

Pitch Control for a Semi-track Air-cushion Vehicle Based on Optimal Power Consumption

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ABSTRACT

A new kind of integrated semi-track air-cushion pitch controller is proposed in this paper. The controller first compute the target working point based on a weighed function, which is the combination of optimal power consumption and pitch angle control demand. Then the sequential quadratic programming algorithm distributes the general target values to specific control values. The performance of the controller is verified through co-simulation between Matlab/Simulink and ADAMS/View. The simulation results show the effectiveness of the control algorithm and the correctness of the choice in physical configuration with two air cushions for vehicle body pitch control.

KEYWORDS - Vehicle Dynamics, Pitch Control, Optimal Power Consumption, Semi-track Air-cushion Vehicle, Soft Terrain

INTRODUCTION

In order to improve the transportability and the traction performance of a vehicle on soft terrain and wet fields, such as marshlands, bog soil and paddy fields, etc., a new type of combined vehicle, semi-track air-cushion vehicle, was developed [1]. It combines short track propulsion system with air cushion system to maximize the vehicle performance on soft terrain. By introducing adjustable suspension-like spring linkage between the propulsion system and air cushion system, the relative height of vehicle body to ground can be adjusted and the load supported by each system can be measured easily

and seen visually. With the help of frequency converter the load distribution between the track and air force was realized properly [2][3]. But there will be situations when the vehicle undergoes the acceleration, enters into non uniform soft terrain or the different sinkage among wheels or tracks due to the load distribution difference alone the longitudinal axis of vehicle. If there is only one unique air force the vehicle body's attitude can not be adjusted properly when there is sinkage or terrain level difference between front wheel and rear tracks, hence preventing the vehicle being used widely in industry.

To solve the above problem, a new type of changeable air passage between upper plenum chamber and lower cushion chamber is introduced here. The main concern is focused on pitch control in this paper, one part of the attitude control. But according to early research [4][5], the power consumption of the semi-track air-cushion vehicle has to be minimized. So the power consumption is evaluated combined with the pitch control. And to enhance the pressure difference between front air cushion and rear, a soft curtain is used in the lower chamber approximately in the middle part. The demonstration of the air flow is shown in figure 1.

MAIN SECTION

1 VEHICLE MODELING

First, the vehicle configuration is introduced as shown in figure 2. Generally the vehicle is composed of two parts, one is the propulsion system part and the other is air cushion part. For the propulsion part there are front

guide wheel and rear semi-tracks. For the air cushion part there are fan, fan motor, plenum chamber, air cushion chamber and skirt. The link between propulsion part and the air cushion part is through springs, so the weight distribution change can be seen and measured from the length change of springs. To realize the plenum chamber pressure, fan speed is adjusted by a frequency converter, and to realize the front and rear air cushion chamber pressure, changeable air passage valves are designed and controlled by step motors. To realize two semi-tracks' slip ratio control, three phase brushless DC motors are used.

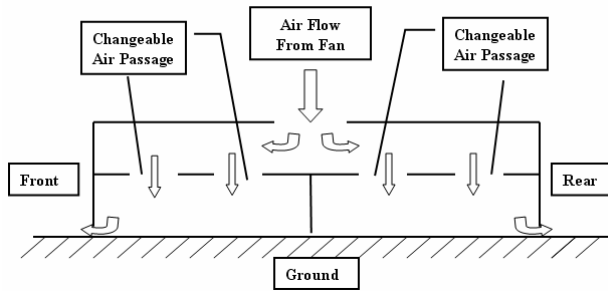
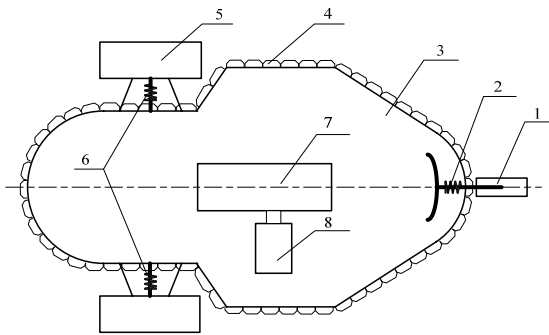


Figure 1 Air flows in vehicle body



1. Guide wheel; 2. Front spring; 3. Vehicle body; 4. Skirt;
5. Semi-track; 6. Rear springs; 7. Fan; 8. Fan motor

Figure 2 Structure sketch of vehicle

Next, some assumptions and simplifications are made to simplify the problem analysis as shown below:

- (1) The soft terrain surface is totally level before vehicle entering;
- (2) Track-terrain contact area is with uniform normal pressure distribution;
- (3) The skirt is made long enough to keep constant clearance height h_c ;
- (4) The soil bulldozing resistance of front wheel is ignored;
- (5) The front wheel is modeled as rigid wheel;
- (6) The body pitch angle and slope angle created by sinkage are small.

Then, for the convenience of modeling, the forces acting on the vehicle and some related symbols are shown in figure 3.

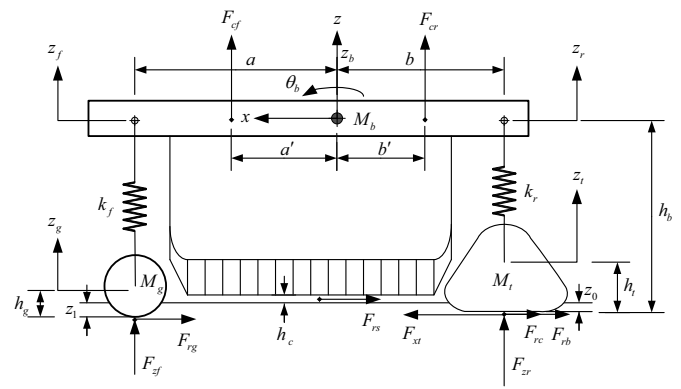


Figure 3 Forces acting on the semi-track air-cushion vehicle

1.1 Vertical Dynamics

1.1.1 Ground Contact Forces

Constructing the torque balance at rear contact point and front one in figure 3, the ground contact forces of front wheel and rear tracks F_{zf} F_{zr} can be obtained as follows

$$F_{zr} = \frac{1}{2L} [-M_g a_x h_g + M_b g a - M_b a_x h_b + 2M_t g L - 2M_t a_x h_i - F_{cf} (a - a') - F_{cr} (a + b')] \quad (1)$$

$$F_{zf} = \frac{1}{L} [M_g g L + M_g a_x h_g + M_b g b + M_b a_x h_b + 2M_t a_x h_i - F_{cf} (a' + b) - F_{cr} (b - b')] \quad (2)$$

where M_g is the mass of front wheel, M_b is the mass of vehicle body, M_t is the mass of one track, a_x is the longitudinal acceleration, h_g is the height of center of gravity of front wheel, h_b is the height of center of gravity of vehicle body, h_i is the height of center of gravity of track, a is the distance between the front axle and the center of gravity of vehicle body, b is the distance between the rear axle and the center of gravity of vehicle body, L is the wheel base, a' is the distance between the point of application of front air cushion force and the center of gravity of vehicle body, b' is the distance between the point of application of rear air cushion force and the center of gravity of vehicle body, F_{cf} is the front air cushion force, F_{cr} is the rear air cushion force, g is acceleration due to gravity.

1.1.2 Air Cushion Forces

There is an important parameter, the load distribution ratio δ , to describe the working situation of air cushion vehicle. It is defined as

$$\delta = \frac{W_c}{W} \quad (3)$$

where $W_c = F_{cf} + F_{cr}$ is the weight supported by air cushion force and W is the total weight of the vehicle. In the expression of W_c , F_{cf} and F_{cr} are the air Cushion forces, which can be expressed in detail by

$$\begin{cases} F_{cf} = p_{cf} S_{cf} \\ F_{cr} = p_{cr} S_{cr} \end{cases} \quad (4)$$

where p_{cf} is the front air cushion pressure, S_{cf} is effective front air cushion area, p_{cr} is the rear air cushion pressure, and S_{cr} is effective rear air cushion area.

1.1.3 Front Wheel and Rear Tracks Sinkage

According to Bekker's pressure sinkage relationship [6], the sinkage of front wheel and rear tracks can be calculated by

$$z_1 = \left[\frac{3F_{zf}}{(k_c + b_g k_\phi) \sqrt{D}(3-n)} \right]^{\frac{2}{2n+1}} \quad (5)$$

$$z_0 = \left[\frac{F_{zr}}{(k_c + b_l k_\phi) l_t} \right]^{\frac{1}{n}} \quad (6)$$

where z_1 is the front wheel sinkage, z_0 is one of the rear tracks sinkage, k_c is cohesive modulus of terrain deformation, k_ϕ is frictional modulus of terrain deformation, n is exponent of terrain deformation, b_g is the width of front wheel, b_l is the width of one rear track, D is the diameter of front wheel, l_t is the length of the track in contact with the terrain.

1.1.4 Suspension Vertical Motion Equations

The dynamical vertical motion of the vehicle body can be described by

$$\begin{cases} M_b \ddot{z}_b = F_{kf} + 2F_{kr} + F_{cf} + F_{cr} \\ I_y \ddot{\theta}_b = -aF_{kf} + 2bF_{kr} - a'F_{cf} + b'F_{cr} \end{cases} \quad (7)$$

where F_{kf} and F_{kr} are the dynamical spring forces, which can be expressed by

$$\begin{cases} F_{kf} = k_f (z_g - z_f) \\ F_{kr} = k_r (z_i - z_r) \end{cases} \quad (8)$$

where k_f is the equivalent spring stiffness of front suspension, k_r is the equivalent spring stiffness of rear suspension, z_g is the dynamical front wheel displacement, z_f is the dynamical front suspension displacement, z_i is the dynamical rear truck

displacement, and z_r is the dynamical rear suspension displacement. There are relationships between dynamical body displacement z_b and dynamical front and rear suspension displacements z_f , z_r , which are given by

$$\begin{cases} z_f = z_b - a\theta_b \\ z_r = z_b + b\theta_b \end{cases} \quad (9)$$

where θ_b is the vehicle body pitch angle, which is positive when the front vehicle body falls.

1.2 Longitudinal Dynamics

The equation of motion along the longitudinal axis of the vehicle is given by

$$Ma_x = 2F_{xt} - 2F_{rc} - 2F_{rb} - F_{rg} - F_{rs} \quad (10)$$

where M is the vehicle mass, a_x is the longitudinal acceleration of the vehicle, F_{xt} is the tractive effort that one track can get from terrain, F_{rc} is the soil compaction resistance of one track, F_{rb} is the soil bulldozing resistance of one track, F_{rg} is the motion resistance of the guide wheel, F_{rs} is the skirt contact drag.

In detail, F_{xt} is given by [7]

$$F_{xt} = (Ac + F_{zr} \tan \phi) \left\{ 1 - \frac{K}{|s_r| l_t} [1 - \exp(-\frac{|s_r| l_t}{K})] \right\} \text{sgn}(s_r) \quad (11)$$

where s_r is the track slip ratio, it can be given by

$$s_r = \begin{cases} \frac{r\omega - V}{V} & , \text{ braking} \\ \frac{r\omega - V}{r\omega} & , \text{ driving} \end{cases} \quad (12)$$

where r is the effective turning radius of rear track, ω is the turning speed of track drive motor, and V is the vehicle longitudinal speed.

F_{rc} is given by [7]

$$F_{rc} = \frac{1}{(n+1)(k_c + b_l k_\phi)^{1/n}} \left(\frac{F_{zr}}{l_t} \right)^{\frac{n+1}{n}} \quad (13)$$

In soft terrain or loose soils, the bulldozing resistance F_{rb} may be estimated using the following equation [7]

$$F_{rb} = b_t (0.67cz_0 K'_{pc} + 0.5z_0^2 \gamma_s K'_{py}) \quad (14)$$