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ATS/AGV
Design, Implementation and Evaluation of a High Performance AGV

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Abstract

Within the framework of a full scale hardware-in-the-loop intelligent vehicle simulator called VEHIL, an automatic guided vehicle is developed in order to simulate traffic participants. This vehicle, called ATS/AGV, is based on a four wheel drive and four wheel steer concept which allows for a high handling performance in combination with the ability to carry out very complex manoeuvres. A structured design methodology appears to be essential to a cost and time effective design and technical specification of the AGV. This design methodology is strongly based on a dynamic model of the AGV which incorporates essential dynamics and non-linearities, the latter mainly arising from the tyre–road contact behaviour. The other important aspect is the application of test rig experiments. Within this design methodology, the development of the path control system takes place in an early stage and is carried out in close co-operation with the mechanical hardware specification. This makes the AGV a truly mechatronic product. Experiments show that ATS/AGV is capable of executing complex manoeuvres with a high dynamic performance.

1 Introduction

TNO Automotive is currently developing a professional full-scale Intelligent Vehicle Simulator, called VEHIL [1]. With VEHIL new prototype intelligent vehicles can be tested in a synthetic traffic flow. First the intelligent vehicle under test (VUT) is mounted on a roller bench. Then the VUT is placed in a simulated environment consisting of mobile robots, called Moving Bases. These Moving Bases are key components of the hardware-in-the-loop simulator as they represent the surrounding traffic of the VUT. For this purpose every Moving Base simulates the motion of a specific traffic participant relative to the VUT. As a matter of fact, these Moving Bases are intelligent vehicles in itself as they are capable of autonomous path following. As a consequence the Moving Base is a true Automatic Guided Vehicle (AGV). To be able to carry out the complex manoeuvres associated with the relative motion in VEHIL, this AGV must have an extreme freedom in manoeuvrability. Moreover, its dynamic response on position commands has to exceed the handling performance of modern road vehicles considerably. This results in a set of requirements for the AGV that is new in the world of automated vehicles.

Within the framework described above, TNO Automotive has developed a unique high performance AGV, called *ATS/AGV – Advanced Transport Systems / Automatic Guided Vehicle*. The AGV is presented in more detail in the next section. Section 3 deals with the design methodology which allowed TNO Automotive to develop the AGV in a relative short period of time. Section 4 describes the dynamic model of the AGV. This model proved to be an essential tool in the hardware specification of the AGV as well as in the control system design. The latter will be described in section 5. The experiments carried out to evaluate the performance of the AGV will be discussed in section 6.

2 System Description

ATS/AGV is a robot vehicle that responds to position commands issued by the VEHIL experiment controller [1]. In order to carry out the desired relative manoeuvres, ATS/AGV enables the independent control of its position in x- and y-direction, as well as its yaw angle ψ. In fact, ATS/AGV can best be compared to an industrial three degrees-of-freedom positioning robot. Such manoeuvring flexibility is accomplished through a vehicle platform equipped with independent all-wheel steering, and consequently independent all-wheel drive, as depicted in figure 1. This figure illustrates the platforms capability of motion in *x*-direction (a), crabwise motion in *y*-direction (b), pure rotation around its centre of gravity (c) as well as any combination of these types motion, i.e. rotation around a momentary pole *P* (d).

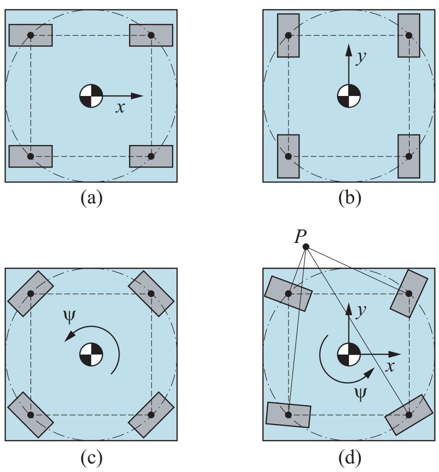

Figure 1 – Types of motion with three degrees of freedom

In practice, an emergency stop of a passenger vehicle corresponds to $10\ \mathrm{m/s^2}$ deceleration maximum. As a consequence, the AGV is capable of accelerating with $10\ \mathrm{m/s^2}$ in order to simulate an emergency stop of the VUT. The dynamic manoeuvring behaviour of conventional passenger cars can be described in terms of yaw response to steer inputs and speed response to throttle/brake input. The corresponding transfer functions typically show a bandwidth in the 1 Hz frequency range. This implies that the AGV must at least have a bandwidth of about 5 Hz in order to minimise positioning phase lag. Finally the top speed, which in view of the relative VEHIL world corresponds to the maximum speed difference between two cars, should at least be equal to 50 km/hr. These additional traction and handling performance requirements are accomplished using four high power brushless DC servomotors for generating the traction forces and four additional brushless DC servomotors controlling the steer angles in a range of -350° – 350°. All electrical drives are of a servo disc type, allowing a compact design with a high torque/weight ratio. Special attention has been paid to the design of a low weight aluminium space frame vehicle chassis with the centre of gravity located close to the road. The on-board energy is provided by a battery package consisting of 288 NiMH D-cells, having an excellent dynamic behaviour with respect to charge/discharge time and a very good energy/weight ratio. The battery pack voltage equals 346 V and the amount of energy carried is 2.2 kWh, allowing for a 15 minute VEHIL test cycle. Table 1 lists the main ATS/AGV specifications.

Table 1 – Specifications of ATS/AGV

vehicle mass	450 kg
wheel base	1.4 m
track width	1.4 m
maximum speed	50 km/hr
maximum acceleration	$10\ \mathrm{m/s^2}$
installed power	30 kW
acceleration from 0 to 50 km/hr	2.1 s
battery pack	NiMH D-cells

Figure 2 shows ATS/AGV in its final shape.

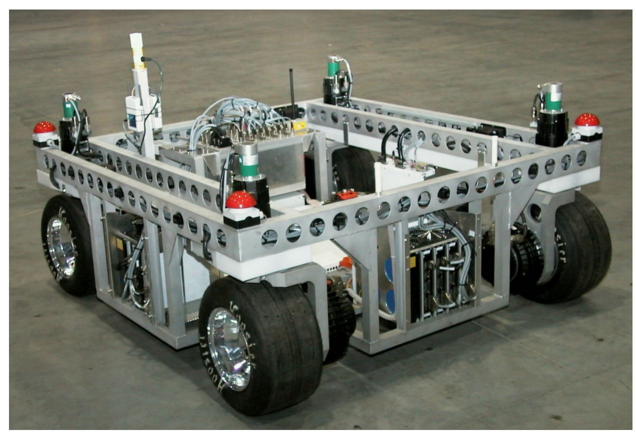

Figure 2 – The ATS/AGV

ATS/AGV is essentially a positioning device that responds to open loop position reference commands x_{ref}, y_{ref} and ψ_{ref}. These reference signals are communicated to the AGV by wireless ethernet. The actual positions x, y and ψ are measured by a laser position sensor system and are fed back to the AGV controller, also by wireless ethernet. The position measurement system, developed by MTI [2], consists of a fast rotating laser on board of the AGV and three beacons that are placed along the border of the area to be covered. Every time the laser hits a beacon, a pulse is registered. In this way the three beacons give three pulses per revolution of the laser as indicated in figure 3. The delay times between these pulses are then used to calculate the *x,y*-position of the AGV. An additional heading pulse, broadcasted from the AGV to the roadside, allows for measurement of the yaw angle ψ.

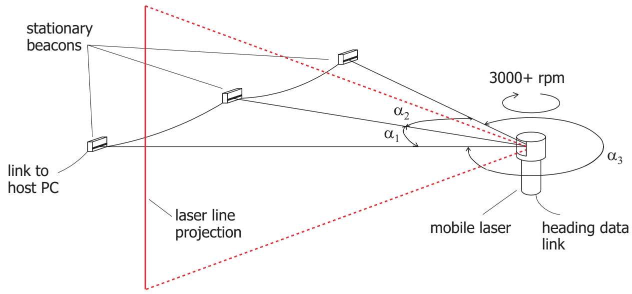

Figure 3 – Operation principle of the position measurement system

3 Design Methodology

Considering the high performance requirement and the tight time schedule, an efficient design process had to be applied. For this reason a thorough design methodology, as shown in figure 4, has been adopted which allows for a structured and efficient design process. The scheme of figure 4 is a proven design methodology in the development of automotive chassis systems such as ABS, active suspension, steer-by-wire, etc.. It consists of four activity clusters, together establishing the development process. These clusters do not represent independent, consecutive project phases but instead have a pronounced interaction.

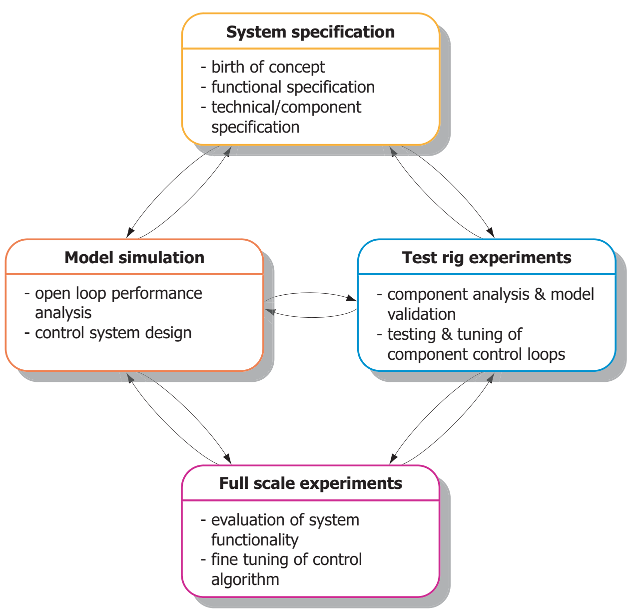

Figure 4 – Model based design methodology

Generally, the design process starts with the concept definition, followed by a functional specification and a component specification. These activities belong to the system specification phase. In this phase it was decided to apply a four wheel drive and steer concept for ATS/AGV according to figure 1 because of its potential to carry out complex manoeuvres at high speeds and accelerations.

An important feature of the design methodology is the application of a dynamic simulation model, in the Model simulation cluster, to support the system specification. This model played a crucial role in the specification of the electric drives, the desired tyre characteristics, the battery package and the sensors. Preliminary control system design already takes place in this design stage as well. This makes the design methodology truly mechatronic, in the sense that the mechanical hardware and the control system are designed from the very beginning to support each other; in practice this usually meant that high quality mechanical hardware was applied in order to minimise control system development effort. Emphasising the design methodology and the resulting mechatronic nature of the AGV, this paper mainly deals with the dynamic model (section 4) and the control design (section 5).

The result of the specification phase is a detailed technical specification, based on which the mathematical model can be further developed. This in turn allows for the final path controller design. Another iterative phase is entered when the control system is implemented in the vehicle and full scale experiments are carried out. Due to unmodelled phenomena such as resonance dynamics and coulomb friction in the mechanical systems, implementation of the ATS/AGV control system in this phase of the project led to adaptation of the mathematical model and thus a re-design of the control system.

In all stages of the design process, test rig experiments play an important role as they allow for the validation of component models, component specification and the testing/tuning of local control loops. During the ATS/AGV design for instance, specific test rig experiments were carried out for the electrical drives, their local speed or steer angle control loops and the selection of the battery cells. The overall performance evaluation however must be based on full scale ATS/AGV experiments, as shown in section 6.

4 Dynamic Model

In order to analyse the four wheel drive and four wheel steer AGV concept as part of the design methodology, it is necessary to develop a mathematical model of the relevant static and dynamic behaviour of the AGV. Consider for this purpose figure 5. This figure shows the body of the AGV as well as the forces and torques acting on it. These forces and torques, originating in the tyre–road friction of the four wheels, cause the vehicle body to move over a certain path. This path can be described as a rotation with respect to a momentary pole *P*. As the car body is assumed to be rigid, the path of each specific point of the car body has the same momentary pole *P*. Figure 5 demonstrates this principle by means of the paths of the body corners and the centre of gravity (c.o.g.). The resulting position of the body is uniquely described by three co-ordinates, being x, y and the yaw angle ψ of the c.o.g. with respect to a fixed point *O*. Note that vertical motion of the body is neglected because the ATS/AGV does not have wheel suspension. This leaves a negligible vertical motion caused by elastic deformation of the tyres. The speed and acceleration of the body can also be described in Cartesian absolute co-ordinates. It is however common to define speed v and acceleration a in longitudinal and lateral direction with respect to the body, indicated by the subscripts *long* and *lat* in figure 5 respectively.

Based on the above, the body model can be formulated in the Laplace domain as:

$$\begin{aligned} &\boldsymbol{M}s^2\boldsymbol{x} = \boldsymbol{f}\left(\boldsymbol{F}_{1L},\boldsymbol{F}_{1R},\boldsymbol{F}_{2L},\boldsymbol{F}_{2R},\delta_{1L},\delta_{2L},\delta_{1R},\delta_{2R}\right) \\ &\boldsymbol{F}_{ij} = [F_{ij,long} \quad F_{ij,lat} \quad M_{ij,z}]^{\mathrm{T}} \quad i = 1,2 \quad j = L,R \end{aligned} \tag{1}$$

where $F_{ij,long}$ (with $i = 1,2$ indicating the front and rear side respectively and j = L,R indicating the left and right side of the vehicle) are the longitudinal slip forces, $F_{ij,lat}$ the lateral forces, $M_{ij,z}$ the torques acting in the tyre–road contact patch and δ_{ij} are the steer angles. The vector $\boldsymbol{x}$ is the generalised position vector $\boldsymbol{x} = [x \quad y \quad \psi]^{\mathrm{T}}$ of the c.o.g. and $\boldsymbol{f}$ is a vector function describing the resulting forces in x- and y-direction and the resulting moment around the vertical axis at the c.o.g.. Finally $\boldsymbol{M}$ is a diagonal mass matrix. Note that all variables are functions of the complex frequency s. The external inputs of this part of the model are the longitudinal tyre forces $F_{ij,long}$ (via the drive line) and the steer angles δ_{ij} (via the electrical steering systems).

As already indicated, the main forces that control the

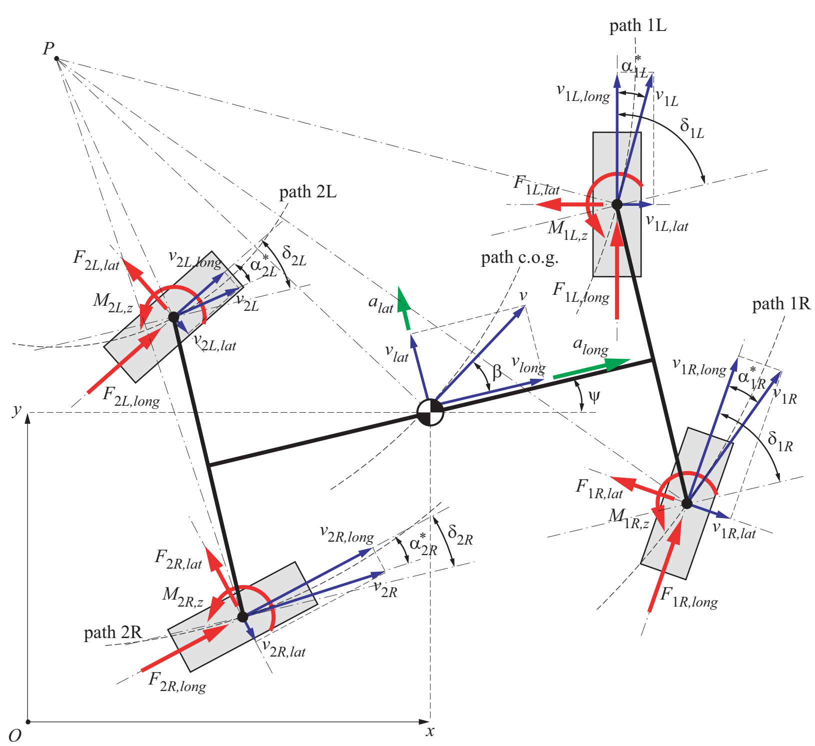

Figure 5 – Two-track vehicle model

course of the body are generated and transmitted in the contact patches between the tyres and the road. The longitudinal friction forces $F_{ij,long}$ are calculated using the Magic Formula [3,4], resulting in a friction curve which describes $F_{ij,long}$ as a function of the longitudinal slip κ_{ij}. Figure 6 shows on the left side the pure longitudinal slip curve as a function of κ_{ij}, parameterised to the local vertical load $F_{ij,z}$.

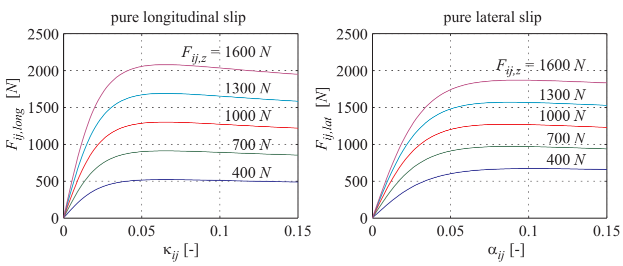

Figure 6 – Pure longitudinal and lateral slip characteristics, parameterised to vertical load $F_{ij,z}$

From this figure it follows that the tyre behaviour shows a highly non-linear character which strongly depends on the vertical load. This latter property plays an important role due to tyre load transfer during acceleration/deceleration and cornering. Although the vertical motion is not included in the model, these weight transmission effects are.

Introducing the approximately constant relaxation length σ_κ, the transient characteristic of the tyre in longitudinal direction can be described using a 1st order non-linear transfer function [5,6]:

$$\kappa_{ij} = \frac{1}{\sigma_\kappa s + \left| v_{ij,long} \right|} \cdot \left(R_e \omega_{ij} - v_{ij,long} \right) \tag{2}$$

where s is the complex frequency, ω_{ij} the wheel rotational speed (with $i = 1,2$ and $j = \mathrm{L,R}$), R_e the effective tyre rolling radius and $v_{ij,long}$ the longitudinal forward velocity of the tyre centre. With respect to the lateral tyre behaviour, the same type of friction curve applies as can be seen on the right hand side of figure 6. As a result of the tyres' construction, the longitudinal slip stiffness, i.e. the slope of the longitudinal curve at $\kappa_{ij} = 0$, is approximately three times higher than the lateral slip stiffness. The lateral slip α_{ij} corresponds to the tangent of the slip angle α_{ij}^*, refer to figure 4, and can be calculated similar to (2):

$$\alpha_{ij} = \frac{1}{\sigma_\alpha s + \left| v_{ij,long} \right|} \cdot v_{ij,lat} \tag{3}$$

where σ_α is the lateral relaxation length and $v_{ij,lat}$ is the lateral velocity component of the tyre centre. From (2) and (3) it follows that, on top of the non-linear and load dependent static behaviour of a tyre, the dynamic behaviour depends on the speed. In fact at low speeds, i.e. $v_{ij,long} \to 0$, the transient tyre behaviour approaches a pure integral behaviour. It can be shown that this physically corresponds to an undamped spring behaviour which is from a control point of view not very desirable. Fortunately, the control of vehicles at low speeds is inherently not very critical. It should be noted that figure 6 only shows the pure longitudinal and pure lateral tyre characteristics. In case of combined cornering and braking manoeuvres however, the total friction force is not equal to the vector addition of both the longitudinal and lateral forces. Instead an important part of the Magic Formula [3,4] deals with this combined situation which is also included in the model. Finally we mention the self aligning torques $M_{ij,z}$ and the rolling resistance of the tyres which have been included in the model as well.

The AGV is equipped with eight electrical drives: four for each steering system and four motors as independent wheel drives. Each electrical drive is torque controlled with a dynamic bandwidth largely exceeding the frequency region of interest to vehicle path control. For this reason the torque controlled drives could simply be modelled as 1st-order dynamic systems generating a desired drive torque:

$$T_{ij} = \frac{1}{\tau_e s + 1} \cdot T_{ij,ref} \tag{4}$$

where $T_{ij,ref}$ is the desired drive torque at wheel i,j and T_{ij} the actual torque; τ_e is the electrical time constant of the drive. The electrical steering drives are modelled using a similar dynamic model.

Figure 7 shows the response of the model to a reference speed v_{ref}, applying a temporary speed controller. The speed reference curve is carefully chosen not to 'push' the tyre characteristics into their non-linear behaviour too much and is limited by the maximum available battery power (lower acceleration at higher speed). This figure clearly shows that the desired acceleration of 10 m/s^2 is reached (up to 20 km/hr). The lower left figure shows the reference drive torques $T_{ij,drv,ref}$, being the controller outputs. It can be seen that the rear drives ($i = 2$) have a higher reference torque than the front drives ($i = 1$). This control strategy will be explained in the next section. The mechanical power curve in the lower right corner, illustrates that the ATS/AGV is almost performing at its maximum.

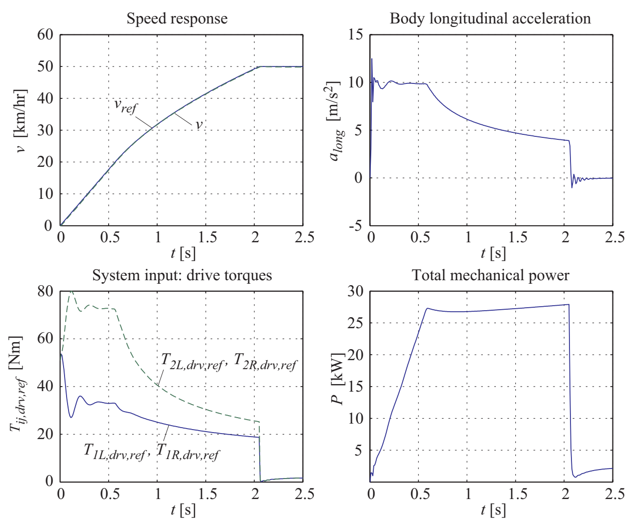

Figure 7 – Simulated speed and acceleration response

Summarising, the mathematical model shows that ATS/AGV is a non-linear system with respect to spatial distribution of its actuators. A more severe non-linearity lies in the tyre–road contact behaviour which is also responsible for the speed dependent dynamical behaviour of the AGV. An adequate mathematical description of these phenomena is of crucial importance to control design.

5 Control Design

The ATS/AGV, like any other AGV, has to drive from its current position to a designated position along a desired trajectory. This trajectory, which can be expressed in terms of a generalised reference position $\boldsymbol{x}_{ref}$ as a function of time t:

$$\boldsymbol{x}_{ref}(t) = [x_{ref}(t) \quad y_{ref}(t) \quad \psi_{ref}(t)]^{\mathrm{T}} \tag{5}$$

is generated from within the VEHIL experiment controller [1]. This reference trajectory arises from the relative motion of two vehicles with respect to each other. It is generated using dynamic vehicle models and the state of the VUT itself. The shape of the reference trajectory is therefore 'bounded' by the physical properties of road vehicles. Because the handling properties of ATS/AGV largely exceed those of ordinary road vehicles, ATS/AGV is always physically capable of following a generated trajectory. In other words: a generated trajectory is guaranteed to be feasible.

In order to realise the reference trajectory, the control system acts on the four reference torques $T_{ij,drv,ref}$ ($i = 1{,}2$, $j = \mathrm{L,R}$) of the electrical drives and the four reference torques $T_{ij,str,ref}$ of the electrical steering motors. The fact that there are three control objectives according to (5) and eight actuating system inputs, implies that the AGV is an over-actuated system. As a consequence there exist an infinite number of combinations of the system inputs that realises the control objectives. Obviously, a large number of combinations are undesired, e.g. front and rear drive torques acting in opposite directions in order to achieve a stand still of the AGV. This problem can be solved by introducing additional control objectives, based on a maximum performance requirement and/or a minimum energy consumption requirement. Both types of requirements are met to a certain extend when the desired drive line torques $T_{ij,drv,ref}$ take the tyre characteristics as shown in figure 6 into consideration: the higher the vertical load of a specific wheel, the higher the traction force and thus the drive line torque can be. This means in fact that the control system takes the weight transfer during accelerating/decelerating and cornering into account. This strategy is implemented in the so-called Motion Control which controls the generalised velocity $\dot{\boldsymbol{x}} = [\dot{x} \quad \dot{y} \quad \dot{\psi}]^{\mathrm{T}}$ of the vehicle's c.o.g. In the Motion Control first a desired generalised force in the c.o.g. is determined and then this force is distributed across the four wheels using inverse kinematics while obeying the weight transfer requirement. With the Motion Control being a velocity controller, the steer angles δ_{ij} are simply set according to the direction of body movement at the wheel position which is uniquely determined by the generalised reference velocity vector $\dot{\boldsymbol{x}}_{ref} = [\dot{x}_{ref} \quad \dot{y}_{ref} \quad \dot{\psi}_{ref}]^{\mathrm{T}}$ and the body geometry. In other words: the course of the vehicle is controlled by the steer angles. This is implemented using four independent steer angle control loops acting on the reference steer torques $T_{ij,str,ref}$. In fact this simple steer angle control mechanism is the main reason for application of velocity control. As a result of the Motion Control, the four wheel angular velocities and steer angles are controlled in such a way that the effective tyre slip per wheel is minimised.

The specific Motion Control loop design results in a decoupled system, i.e. each input $\dot{x}_{ref}$, $\dot{y}_{ref}$ and $\dot{\psi}_{ref}$ independently controls the corresponding output $\dot{x}$, $\dot{y}$ and $\dot{\psi}$ without any cross links (approximately). The control objective however is to realise a reference position trajectory. This tracking functionality is implemented by an outer position control loop. The resulting block scheme is depicted in figure 8. Because the Motion Controlled AGV is a decoupled system, the Tracking simply consists of three independent position control loops, one for each element of the generalised position. Note that the ATS/AGV including the Motion Control can be regarded as an AGV with an imaginary speed actuator located in the c.o.g. of the vehicle.

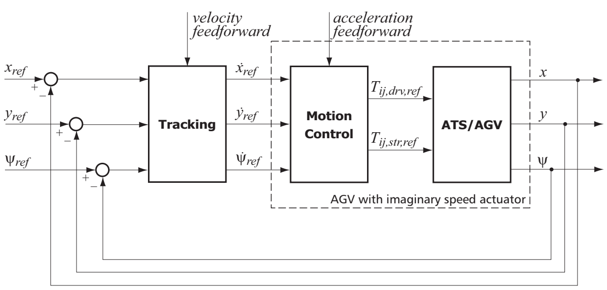

Figure 8 – Path control strategy of ATS/AGV

As already mentioned at the beginning of this section, the generalised position reference is generated using vehicle models. This implies that not only the generalised reference position is known, but also the generalised reference velocity and even the generalised reference acceleration. These can be very usefully applied as feedforward signals, greatly enhancing the dynamic performance of the path controller.

6 Performance Evaluation

Although the ATS/AGV is not yet fully operational, first experiments have already been carried out. These experiments can be categorised into dynamic response tests, indicating response time, bandwidth etc., and manoeuvrability tests, indicating the ability of the vehicle to follow an arbitrary, but feasible, trajectory $\boldsymbol{x}_{ref}(t)$.

Figure 9 shows an example of a response test. Here the drive line was separately tested using a fly wheel mounted to the driven axle, representing a quarter of the vehicle's inertia. Figure 9 shows the response to a smooth position 'step' of both the drive line model and the measured response. The smooth 'step' requires a maximum acceleration of 10 m/s^2. Tests have been carried out without and with feedforward in the path controller (refer to figure 8). Figure 9 clearly demonstrates the effect of using feedforward information for improvement of the dynamic tracking behaviour of the AGV: phase lag is decreased to a large amount. This advantageous property of feedforward has already been identified at a simulation level and is now confirmed with hardware measurements. As can be seen from figure 9, the model and the measured response quite well match, showing a fast response with acceptable overshoot.

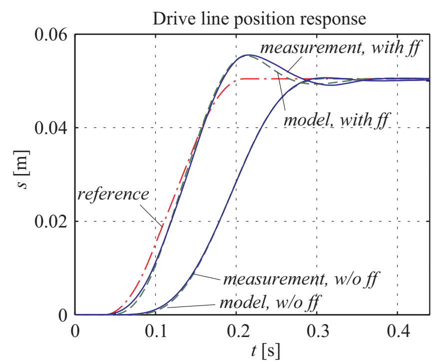

Figure 9 – Time response of position controlled drive line

The bode diagram of the same system in figure 10 confirms the fast response, showing a bandwidth of almost 5 Hz.

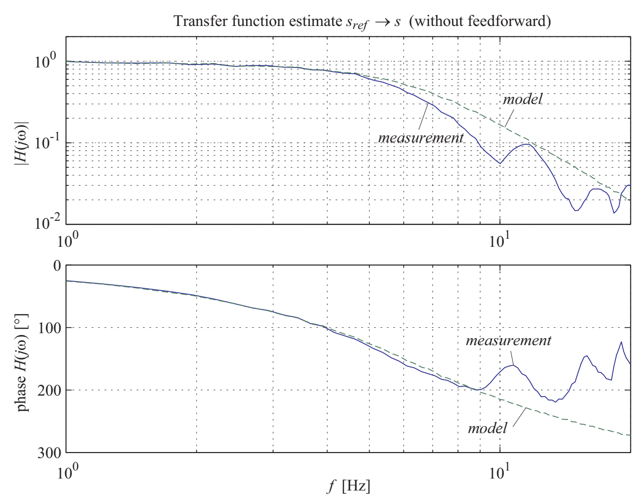

Figure 10 – Frequency response of position controlled drive line

This result is gained even without feedforward, which could not be applied due to the fact that the Bode diagram has been determined using a noise reference position input (recall that feedforward requires the reference velocity and acceleration). From figure 9, we can expect a significantly higher bandwidth in case feedforward is applied. The corresponding reduction in phase lag is of essential importance to a successful VEHIL test.

Finally, figure 11 shows the preliminary result of a manoeuvrability test, being the measured response to an 8-shaped position reference track $\{x_{ref}(t), y_{ref}(t)\}$ with constant orientation $\psi_{ref}(t) = 0$. The vehicle speed during this track is 5 km/hr. Deviations of the actual position from the reference track can hardly be distinguished in this plot and appeared to be less than 6 cm. The test was carried out with a path control strategy without the velocity and acceleration feedforward information, but with MTI position information for the feedback control loop. The feedback gains are not tuned up to their maximum yet.

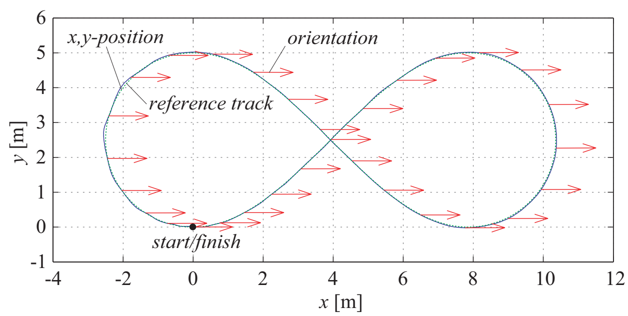

Figure 11 – 8-track response with constant orientation

7 Conclusions

The research presented here leads to the following conclusions:

- The applied design methodology enables the efficient development of a high performance AGV within a relative short period of time.
- The dynamic model is of essential importance within the methodology.
- ATS/AGV is capable of complex manoeuvres with a high dynamic performance.
- Application as a moving base within the VEHIL simulator as well as 'stand alone' AGV appears to be very feasible.

Further research is directed towards the applicability of the developed path control algorithm to different types of vehicle platforms, with respect to actuator configurations. Another important research theme is the robustification of the developed path control algorithm, in order to achieve a high level of safety and reliability of the controlled AGV with respect to varying environmental conditions and neglected (vertical) dynamics of the AGV.

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