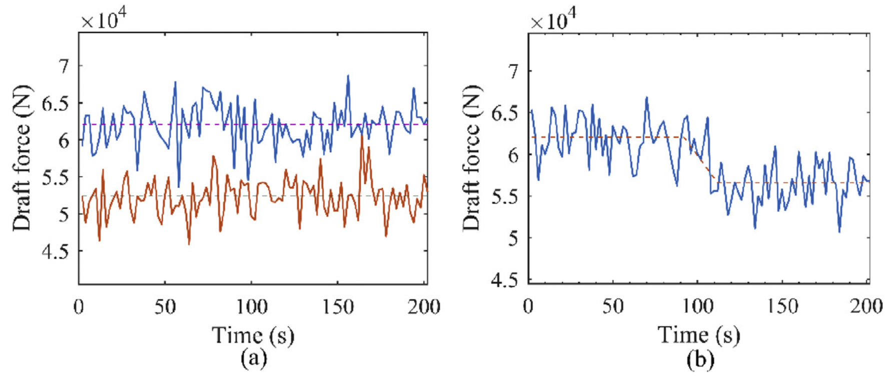

Fig. 9 – Draught forces of disk harrow under certain working conditions.

depression of the brake, or depression of the clutch. In addition, to avoid unnecessary shifting, downshift delay are used in the gear-shifting control.

To verify the validity and determine the advantages of an automatic gear-shifting strategy, it is necessary to design driving conditions. Although some typical driving cycles have been established for road vehicles (Hu, Zeng, Xu, Fu, & Qin, 2015), they are not suitable for tractors and farmland work. Some organisations, such as the American Society of Agricultural and Biological Engineers (ASABE) and the German Agricultural Society, have proposed performance test standards for tractors (Roeber, Pitla, & Luck, 2015). Driving conditions include multiple speeds and multiple loads and the cycle was designed to test the fuel efficiency of tractors with continuously variable and standard geared transmissions (Christopher, Michael, Roger, & Erin, 2013). Three driving cycles with a set point travel speed and 17 different drawbar loads were designed to test the fuel efficiency of tractor with a continuously variable transmission (Coffman, Kocher, Adamchuk, Hoy, & Blankenship, 2010). However, these test conditions were mainly focused on the performance tests of the tractor itself, rather than the control strategy. Therefore, a test condition is proposed in this paper to verify the control strategy of automatic gear shifting. It is not exactly the same as real working conditions. The cycle is mainly used to verify the control strategy performance of automatic gear shifting under the influence of different throttle openings, load, etc.

A disc harrow (*Type two* farm tool) with a machine width of 4 m working in farmland with fine soil under secondary tillage is taken as an example for analysis. As Fig. 11 shows, during

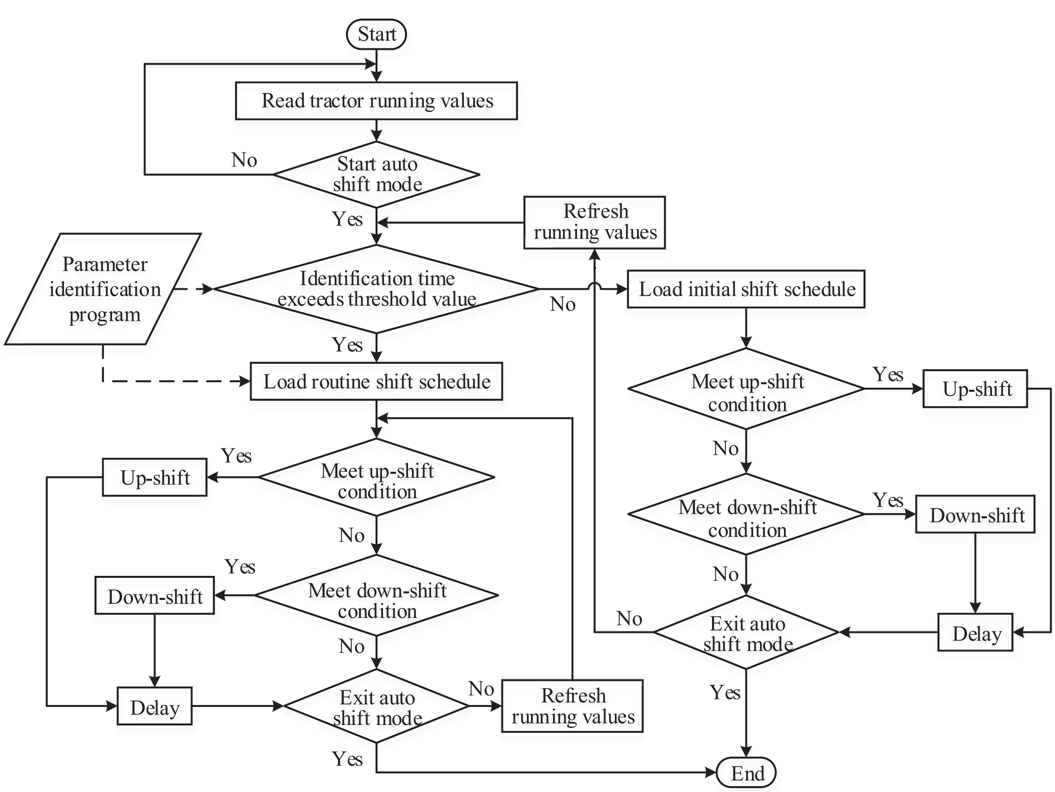

Fig. 10 – Control flow chart for automatic gear shifting.

the tractor operation, the change in transmission gear will be observed by changing the throttle opening and load (by changing the slope and depth of the farmland), thereby verifying the effectiveness of automatic gear-shifting strategy.

In order to further observe the working region of the engine, as well as verify the influence on fuel economy, tillage with a disc harrow on fine soil in an automatic gear-shifting mode was analysed by randomly changing the throttle opening and resistance. In the existing control strategy, to avoid gear shift cycle caused by the unknown resistance change characteristic, a large safety threshold has to be set. To evaluate the fuel-efficient performance of proposed method in this study, the automatic gear-shifting strategy for agricultural tractors which is common used in practice was used as comparison object with the results are shown in Section 5.

5. Results and discussion

The values of condition factor were identified in real time under different fluctuation amplitudes of draught force, and the identification results are shown in Fig. 12. Figure 12(a) shows the identification result under relatively big fluctuations of draught force (e.g. with mouldboard plough) and Fig. 12(b) shows the result with relatively small fluctuations (e.g. using a disk harrow) when tilling fine soils. It can be seen that the values of condition factor can be identified and they stabilised near the theoretical values within 75 s. As the identification time increases, the identification values of condition factor get closer to the theoretical values both with relatively large and small fluctuations of draught force. This meets the use requirement for automatic gear shifting. In addition, the identification results with small fluctuations are better than those with large fluctuations. Therefore, the proposed method for real-time identification of draught force variation characteristic by using the recursive least squares algorithm is effective and can be applied to automatic gear shifting tractors. It should be noted that although there are differences in the amplitude and frequency of draught force fluctuations using the different farm tools and using different soil properties and operating conditions, but the identification method is universal.

Figure 13 shows the variation in tractor parameters during field work with disk harrow under designed condition. The change histories of throttle opening and farmland slope are shown in the first graph of Fig. 13. In this test condition, the tractor's travel parameters, including gear, BFSC, travel speed, engine speed, engine torque, condition factor, and slip, change under the mode of automatic gear shifting, as shown in the second to fifth graphs of Fig. 13.

After the tractor starts in the 5th gear, it gradually increases to 9th gear and then remains in automatic gear-shifting mode and full throttle opening. At this stage, BFSC, travel speed, engine speed and engine torque are all different from their initial values and are in a relatively stable state. The value of the condition factor tended to be stable within 80 s.

From 120 s to 320 s, the throttle opening was reduced to 30% step by step. Correspondingly, the transmission gear changed to the 11th gear automatically. The BFSC and engine speed significantly decreased at this stage. The engine torque increased due to the speed ratio of transmission reducing. From 380 s to 580 s, the throttle opening was raised to 100% from 30% step by step, and the transmission gear automatically decreased to the 9th gear again. The BFSC and engine speed increased again, and engine torque also decreased again. In the whole process of throttle change, the travel speed and slip were relatively stable. It can be seen that the gear increased automatically and BFSC reduced when the throttle opening was reduced. This shows that the designed gear-shifting schedule is in line with the well-known fuel saving strategy of ‘gear up and throttle-down (GUTD)’ (Gotoh et al., 2010; Grisso et al., 2009; Schrock & Liu, 1990; Stephens, Spencer, Floyd, & Brixius, 1981).

From 650 s to 1150 s, with the change in field slope, the gear was automatically shifted. The tractor automatically downshifts to adapt to the increased resistance caused by the increased field slope but upshifts when the slope decreases. At 1300 s, when the tillage depth was reduced, the value of the condition factor changed, and the tractor upshifted in time to increase the engine load rate and reduce fuel consumption. In this stage, the engine maintained a relatively stable speed.

In the whole simulation process, engine torque was maintained above 550 N m. No matter the working condition,

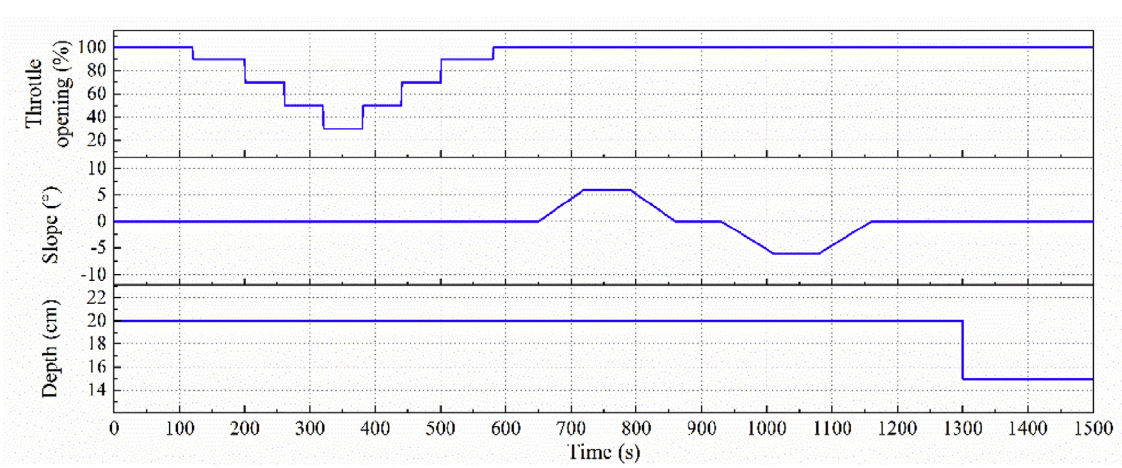

Fig. 11 – Test condition for tractor with disk harrow.

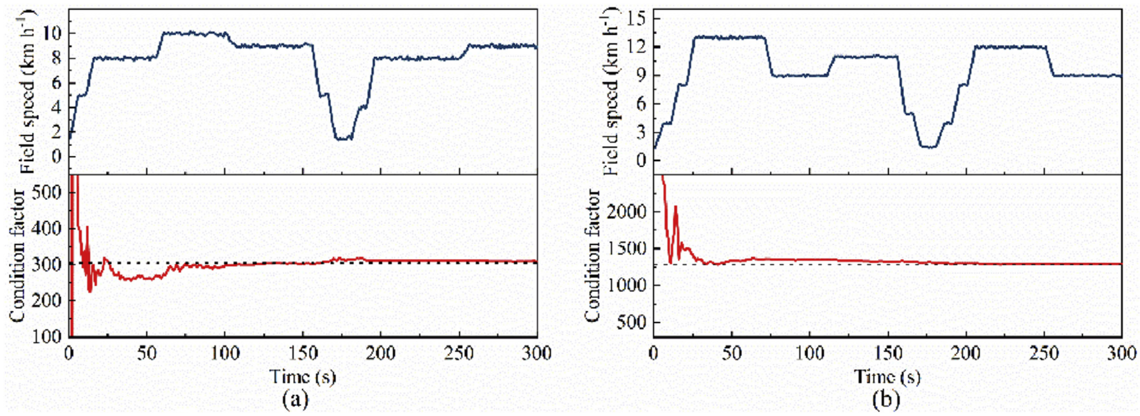

Fig. 12 – The parameter identification results: (a) result with large fluctuations of draught force; (b) result with small fluctuations of draught force.

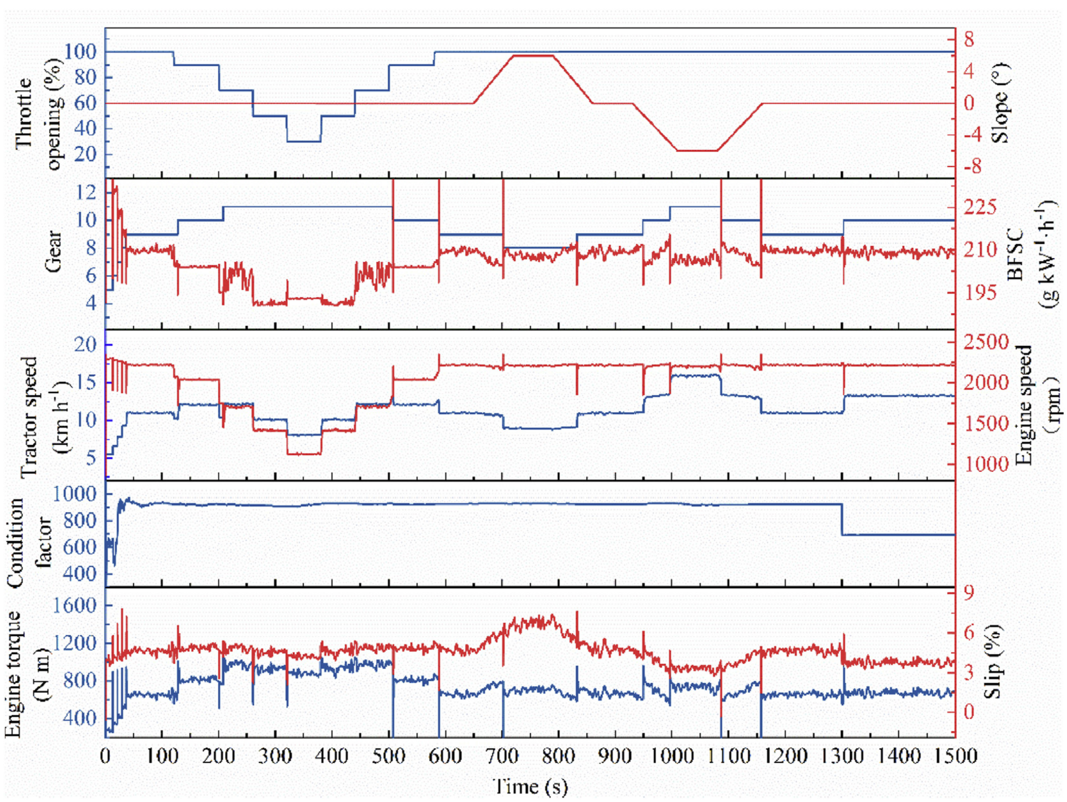

Fig. 13 – Parameter variation during tractor–harrow combination operation.

the engine can always operate at a high-load rate by automatically adjusting the transmission gear, thereby improving the fuel economy. This shows that the gear-shifting schedule has the desired effect.

At about 1000 s, the tractor speed exceeded 15 km h^{-1} when the field slope was negative, this may exceed the desired work speed of the driver. Therefore, setting a speed limit according to driver's intention for field work under automatic gear-shifting mode is a viable option. Figure 14 shows the work process of a tractor under automatic gear-shifting mode with work speed limit of 15 km h^{-1}. The working condition is consistent with that shown in Fig. 13. In the whole process, the work speed was always below 15 km h^{-1}. At about 1000 s, the gear did not rise and the work speed did not increase, which is different from the process shown in Fig. 13. Although this caused a slightly higher fuel consumption at the time, it was considered more important to follow the driver's intentions and ensure work quality.

In order to further verify the fuel economy performance, tillage using automatic gear-shifting mode was analysed by randomly changing the throttle opening and resistance. The engine operating point data (combination of engine speed and

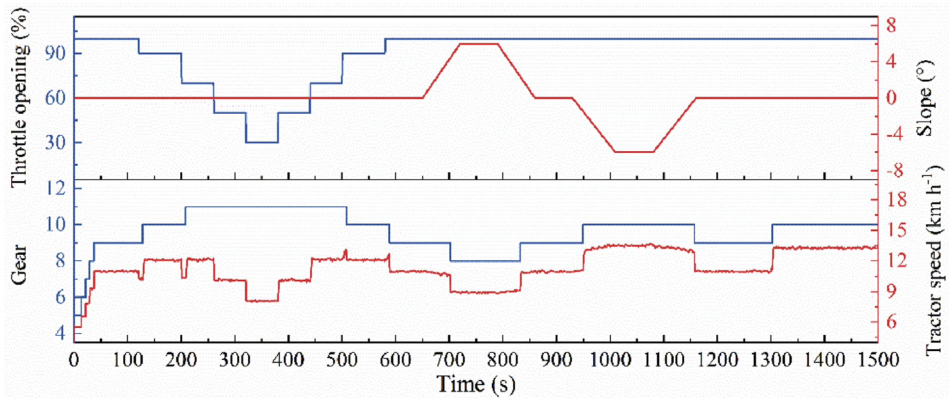

Fig. 14 – Parameter variation during tractor–harrow combination operating with speed limit.

engine torque) was randomly collected and displayed on the engine universal characteristics map, as shown in Fig. 15 (yellow points). It can be seen that the engine operating points were distributed in the region with high load rate, low fuel consumption and sufficient backup torque. Therefore, the gear-shifting schedule designed in this paper can maintain high fuel economy on the premise of ensuring power performance.

In order to compare the comprehensive fuel consumption of the proposed control strategy with that of the existing method, a tractor with a disk harrow and a machine width of 4 m was taken as an example for analysis. Tractor usually operated at a higher throttle opening during tillage to increase work efficiency and save time. Therefore, the throttle opening was set within the range of 50–100% in the simulation condition. Three types of soil texture (fine, medium, and coarse as defined by the ASABE (ASABE, 2011)) and various tillage depths were used for the analysis, as shown in Table 2.

In simulation, the initial soil texture, throttle opening, and tilling depth were set as fine, 50%, and 100 mm respectively. Whenever the tractor travelled 100 m, one of these three parameters was progressively increased once. The step sizes of the throttle opening and tilling depth were 1% and 1 mm, respectively. After the operating parameters reach the end of the boundary (soil texture, throttle opening, and tilling depth reach coarse, 100%, and 250 mm respectively), one of these three parameters was progressive reduced once every 100 m the tractor travels. The simulation proceeded until all combinations of these three parameters had been used twice. Then, the fuel consumption and time spent per hectare tilling was calculated based on total fuel consumption, total time spent, and total tilling area. The results showed that simulated fuel consumptions were 8.21 l ha^{-1} and 8.59 l ha^{-1} under the proposed method and exiting methods, respectively. Compared with the existing method, the control strategy

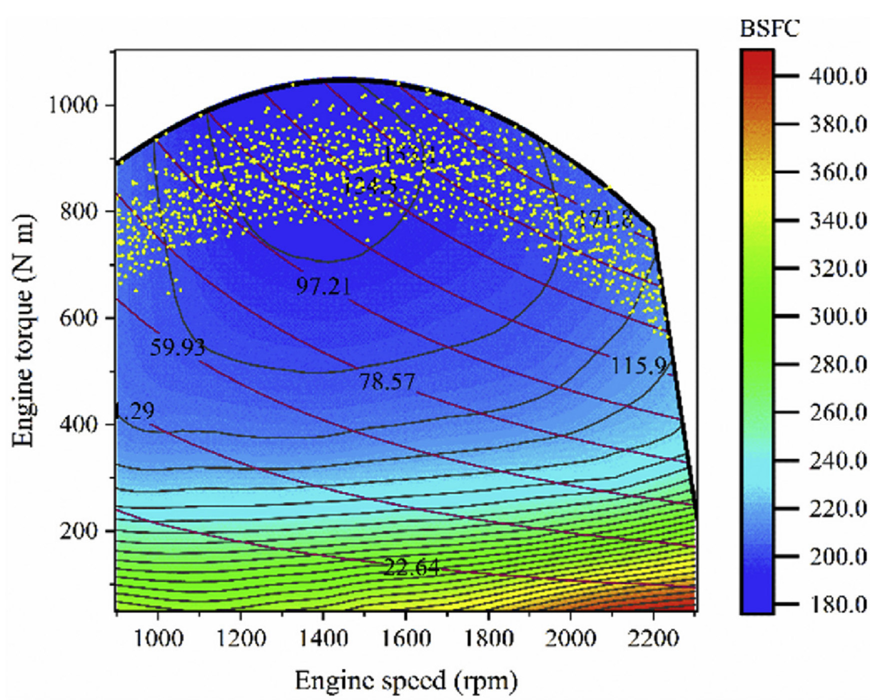

Fig. 15 – Engine operating points under automatic gear-shifting mode.

Table 2 – Simulation conditions.

Soil texture	Throttle opening (%)	Tilling depth (mm)
Fine	50	100
	51	101
Medium	52	102
	…	…
Coarse	99	249
	100	250

proposed in this paper could save 4.41% of diesel fuel during harrowing. In addition, the control strategy in this paper was predicted to save 50.3 s ha^{-1} of working time (working time is 778.4 s ha^{-1} and 828.7 s ha^{-1} under the proposed method and existing methods respectively, which improves work efficiency. This is because the fuel-saving control strategy tends to use higher gears (Gotoh et al., 2010).

6. Conclusion

To achieve the purpose of saving energy during tillage, a fuel-saving shifting strategy for agricultural tractors with power-shift transmission was designed by analysing the forces acting on the tractor–implement combination and the characteristics of the tractor powertrain. To realise automatic gear shift of tractor with this gear-shifting strategy in practice, a method using the recursive least squares algorithm was proposed for real-time identification of the changing characteristic of draught force. The validity and accuracy of real-time identification method and the fuel-saving performance of developed gear-shifting schedules were verified by designed operating conditions and using simulation analysis.

Analysis results show that proposed real-time identification method using the recursive least squares algorithm can identify the draught force changing characteristic with relative accuracy in real time, which is sufficient for automatic gear shifting of tractor–implement combination in field work. The designed four-parameter gear-shifting strategy can realise automatic gear shifting with the desired effect under various operating conditions and drivers' intentions. Accordingly, the engine always operates in the region with high load rate, low fuel consumption, and sufficient backup torque, which indicates that the gear-shifting schedule provides good fuel economy and power performance for the tractor during farm work. The control strategy proposed in this paper improves the fuel economy and work efficiency (saved working time) of tractor–implement combination during tillage.

This paper provides a theoretical basis for realising tractor intelligence and fuel economy. Future studies of this work include development of transmission control unit and carrying out field experiments to verify the performance of developed gear-shifting control strategies.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgments

The authors thank the State Key Laboratory of Mechanical Transmission of Chongqing University for the support and contribution. The work presented in this paper supported by financial support from the National Key R&D Program of China [No. 2016YFD0701100] and National Natural Science Foundation of China [No. 51875055].

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