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Turning efficiency prediction for skid steering via single wheel testing ☆

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Abstract

Wheel design can be enhanced through experimentation, testing, and iteration. Unfortunately, the time and money needed to test full vehicles is costly. A cheaper, less conflated alternative could be to incorporate single wheel testing. The algorithm discussed in this paper uses single wheel testing to predict the full vehicle performance in a skid steer turn. With this prediction algorithm, skid steering can be easily enhanced by iterating on the design of a single wheel without the cost of vehicle testing. To validate this algorithm and explore skid steering enhancement several single wheel skid steering experiments were done and the results were compared to a full vehicle's turning performance.

Keywords: Skid Steer; Robotics; Testing; Wheel to soil; Terrain; Single wheel testing

1. Introduction

Compared to other methods, skid steering has several advantages such as simplicity in design and control which leads to fewer components, less weight, and a more robust vehicle design [12]. Because of these advantages skid steering has found many applications in construction, military, and agriculture. Skid steering is not new to the planetary rover field either. The Russian Cold War era rovers Lunokhod I and II used skid steering and were extremely successful in their missions [9].

Skid steering does pose some deterrents as a steering methodology. Predicting the behavior of the vehicle in a skid steer turn is difficult due to the unmeasurable sliding that occurs. Much work is being done currently on analytical solutions for skid steering modeling such as [10,1]. Another deterrent is the low efficiency that results from energy being used to skid the wheels in a turn. Empirical methods can mitigate these disadvantages but full vehicle tests can be expensive and time consuming.

To avoid the costs of full vehicle field tests, we suggest using single wheel testing as part of the iterative design process for improving skid steering. For single wheel tests to be of any use, the data that they generate must have some significance in the real world. The performance in the single wheel testing machine must transfer to predict the behavior exhibited on a multi-wheel vehicle doing typical maneuvers in field conditions. What this paper discusses is the algorithm that allows a transfer to be possible by comparing the predicted vehicle rotation per wheel rotation to that of a vehicle incorporating the same wheels.

2. Wheel forces in a skid turn

This paper will focus on a zero radius skid steer turn on a flat hard surface. When a vehicle initiates a turn its rotation (in the X–Y or ground plane) will accelerate up to a certain spin rate Ω at which point it will stabilize and the

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moment about its center (M_o) will equal zero simplifying it to a statics problem. Fig. 1 shows a top view force body diagram of a vehicle in an equilibrium skid steer turn. The assumptions for the system are:

- Other than ground interaction and gravity there are no external forces or accelerations acting on the vehicle.
- Vehicle weight is evenly distributed.
- All four wheels rotate at the same rate throughout the test.
- Normal forces are the only loading from suspension to the wheels.

F_y is the force along the wheel's spinning direction due to its traction where f_y is the opposite force due to the rolling or bulldozing resistance of the wheel. F_x is along the wheel's axis and symbolizes any lateral traction induced by the wheel's rotation by its tread, where f_x is conversely opposite of F_x and is due to the lateral sliding friction of the wheel. F_X and F_Y symbolize the resultant component forces on the wheel.

$$\Sigma M_o = 0. \tag{1}$$

$$M_{motor} - M_{friction} = 0 \tag{2}$$

$$F_Y = F_y - f_y \tag{3}$$

$$F_X = f_x - F_x \tag{4}$$

$$\Sigma(F_Y R\cos(\Theta)) - \Sigma(F_X R\sin(\Theta)) = 0. \tag{5}$$

$$F_Y = F_X\tan(\Theta). \tag{6}$$

Eq. (6) describes a relationship between F_X and F_Y at the turning equilibrium point and is dependent upon the vehicle geometry (Θ). If the vehicle were slender (Fig. 2a) then Θ would be larger than $\frac{\pi}{4}$ and F_X would be much smaller than F_Y. If $\Theta = \frac{\pi}{2}$ then $F_Y = \infty$. This would mean that no matter how much force a non-directional wheel could exert on the ground the vehicle's spin rate Ω would always be zero. If, on the other hand, Θ were equal to zero, as in Fig. 2b, then $F_Y = F_y + f_y$ (the net force of power and resistance) would be equal to zero. This configuration is better known as differential steering, or in the case when the wheel orientation can be adjusted to remain tangent to the turn radius is Ackerman steering, originally developed by Erasmus Darwin [8]. This means that the wheels have no lateral slip and assuming there is no longitudinal slip then the turning rate, Ω, can be calculated by the following:

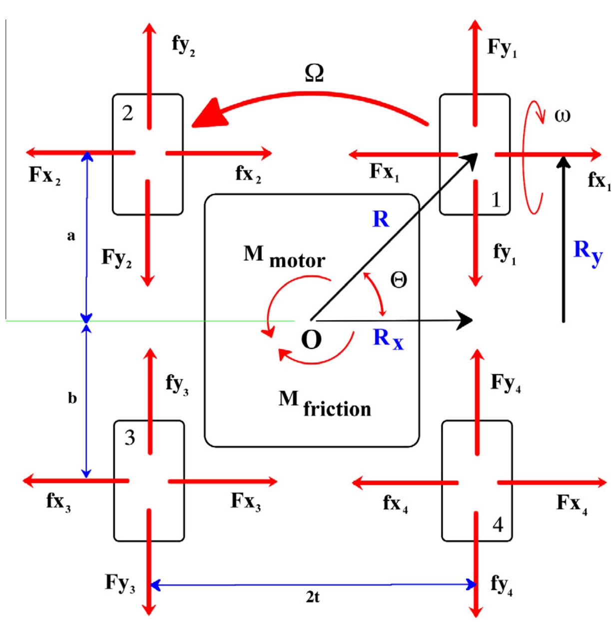

Fig. 1. Skid steer force body diagram.

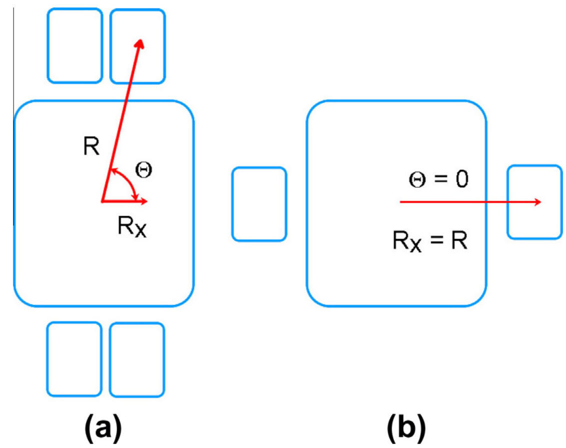

Fig. 2. Skid steer geometry configurations (a) slender (b) wide.

$$V_{@wheel} = \Omega R \tag{7}$$

As well as:

$$V_{@wheel} = \omega r \tag{8}$$

$$\Omega_{Ackerman} = \frac{\omega r}{R}, \quad F_Y = 0. \tag{9}$$

where ω is the wheel angular velocity in radians per second, r is the wheel radius, and R is the distance from the center of the wheel to the center of rotation of the vehicle (Fig. 2b).

Eq. (9) refers to the Ackerman turning rate $\Omega_{Ackerman}$ without longitudinal slipping. To calculate Ω for a skid steer vehicle ($\Theta \neq 0$), Θ must be taken into account and is reflected in Eq. (10). $\Omega_{TheoreticalMax}$ refers to the theoretical maximum a skid steer vehicle can spin, but F_Y, at $\Omega_{TheoreticalMax}$, is not zero (Fig. 2a).

$$\Omega_{TheoreticalMax} = \frac{\omega r}{R}\cos(\Theta), \quad F_Y \neq 0. \tag{10}$$

To find the value of $\Omega_{F_Y=0}$, which is the spin rate at which there is no longer a net force in the Y direction, the longitudinal velocity (V_y) (Fig. 2b) of the ground under the wheel must be equal to the velocity of the wheel rim (ωr) therefore making $F_Y = 0$ (no slip). Eq. (15) explains this relationship (see Figs. 3,10).