Control System Design of Power Assisted Bike Based on Planetary Gear

Chun-Lin, Chen, Chun-Chin Wang, Mi-Ching Tsai*, and Po-Jen Ko
Department of Mechanical Engineering
National Cheng Kung University
Taiwan
mctsai@mail.ncku.edu.tw

***Abstract*—A planetary gear based transmission system for the power assisted bike (PAB) is proposed in this study because of its appealing 2-input/ 1-output transmission feature. Considering the practical requirements for power assistance, the electronically controlled continuously variable transmission (E-CVT) and the torque assistance modes were realized for the PAB system. Unlike the conventional approaches which only provides either torque assistance or automated gear-shifting function, the planetary gear based transmission enables the PAB to provide a suitable power assistance mode in accordance with different riding conditions. The feasibility and effectiveness of this study is verified by experiment.**

Keywords—power-assisted bike, planetary gear, E-CVT, torque assistance, switching

I. Introduction

Considering the need for reducing transportation costs and improving city mobility, the power-assisted bikes (PABs) have gained more and more popularity in the recent past [1][2]. A PAB, regarded as a conventional bicycle with power assistance, will be only activated while pedaling. The rider and the power assistance system in PAB share the driving effort such that the cyclist's fatigue and physical effort can be reduced compared to a standard bike [3].

In this study, a planetary gear based transmission system is adopted for PABs due to its appealing features of 2-input and 1-output transmission. The power assistance, which is realized by an integrated electric motor, is generated based on the predesigned assistance regulation strategies. Generally, the pedaling conditions can be divided into the low-speed set-off stage and the high-speed shifting stage. Considering the riding comfort and requirements, the power-assisted system should also provide different transmission assistance according to the riding environment and pedaling conditions [4][5]. Watterson [6] proposed the transmission system incorporated with two power assistance units to provide the automated gear shifting and torque assistance at the same time. However, it is not suitable for practical realization. The two power assistance functions should be implemented separately depending on the riding situations instead of being implemented simultaneously *in* the entire riding.

Accordingly, this study proposes the control approach based on a planetary gear transmission system, which can provide different power assistance modes for PABs. In the high-speed cruise, a speed regulation is determined by a predesigned electronically controlled continuously variable transmission (E-CVT) such that a preferred pedaling cadence can be achieved. In the low-speed set-off stage, an assisted torque is generated based on a pre-designed model reference impedance control (MRIC)fi. Therefore, the required power assistance modes can be provided for different riding conditions by one transmission system such that it can achieve the desired riding comfort and performance requirements. The feasibility and effectiveness are evaluated by experimental results.

II. System Descriptioin

A. Modeling of Bike System

The dynamics of a standard bike system is considered as shown in Fig. 1. From control point of view, the power assistance control system of PAB should notice the road load and the transmission mechanism. When the bike is balanced and moves straight, the dynamic equation is expressed in the following equations

$$\begin{aligned}&\text{longitunal: } F_R - F_F - F_d - (M+m)g\sin\theta = (M+m)a\\&\text{rear wheel: } T/N - B_{\text{Wheel}}\omega - F_R R = I\alpha\\&\text{front wheel: } F_F R_{\text{Wheel}} - B_{\text{Wheel}}\omega_{\text{Wheel}} = I\alpha\end{aligned} \quad . \tag{1}$$

The parameters of (1) are defined in Table I. Note that the subscript i=F, R denotes the front and rear wheels, respectively. Combining the equations of forces acting on the bike, (1) can be rewritten as

$$\begin{aligned}T &= (2I + MR^2)\alpha + 2B_{\text{w}}\omega + F_d R + R(M+m)g\sin\theta\\&= J_{\text{Bike}}\alpha + B_{\text{Bike}}\omega + T_{Load}\end{aligned} \quad . \tag{2}$$

According to (2), a block diagram description of the bike system is given in Fig. 2. The different power assistance modes

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can then be designed for different riding and pedaling conditions.

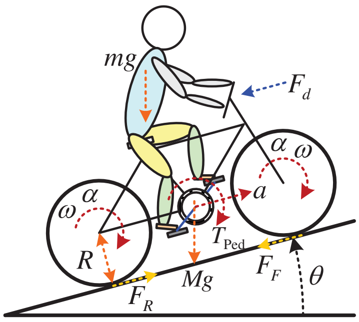


Fig. 1. Free-body diagram of bike system

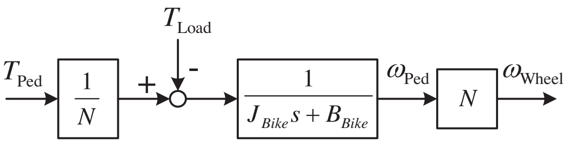


Fig. 2. Equivalent block diagram of bike system

TABLE I. PARAMETERS IN BIKE SYSTEM

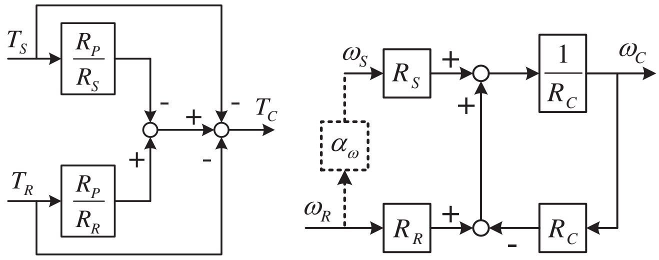
Symbol	Quantity
M, m	mass of bike and cyclist, respectively
g	gravitational acceleration magnetic induction
R_{Wheel}	radius of wheel
B_w	damping coefficient of wheel
θ	slope angle
N	speed ratio of transmission system
F_i, F_d	frictional force, wind resistance
τ_i	induced torque by frictional force
a, α	longitudinal and angular acceleration
ω_{Wheel}	angular velocity of wheel
J_{bike}, B_{bike}	equivalent moment of inertia and damping coefficient
T_{Load}	load torque include of wind resistance, friction, and slope force

B. Transmission of Planetary Gear

Generally, a planetary gear is composed of a carrier, a sun gear, a ring gear, and planet gears. 6 different input/output combinations can be selected depending on its applications. The transmission gearbox in PABs should have the increasing speed ratio such that the pedaling cadence can result in an increased wheel speed. Accordingly, the ring gear and the carrier are determined respectively as one of the input ports (for pedaling) and the output port in this study. The sun gear, which is utilized as the other input port, is connected to the driving motor. The torque and speed kinematic relationships of the transmission configuration are given as

$$\omega_C = \frac{R_R}{2R_C}\omega_R + \frac{R_S}{2R_C}\omega_S \ \ \& \ \ T_C = -\frac{R_C}{R_R}T_R - \frac{R_C}{R_S}T_S\,, \tag{3}$$

where R_i, ω_i and T_i are the radius, speed, and torque of each gear component, respectively. The subscript i=*C, S, R, P* denotes separately the carrier, sun gear, ring gear, and planet gears. In Fig. 3, both torque and velocity of the pedaling and those of driving motor can be integrated at the output port. Based on the kinematic analysis, different power assistance modes can be implemented by the second input (driving motor).



(a) Torque regulation (b) Speed regulation

Fig. 3. Block diagrams of torque and speed regulation

III. POWER ASSISTANCE SYSTEM

Fig. 4 shows the proposed control architecture for the power assistance system of PAB. A speed control system of integrated electric motor is utilized for both the E-CVT and the torque assistance such that the reference pedaling cadence ω^*_{pedal} will be determined based on the required power assistance modes.

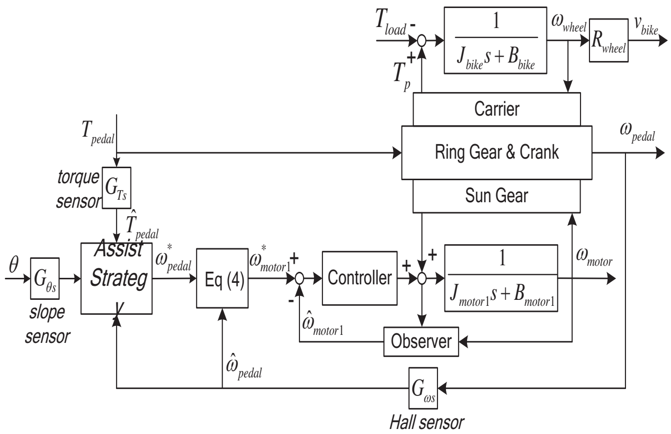


Fig. 4. Control structure of power assistance system

A. Speed Regulation Based on E-CVT

In order to achieve the desired pedaling cadence ω^*_{pedal}, the motor speed reference ω^*_{motor} is determined by ω^*_{pedal} and the measured angular velocity of wheel $\hat{\omega}_{wheel}$ such that

$$R_S\omega^*_{motor} = 2R_C\hat{\omega}_{wheel} - R_R\omega^*_{pedal}\,. \tag{4}$$

Since the human often has a narrow preferred pedaling cadence, it is necessary to properly change the gear ratio in high-speed shifting stage [6] such that it can accommodate undulating terrains or city traffic. Accordingly, the planetary gear based E-CVT is realized for the automated speed regulation. In Fig. 3(b), The transmission regulation index α_ω, which is formulated by an S-shape function in this study, is given as

$$\alpha_\omega(\omega_R, \omega_L, \omega_H) = \begin{cases} 1-2\left(\dfrac{\omega_R-\omega_H}{\omega_H-\omega_L}\right)^2 & \omega_L < \omega_R \le \dfrac{(\omega_L+\omega_H)}{2} \\ 2\left(\dfrac{\omega_R-\omega_H}{\omega_H-\omega_L}\right)^2 & \dfrac{(\omega_L+\omega_H)}{2} < \omega_R \le \omega_H \end{cases} . \quad (4)$$

where ω_L and ω_H are the pre-defined indexes for the transmission regulation.

According to (4), the motor regulates the sun gear with $\omega_S = \omega_{pedal}$ when the pedaling cadence ω_{pedal} is lower than ω_L. The E-CVT system is operated at the maximum assistance ratio $\alpha_\omega = 1$. Then, α_ω gradually decreases with the determined S-shape function while ω_{pedal} increases in the range $\omega_L < \omega_C \le \omega_H$. The assistance will be suspended while ω_{pedal} is greater than ω_H to avoid over assistance. Since the S-shaped E-CVT regulation is continuously differentiable, the speed ratio can be smoothly regulated along the entire transmission ratio range as shown in Fig. 5.

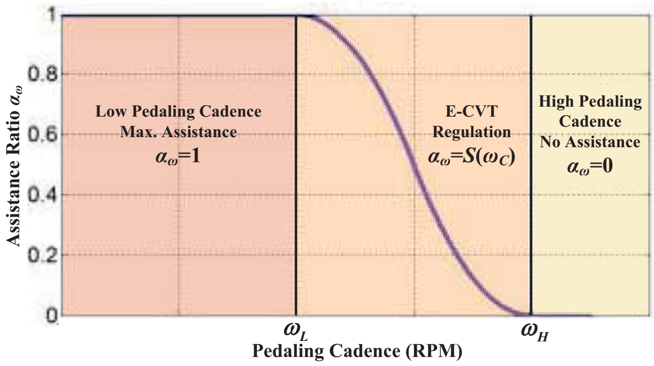


Fig. 5. Speed regulation of PAB

B. Torque Assistance Based on MRIC

In the low-speed set-off stage, a cyclist needs to overcome static friction and increase the bike speed quickly. This situation requires additional torque assistance instead of the automated gear-shifting. In order to reduce the pedaling torque during set-off riding with stable and upright cycling movement, the MRIC is employed in this study. By this approach, the power assistance system can adopt a lighter pedaling impedance such that it can endow the rider with an easier riding experience. In the following design process, the pedaling impedance Z_p is defined as [7]

$$Z_p = \frac{\omega_{Pedal}}{T_{Pedal}} = \frac{1}{J_{Pedal}s + B_{Pedal}} . \quad (5)$$

where J_{Pedal} and B_{Pedal} are the moment of inertia and damping coefficient. The purpose of torque assistance mode is to provide a predesigned smaller pedaling impedance (J_p^*, B_p^*) such that a cyclist can obtain a higher cadence with less pedaling torque, which can make the set-off riding easier.

IV. Experimental Results

The feasibility of the proposed control approach is evaluated by a spinning bike shown in Fig. 6, where the pedaling operation is connected to a planetary gear set and belt. In the experimental platform, two electric motors are used for the realization of power assistance and road loading, respectively. Moreover, the torque sensor and the Hall sensor are employed for the measurement of pedaling torque and cadence.

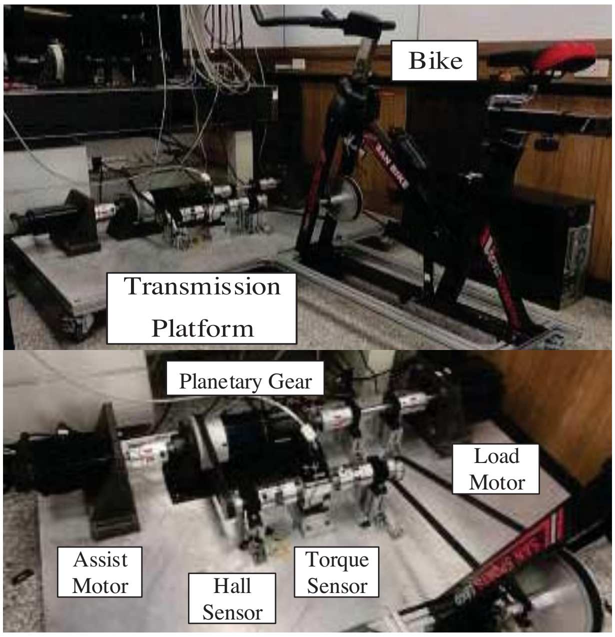
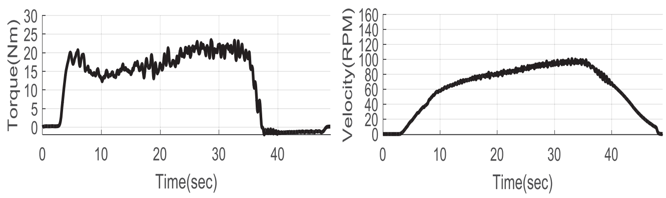


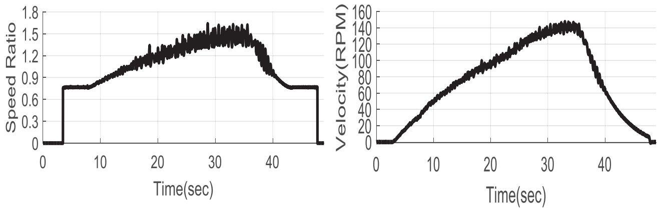
Fig. 6 Experimental platform for PAB

A. Speed Regulation

By employing the speed mode in the high-speed shifting stage, Fig. 7 shows that the wheel velocity as well as the speed ratio increased along with the pedaling velocity according to the predesigned E-CVT curve.



(a) Pedaling torque (b) Pedaling velocity

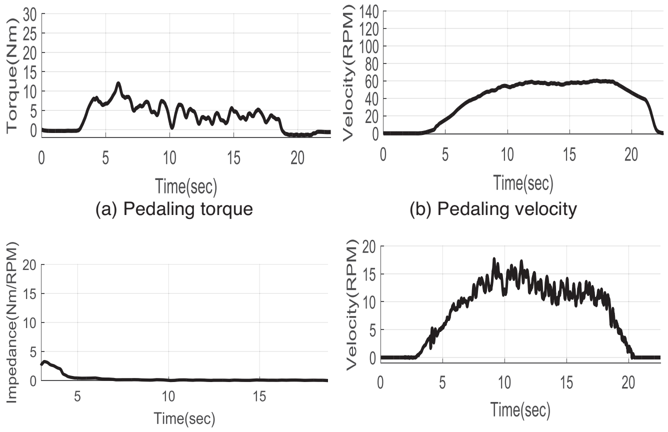


(a) Speed ratio (d) Wheel velocity

Fig. 7. Dynamic performance of the speed mode in the high-speed shifting stage

B. *Torque Assistance*

Fig. 8 shows that only half of the pedaling torque is needed to maintain the pedaling velocity of 60rpm when using the torque mode assistance. This amounts to a much smaller pedaling impedance in the beginning. However, the wheel velocity increases slowly, which means that the torque mode is suitable in the low-speed set-off stage but not the high-speed shifting stage.



(a) Pedaling torque (b) Pedaling velocity

(c) Pedaling impedance (d) Wheel velocity

Fig. 8. Dynamic performance of the speed mode in the high-speed shifting stage

V. CONCLUSION

A new power assistance approach is proposed in this study to achieve different required performances of a PAB, and is capable of dynamically switching between the torque assistance and the E-CVT. An assistant torque based on a pre-designed MRIC that reduces required pedaling torque at low-speed set-off stage, as well as the automated speed ratio regulation via E-CVT at high-speed cruise, is achieved through the same planetary gear based transmission system. The feasibility and effectiveness of the proposed approach has been verified in experimental results.

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