GAIN-SCHEDULE CONTROL BASED ON MASS ESTIMATION APPLIED TO THE BRAKING SYSTEM OF AN URBAN AUTOMATED PEOPLE MOVER

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Abstract. *This paper presents a new control strategy to be applied to the braking system of an urban automated people mover called Aeromovel, a transport system whose vehicle moves on rails and is propelled by pressurized air. The proposed controller is based on a PID control law associated with the gain-schedule strategy. It employs a recursive estimate of the transported mass as a reference for the switching among different gain sets. The control algorithm is evaluated using both simulations and experimental results.*

Keywords*: Gain-Schedule, Braking Control, Adaptive Control.*

1. INTRODUCTION

Aeromovel is a non-conventional Automatic People Mover (APM) that runs on rails over suspended beams, as illustrated in Fig. (1). The system operation principle relies on pneumatic propulsion, inspired by sailing boats. This configuration allows the propeller system to be fixed to the ground, separated from the vehicle that is, therefore, lighter than the similar ones that are used in most of other technologies. The advantage of this arrangement is a significant weight reduction of the whole system, so that both civil structure and stations may have a relatively small size compared to similar transport systems. According to Medeiros (1985), the Aeromovel was originally designed in the end of decade of 1960, and had the first physical prototype developed in 1977. At the beginning of the decade of 1980, it was constructed the first experimental line in the city of Porto Alegre - Brazil, with a total length of 960 m and two stations separated by 655 m. Today, such line is employed in studies regarding the technical and economic viability of the Aeromovel technology as an alternative for developing efficient and environment-friendly urban transport systems.

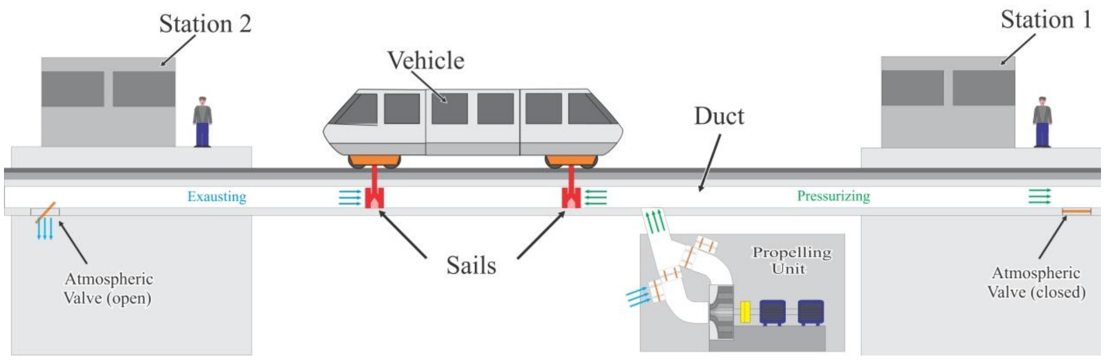


Figure 1. Schematic view of the Aeromovel system.

In modern people movers, to avoid the invasion of the guideway by non-authorized people, a common solution is the utilization of automatically controlled sliding doors between vehicle and stations, similar to those found in elevators. Due to the use of these doors, it is necessary to stop the vehicle within strict tolerance limits, because its doors must open only when aligned with those of the stations, allowing the safe circulation of the users.

The vehicle uses a friction brake system to make the vehicle to stop in a specific position on destination station. In the present configuration, the control algorithm designed for this task is a standard PID controller with fixed gains. The same algorithm is responsible for controlling the breaking procedure for all situations of vehicle loading. Nevertheless, experimental tests showed that such control scheme is not efficient when precision requirements are severe, especially when the number of passenger varies significantly.

In order to achieve acceptable levels of braking precision for the Aeromovel system, this work proposes a novel control strategy based on the values obtained continuously through an recursive estimator of the mass transported used as a reference for selecting the set of gains to be applied to a PID control law. Thus, the most appropriate set of gains can be used to match the variations in the dynamic behavior of the system as its transported mass changes with the number of transported passengers, so that station-stopping errors of the vehicle can be reduced.

This paper is organized as follows. Section 2 presents the characteristics of the original braking apparatus of the system. Section 3 is dedicated to presenting the mathematical model of the Aeromovel system with its friction brake equipment. In Section 4, the proposed control algorithm is described. Section 5 is devoted to the presentation of simulating results, illustrating the application of the control algorithm with a computational model of the system, whereas experimental results are discussed in Section 6. Finally, the main conclusions are outlined in Section 7.

2. Evaluation of the Braking Apparatus

The civil structure of the pilot line is comprised by two stations ("*Gasômetro*" and "*Fazenda*"), and the duct connecting them. In the middle of this duct there is a Propulsion Power Unit (PPU), which consists of a centrifugal fan coupled to an electric motor and a group of four flow-directional valves. Such valves allow blowing air from the atmosphere to the duct or from the duct to the atmosphere, according to the desired direction for the vehicle movement.

The operational procedure adopted by Aeromovel Brazil S. A. (ABSA) to transport passengers between stations is defined in 4 steps, in a way that one typical trip can be described as follows:

1 º. The vehicle is maintained still at one of the stations until all passengers are positioned inside the vehicle, then, the doors are automatically closed;
2 º. PPU starts the operation in *Pull* mode, exhausting air from the duct in order to pull the vehicle;
3 º. After the vehicle crosses the PPU position on the line, the system turns into *Push* mode, in which the PPU insufflates air into the duct, causing the vehicle to be pushed towards the destination station;
4 º. The automatic braking procedure is initiated 150 m before reaching the destination station. At this moment, the PPU is shut off and the onboard controller automatically brakes the vehicle. The controller aims to stop the vehicle in a specific position at the station by tracking a pre-specified dynamic pattern. When the vehicle stops, the doors are automatically opened.

As in the case of other APM systems, the final position of the vehicle at the stations must be confined between strict precision limits. On its original configuration, the braking task is carried out by employing a standard PID control law, applied to the system by means of a Programmable Logical Controller (PLC) installed onboard of the vehicle. Using this scheme, it was observed by ABSA that the desired precision limits are not respected, especially when there are significant variations in the mass of the vehicle. This fact was corroborated by further tests made at the *Fazenda* station. In order to observe the performance of the original braking scheme, the vehicle was tested with 4 different values of transported mass: 7735 kg, 9610 kg, 11485 kg and 13360 kg. These values correspond to 0, 25, 50 and 75 passengers on the vehicle, respectively. In all cases, it was observed that the vehicle exceeded the 150 m of desired displacement, stopping behind the specified final position. When the transported mass value was 7735 kg, the mean value of the observed stopping errors is 0,665 m, with a standard deviation of 0,111 m. The corresponding values (mean value and standard deviation) in the cases of the other loading conditions were 0,535 m and 0,116 m for 9610 kg; 0,651 m and 0,095 m for 11485 kg; 0,346 m and 0,180 m for 13360kg. A complete discussion of these experiments can be found in Sarmanho Jr. (2009).

Despite their relatively large values, the mean errors observed can be corrected by a calibration procedure, carried out by changing the starting point of the reference trajectory tracked by the controlled system. Thus, the main challenge of the control algorithm is to reduce the non-systematic errors inherent to the braking process, so that the standard deviation of the positioning error becomes bounded within its specified limit of 0,10 m (10 cm). This value is assumed by ABSA as being precise enough for most of the applications of the Aeromovel system. Through the experimental results above presented, it can be observed that the variation of the mass transported bears a significant influence on the value of the standard deviation of the vehicle stopping errors, especially if the mass is large. For this reason, the proposed algorithm must be able to change the control behavior in accordance with the mass that is transported in each trip. The project of the proposed controller is addressed in Section 4 of this work.

3. DYNAMIC MODEL OF THE VEHICLE

The objectives of the mathematical model presented in this work are twofold: **(i)** to allow the mass estimation algorithm to perform its task with satisfactory accuracy so that the switching of the gain sets employed by the PID controller can be performed correctly, **(ii)** to calculate the gain sets that are most suitable for each operation condition of the vehicle. For these purposes, it is initially defined a coordinate system as depicted in Fig. (2). In this figure, x is the direction of the translational movement and y is the displacement in the axis perpendicular to x in the plane of the figure. The variables $\dot{x}$ and $\ddot{x}$ represent, respectively, the instantaneous velocity and acceleration of the vehicle and g is the acceleration of gravity.

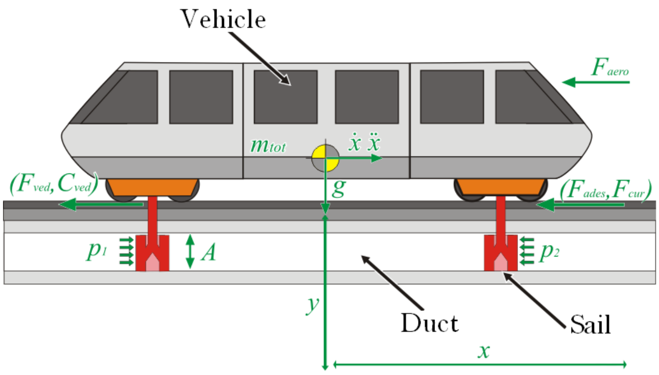


Figure 2. Dynamic equilibrium of the vehicle.

Based on Newton's Second Law, the dynamical equilibrium forces acting on the vehicle can be expressed as:

$$F_{tot} = m_{tot}\ddot{x} = A(p_1 - p_2) - F_{ved} - c_{ved}(\dot{x}) - F_{aero} - F_{ades} \tag{1}$$

where F_{tot} is the resultant force that moves the vehicle, m_{tot} is the vehicle mass, A is the cross section area of the sails, p_1 and p_2 are, respectively, the pressures applied on sails at rear and front parts of vehicle, F_{ved} is the static friction force between the seal rubber and the sail shaft, c_{ved} is the dynamic friction coefficient between the seal rubber and the sail shaft, F_{ades} is the adhesion force wheel/rail according to model proposed by Polach (2005) and F_{aero} is the aerodynamic drag force calculated according to equation presented in Gillespie (1992):

$$F_{aero} = \frac{c_D A_{eq} \rho}{2}(v_{ar} + \dot{x})^2 \tag{2}$$

where c_D is drag coefficient of the vehicle, A_{eq} is the cross-sectional area of the vehicle main body, ρ is the air density and v_{ar} is the wind speed.

The model of the interaction between the braking device and the interface between the wheel and the rail is developed by analyzing the specific characteristics of the test vehicle, whose wheels are completely independent. Thus, effects like cross slips and spins do not occur. The vehicle does not have a suspension system, therefore, the normal force on the wheels can be directly expressed by $m_{tot}g/n_w$, where n_w is the number of wheels of the vehicle. Figure (3) presents the forces acting on a single wheel.

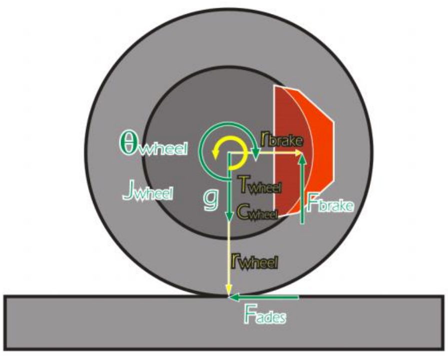


Figure 3. Force balance on a wheel.

The relationship between the forces illustrated in Fig. (3) can be expressed mathematically as:

$$J_{wheel}\ddot{\theta}_{wheel} = T_{brake} + (c_{wheel}\dot{\theta}_{wheel} + T_{wheel}) - F_{ades} r_{wheel} \tag{3}$$

where J_{wheel} is the inertia moment of the wheel, θ_{wheel} is the angular displacement, c_{wheel} is the angular friction viscous coefficient, T_{wheel} is the external torque applied to the wheel and r_{wheel} is the radius of the wheel. The braking torque T_{brake} can be expressed by Eq. (4):

$$T_{brake} = F_{brake} r_{brake} = (\mu_{discE} + \mu_{discV}\dot{\theta}_{wheel}) A_{brake} p_{brake} \tag{4}$$

where r_{brake} is the caliper average radius, μ_{discE} and μ_{discV} are, respectively, the Coulomb friction and the friction coefficients between the disk and the brake pad as a function of angular velocity, A_{brake} is the area of hydraulic piston of the caliper and p_{brake} is the input hydraulic pressure that is applied to the caliper.

4. CONTROL ALGORITHM

The proposed control scheme is composed by 3 parts: a recursive algorithm that is responsible for estimating the transported mass; a PID control law that controls the braking actions; a gain-schedule scheme, responsible for choosing the correct set of gains to be applied to the PID controller in each trip.

4.1 The recursive mass estimator

In transport systems, according to the number of passengers, there is a proportional change in the transported mass. In Aeromovel, these changes may cause the total weight to be duplicated from one trip to another. Since this variable has a significant influence on the performance of dynamic systems, especially regarding the velocity and acceleration of the vehicle, it is desirable that the information about its value is constantly known.

Obtaining the value of the transported mass by direct measure presents difficulties related to the costs of installation and maintenance of the instrumentation system. Besides, procedures for equipment calibration must be carried out frequently, which limits the utility of this solution in practical applications. One suitable alternative is the use of a recursive algorithm for estimating the mass of the vehicle, based on the use of more easily measured variables. In this context, it is possible to use an algorithm originally described by Slotine and Li (1991) to estimate the mass of the vehicle. This algorithm uses the acceleration and the resultant force to estimate the transported mass. It was chosen based on its convenience and simplicity for computational implementation. Defining $\hat{m}$ as the estimated value of the transported mass, the prediction error e_m between the estimated mass times instantaneous acceleration minus the resultant force is calculated through Eq. (5).

$$e_m = \hat{m}\ddot{x} - F_{tot} \tag{5}$$

where, F_{tot} is the total force applied to the vehicle (whose value is obtained by employing Eq. (1)). Additionally, defining the auxiliary term $P(t)$ given by:

$$P(t) = \left[\int_0^t \ddot{x}^2 dr\right]^{-1} \tag{6}$$

Based on the above definitions, the estimating mass law is:

$$\dot{\hat{m}} = -P(t)\ddot{x}e_m \tag{7}$$

By means of theoretical analysis, it is possible to prove that, if the system is persistently excited, the mass estimated by the employed algorithm converges asymptotically to the true value of the mass transported in the system. Experimentally, it is verified that the estimation errors are less than 6% of the nominally transported mass. Full description of this algorithm, including both its theoretical analysis and experimental validation, is presented in Sarmanho Jr, *et al.* (2010).

4.2 The PID controller with Anti-Wind-Up.

PID controllers are largely used for many applications because they are easily deployable, cheap and very versatile, being able to change both transient and steady-state behavior of the controlled systems (Ogata, 2003). In this work a PID control law as described in Eq. (8) is used for controlling the velocity of the vehicle during its braking procedure.

$$u(t) = K_p\tilde{x}(t) + \frac{K_p}{T_i}\int(\tilde{x}(t) + w(t))dt + K_pT_d\dot{\tilde{x}}(t) \tag{8}$$

In Eq. (8), K_p is a proportional gain, T_i is integral time, T_d is the derivative time, $\tilde{x}(t) = x(t) - x_d(t)$ is the position tracking error and $w(t)$ is a auxiliary term associated to anti-wind-up loop. The parameters of the controller (K_p, T_i and T_d) are obtained using the so-called Relay tuning method (see, for instance, Ogata, 2003). This method is based on the application of a *Bang-Bang* control action to the system, so that the amplitude and period of the system response related to a specific control signal are used to obtain the critical parameters K_c (critical gain) and T_c (critical time) that are employed to achieve the suitable gains for the controller through the use of the well-known Ziegler-Nichols relations: $K_p=0,6K_c$; $T_i=0,5T_c$; $T_d=0,125T_c$.

In a standard PID controller, the control signal can saturate during the system operation. In this case, the integral portion of the controller can remain locked at its maximum value regardless the occurrence of changes in the input signal. In order to avoid this condition, it is necessary to add anti-wind-up capabilities to the PID algorithm. In this work, it was used a anti-wind-up technique presented in Bazanella and Gomes (2006), were the term $w(t)$ is given by:

$$w(t) = \frac{1}{T_{tc}}[sat(u(t)) - u(t)] \tag{9}$$

where T_{tc} is the time associated to the automatic correction of the integral term. Aström and Hägglund (1995) propose $T_{tc}=(T_dT_i)^{1/2}$ to determined T_{tc}. The block diagram of the anti-wind-up scheme is presented in Fig. 4.

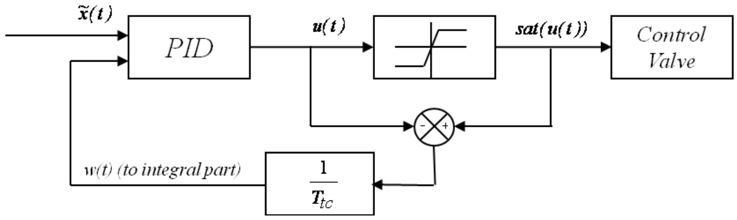


Figure 4. Block diagram of anti-wind-up scheme.

4.3 The Gain-Schedule technique.

It is well known that PID controllers have a limited range of application and that its performance is highly dependent on the operation point at which it is adjusted. One of the ways to alleviate this limitation is to choose more than one operation point, adjusting the most appropriate set of gains for each of them, and switching between these sets as the system goes from one operation point to another. This approach is commonly known the *gain-schedule* control technique (Aström and Hägglund, 1995).

In this study, the rule for changing the gain sets of the controller is based on the value of the estimated mass that is transported by the vehicle, whose calculation is made by means of the algorithm presented in Section 4.1. In Fig. 5, the complete control strategy is illustrated in a schematic way. In this figure, $\dot{x}_d$ is the desired velocity profile, g_{sch} represents the set of gains to be used in the PID controller, Δp is the differential pressure applied on the vehicle sails and u is the control signal. The variables measured onboard the vehicle are Δp (differential pressure between sails) and x (displacement of the vehicle), whereas the velocity of the vehicle is obtained by numerical differentiation of the position signal. The control cycle is started when the vehicle closes its doors and starts to move toward the other station, activating the mass estimation algorithm while the brakes are kept inactive. When the vehicle reaches the point located at 150 m before reaching its destination, the brakes start acting on the wheels. At this point, the estimated mass value is employed to choose the correct set of gains (g_{sch}) to be used by the PID control law during the remainder of the braking procedure. The values of the gain sets and of the masses at which such sets are switched are discussed in sections 5 and 6, in accordance with the specific nature of the test that is considered in each case (simulation or experiment). In Fig. 5, it is presented the block diagram of the full control strategy

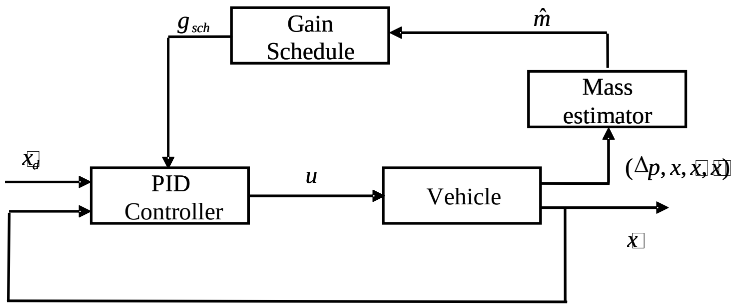


Figure 5. Complete control strategy.

5. SIMULATIONS

The simulations of the new control algorithm operation were performed using a simulation model implemented in Matlab/Simulink®. The velocity reference curve is presented in the following section.

5.1 Reference curve for braking procedure.

The reference curve employed by the controller was developed in a way that the vehicle decelerates at a fix rate of 0,7 m/s^2, starting at a distance of 150 m from the destination station. Using the Torricelli equation, the value of the desired velocity as a function of the position of the vehicle can be expressed as:

$$\dot{x}_d = \sqrt{\dot{x}_i^{\,2} - 2\ddot{x}_d x} \tag{10}$$

where, $\dot{x}_d$ is the instantaneous desired velocity, $\dot{x}_i$ is the velocity of the vehicle when the braking action starts, and $\ddot{x}_d$ is the desired acceleration. The shape of this curve is very similar to the reference braking curves used by Martins (1999) to simulate the braking process in unit freight trains. This velocity reference curve was used to evaluate the proposed brake controller both in simulations and experimental tests.

5.2 Calculus of sets gain to PID controller and Simulation Results.

The results presented by the recursive mass algorithm make it possible to define the following ranges of operation for the technique of gain-schedule: between 6797 kg and 8672 kg for the first set of gains; from 8672 kg to 10547kg for the second set; from 10547 kg to 12422 kg for the third set and between 12422 kg and 14297 kg for the last set. This ranges are useful to define the point of the PID have to be tuned in relation to the transported mass value (7735 kg, 9610 kg, 11485kg and 13360 kg). As a respect to the determination of the gains of the PID controller, it was used the "*Relay method*" described in Section 3.

The sets of tuned gains are presented in Table (2).

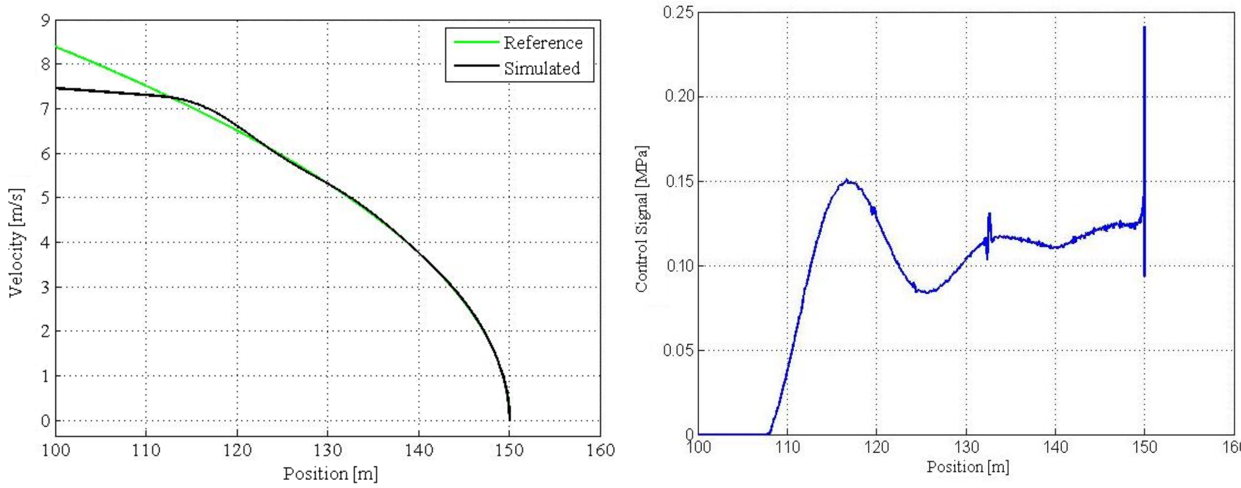
Table 2. Definition of sets of gain for PID.

Set of Gain	Nominal Mass [kg]	K_p	T_i	T_d	T_{tc}
1st	7735	3,8169	3,4509	0,8627	1,7254
2nd	9610	4,8433	3,4274	0,8569	1,7137
3th	11485	5,6357	3,4985	0,8746	1,7492
4th	13360	6,7315	3,6271	0,9068	1,8136

As depicted by Britto (2008), the nominal values of the physical parameters used for the simulation model of the Aeromovel system are: r_{brake} =0,115 [m^2]; μ_{discE} =0,4; μ_{discV} =0,003[s/rad]; A_{brake} =1,81E-3 [m^2]; c_{wheel} =0,005 [N.m.s/rad]; r_{wheel} = 0,2546 [m]; n_w=8; A *=0,98 [m²];* A_{eq}=6 [m²]; c_D=1,5; ρ=1,204 [kg/m^3]; v_{ar}=1[m/s]; c_{ved}=80 [N.s/m]; F_{ved} =450[N]; g =9,81 [m/s²]; p_{atm}=10,1325 [MPa]; μ_{cur}=0.025. The parameters relative to the adhesion force (F_{ades}) at the interface wheel/rail are those presented in Polach (2005) for dry contact conditions between the parts. The integration algorithm used both in the simulations and in the experimental tests is the *Euler* method, with a fixed step of 1 millisecond.

The simulation results presented in figures (4) and (5) are relative to the two extreme loading conditions of the vehicle: 7735 kg (empty vehicle) and 13360 kg (full capacity), respectively. In both cases, the displacement occurs from station *Gasômetro* towards station *Fazenda*.

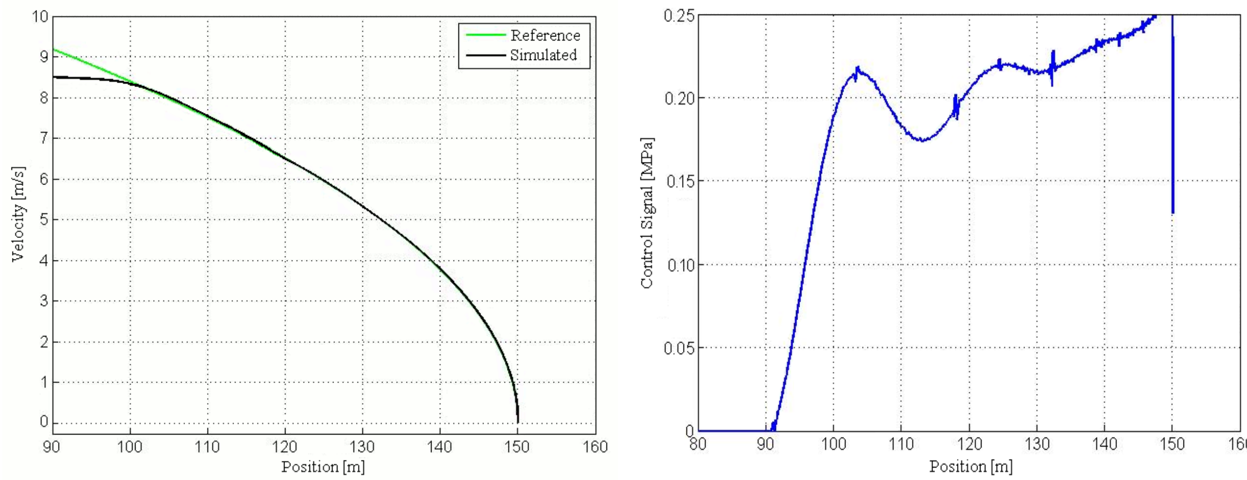
The simulations were carried out in accordance with the operation sequence described in Section 2. To the left of each figure (figures (4.a) and (5.a)), it is illustrated the how closely the controlled system is able to track the desired *position x velocity* profile, whereas the figures to the right (figures (4.b) and (5.b)) present the control signal that is applied to the wheels in each case, expressed in terms of the pneumatic pressure (between 0 and 0,25 MPa) that actuate on the braking system.



a) Velocity *versus* position

b) Control signal *versus* position

Figure 5. Braking action with transported mass of 7735kg (Simulation).



a) Velocity *versus* position

b) Control signal *versus* position

Figure 6. Braking action with transported mass of 13360kg (Simulation).

As observed in figures (5.a) and (6.a), simulation results indicate that the proposed controller leads to a satisfactory performance in tracking the desired velocity profile. The values of the final positioning errors obtained are 0,012 meters (1,2 cm) for the empty vehicle and 0,044 meters (4,4 cm) for full capacity. These values are in complete agreement with the desired limit of 10 cm for the final positioning error of the vehicle.

6. EXPERIMENTAL RESULTS

The experimental evaluation of the proposed control scheme was carried out at the test line installed in Porto Alegre, Brazil. The test vehicle has 8 wheels, divided into 2 groups with 4 wheels each. Its braking system is very similar to those found on commercial trucks, in which a pneumatic pressure acts on the brake booster in order to move a hydraulic piston located inside the master cylinder, which causes the contact of the brake pads with their discs. Then, angular velocity of the wheels is reduced by the action of friction effects on the braking disk. The actuating system of such braking equipment is comprised of two parts. One is exclusively pneumatic, equipped with an air supply system (compressor + reservoir + filter) and a proportional valve (Festo® MPPE-3-1/4-10-420-B). The other is pneumatic-hydraulic, where 4 compositions (power brake booster + master cylinder) are responsible for amplifying the pneumatic pressure and distributing the hydraulic fluid for the brake calipers located in each wheel of the vehicle.

The acquisition and processing of the measured data and calculation of the corresponding control action were effectuated by means of a DSpace® - DS1103 electronic board attached to a standard personal computer. Programming of the system was carried out with a Matlab-Simulink® package. The variations of the number of the passengers transported by the vehicle were emulated by the use of barrels containing water, each of them with mass of 75 kg ± 3%.

The measurement of the position was accomplished by employing inductive sensors and metallic gears attached to the vehicle wheels, in an arrangement similar to that of a standard position encoder. When operating in quadrature mode, this apparatus has an input resolution of 0,008 m (0,8 cm). For measuring the differential pressure on sails, it is used a transducer with a measuring range of -20 kPa to +20 kPa. The complete procedure of calibration procedure of the instrumentation apparatus can be found in Sarmanho Jr. (2009).

The evaluation of the positioning deviation was made manually after the vehicle stopped at the station by using a reference apparatus, as presented in Fig. (7). It is based on a mast stuck positioned beside the vehicle and a metrical tape with nominal length of 5 m and resolution ±0,001 meters (±1 mm). The center of the tape scale (2,5 m mark) was taken as the desired final position of the vehicle, being positioned 150 m away from the point where the braking procedure starts.

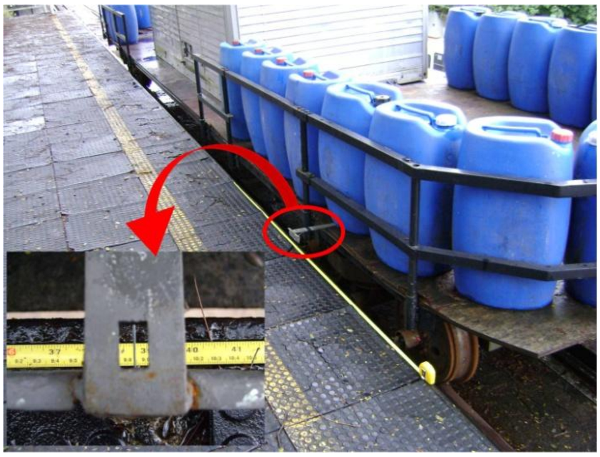


Figure 7. Method to measure the brake error.

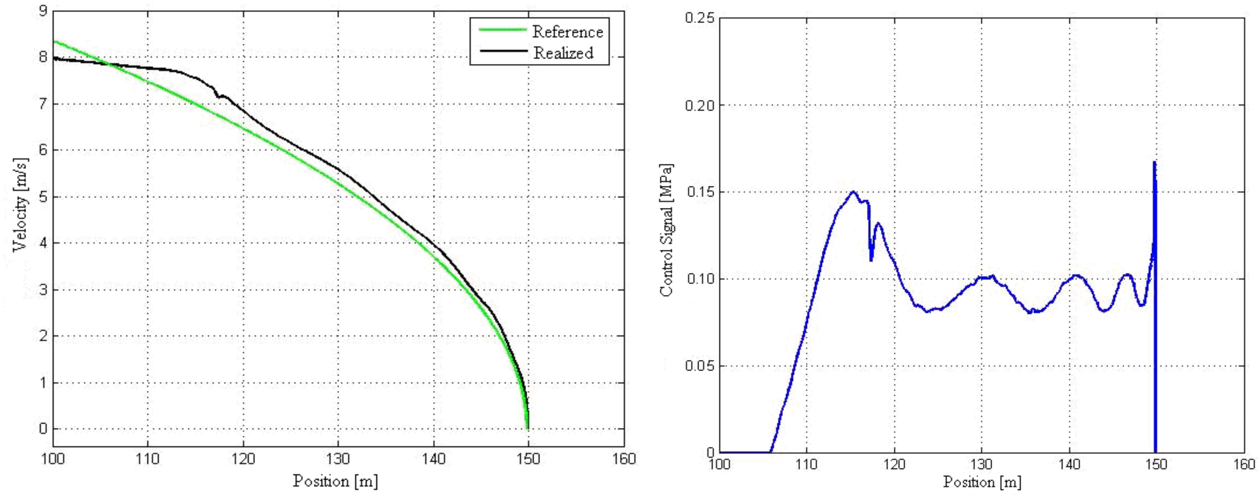
An important feature to be highlighted in this control application is the fact that the control action is exclusively passive. Therefore, it is only possible to reduce the vehicle speed (it cannot accelerate the vehicle). Because of this feature, control action is applied only when the measured speed exceeds the reference vehicle speed during the automatic braking procedure. Thus, the controller acts only in the presence of the positive error deviations. For this reason, the sets of gains applied to PID controllers should be selected in a criterious form, taking into account that if the control action is too large, the error can become negative, making the braking action resulting null. This fact can introduce a significant error if it occurs near the stopping point at the station, because it can induce the vehicle to stop before the reference point. Taking into account this possibility of error, the suitable control gains must be chosen in a way that the error trajectory is small but positive as much as possible along the vehicle trajectory.

The experimental evaluation of the proposed control scheme was carried out in the same condition of the simulation tests discussed in Section 5: the vehicle initiated its trip at *Gasômetro* station and concluded it at *Fazenda* station after following the procedure steps described in Section 6. The values of the gains for each set employed with the PID controller were tuned by direct experiments, taking the values determined by simulations as initial references. In the experimental tests, it was verified that only two different sets of gains in the gain-schedule implementation were enough to render satisfactory positioning results. The values of such gains are presented in Table (3).

Table 3. Experimental sets of gain for PID controller.

Set of Gain	Mass range [kg]	K_p	T_i	T_d	T_{tc}
1^{st}	7735 — 10547	2,0000	3,3333	0,0020	0,0816
2^{nd}	10548 — 13360	4,500	1,6667	0,0100	0,1291

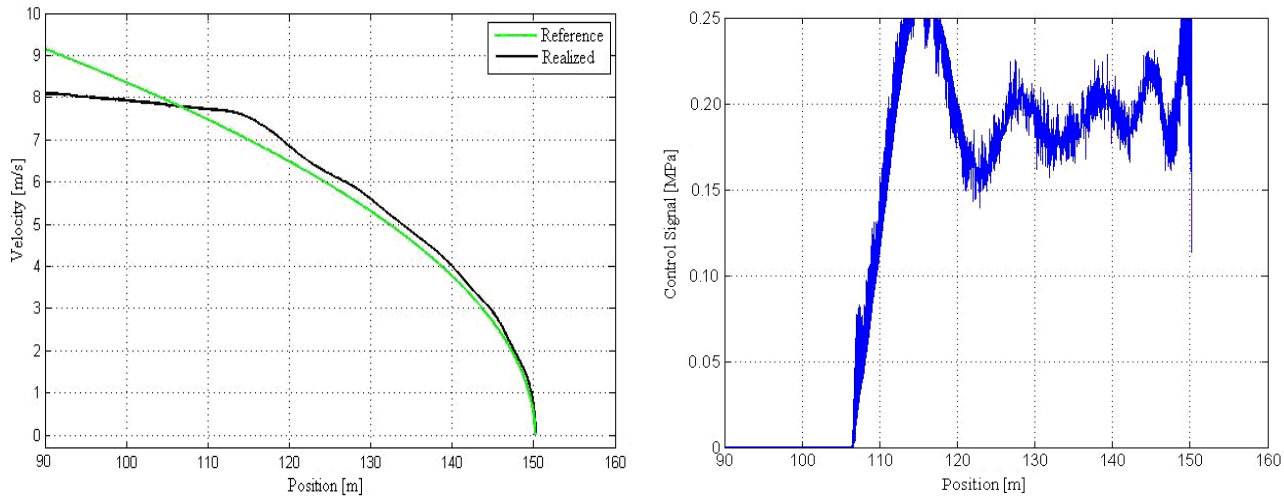
The experimental results observed for the same extreme loading conditions discussed in Section 5 are presented in figures (8) and (9). The presentation order of the results is the same employed in that section. The complete description of the experimental procedures involved in the evaluation of the controller and the data relating to all tests that were performed can be found in Sarmanho Jr. (2009).



a) b)

Figure 8. Braking action with transported mass of 7735kg (Experiments).

In the test with nominal mass of 7735 kg showed in Fig (8.a), it is possible to notice that control action occurs in such a way that the vehicle speed converges smoothly to its reference value. Also, the experimental braking action is seen to be qualitatively similar to its simulated prediction. In this test, the mean error of the final position is 0,243 m and the experimental standard deviation is 0,055 m. This deviation is roughly 45% lower than the desired limit of 0,10 m, allowing the utilization of the automatic doors between vehicle and stations for this case.



a) b)

Figure 9. Braking action with transported mass of 13360kg (Experiments).

In the case of maximum loading condition (Fig. 9), the results are qualitatively similar to those observed in Fig. (8). However, the control action is significantly larger, and its peak is large enough so as to cause saturation of the differential pressure that can be applied to the actuator. This result is not surprising since the transported mass is also significantly larger. It is also observed that the noise level of the signal sent to the valve is also substantially increased in this case, a phenomenon that can be attributed to the larger value of the derivative gain employed by the PID controller (see Table 3). As for the stopping error at the station, its mean value is 0,230 m and its standard deviation, 0,067 m. This deviation is roughly 33% lower than the desired limit of 0,10 m.

The analysis of experimental standard deviations obtained in the discussed tests shows that the use of the gain-schedule technique allows a significant reduction in the deviation of the final position of the vehicle when it stops at its destination. Thus, combined with a small correction in the starting point of the reference velocity profile to be performed by the vehicle, it is possible to ensure that the stopping errors of the vehicle at the station are confined to a range of about 7 centimeters (0,07 m). This value is in accordance with its allowable limit of 0,1 m. Therefore, if the proposed control scheme is used, the vehicle and the stations can be safely employed with automatic doors, as intended for the commercial version of the Aeromovel system.

7. CONCLUSIONS

In this paper, a new control scheme for the friction brake system of the Aeromovel, aiming stopping the vehicles on desired points at the stations was presented. Such controller is based on the association of a recursive algorithm used to estimate the vehicle mass and a PID controller that operates with different sets of gains, according to explicit value of transported mass.

Both simulation and experimental results were used to demonstrate the effectiveness of the control scheme in reducing the standard deviation of the final position of the vehicle at the stations. This values are significantly lower than its allowed limit of 0,10 m, even if its loading conditions are greatly varied. Therefore, it was show that employing the proposed controller, it will be possible to use automatically controlled doors on the vehicle and at the stations, thus improving the overall safety of the system.

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9. RESPONSIBILITY NOTICE

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