

Laboratory Assignments for the course
Multibody Dynamics A, wb1310,
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Assignment 1

The model of a physical pendulum consists of a slender rigid bar made out of steel with length $l = 37$ cm and a circular cross section with diameter $d = 10$ mm. At the top the bar has a hinge joint fixed to the ground. The system moves in a gravitation field with a field strength of $g = 9.81$ [N/kg] in the downward direction.

1. Make a pencil-and-paper model for the pendulum and estimate the Period of Vibration T in [sec] for small oscillations around the equilibrium position.
2. Make a model of the pendulum in ADAMS and simulate the motion of the pendulum for a couple of periods. Take as initial conditions an initial angle of $\phi = 10^\circ$ from rest. Plot the angle as a function of time and measure the Period of Vibration T in [sec]. Compare this with your estimate from above.
3. Plot the forces that the ground exerts on the pendulum in the upper hinge during one period of motion. How big is the average, the amplitude and the frequency of the vertical force? Explain this with your pencil-and-paper model.
4. Now simulate the motion of the pendulum from rest with some larger initial angles, $\phi = 30^\circ, 60^\circ$ and 90° . Measure from the plot the Period of Vibration T and explain the difference with the preceding findings.
5. Investigate the influence of the desired integration accuracy by continuing the simulation for at least 50 Periods of Vibration. Next, decrease for example the Error from 10^{-3} to 10^{-6} and compare the results. This parameter is part of the Simulation Settings. (Note: Click Defaults in order to return to the default values!) What can you say about the accuracy of the solution?

Assignment 2

In order to examine the steering forces and moments of a bicycle in motion, the front wheel of a bike is put to a further test. The model of this front wheel consists of a thin hoop with a

mass of $m = 2.7$ [kg] and a diameter of $d = 700$ mm, which can rotate around its own axle, ϕ . It is assumed that the mass is concentrated along the perimeter of the wheel. Perpendicular to this axis of rotation, a second hinge has been attached in order to be able to rotate the ϕ -axle, the so-called steering, around an angle ψ .

1. Make an estimate, by means of the Euler equation of motion for a rigid body, $\mathbf{M} = \mathbf{I}\dot{\boldsymbol{\omega}} + \boldsymbol{\omega} \times (\mathbf{I}\boldsymbol{\omega})$, of the size and direction of the moment \mathbf{M}_1 that is exerted on the wheel in the first hinge at a constant riding speed of $v = 20$ [km/hour] and a constant steering angular velocity of $\dot{\psi} = 60$ [°/sec].
2. Make a model of the steering front wheel in ADAMS. Remember that there is a body between the two hinges, namely the fork. For the sake of simplicity, the fork offset (trail) and head angle are assumed to be zero. Simulate the motion for some time with the initial conditions from above. Plot the moments in the hinge of the wheel axle as a function of time and compare this with your estimate of \mathbf{M}_1 . By generating a Measure you will be able to follow the rotating hinge moment during simulation.

Assignment 3

In order to examine fast and correct positioning of a container on an Automated Guided Vehicle, we study the model of an overhead container crane (gantry crane). Here we only consider the motion of the crab plus container, the hoisting motion will not be considered. The cables in which the container hangs, are modelled as compliant spring-damper elements. The container measures 8x8x40 [ft] at a total mass of 30 [metric tons] (dead weight plus load). You may assume a uniformly distributed mass for the calculation of the mass moment of inertia quantities. The vertical distance between the bottom of the crab and the top of the container is 25 [m]. The supporting cables are modelled by 4 cables which are mounted on the container in a purely vertically centered way, with a distance of 2 [m] in transverse and 6 [m] in longitudinal direction between them. Each individual cable has a stiffness of $3.0 \cdot 10^6$ [N/m] and a relative damping of 5 %. The driving motion of the crab is thought of as: first a constant acceleration, then a constant speed and finally a constant deceleration. The maximum speed is $v_{max} = 240$ [m/min] and the maximum ac(de)celeration is $a_{max} = 0.7$ [m/sec²]. A crane expert told us that for the load to hang still after the crab's driving motion, we have to accelerate and decelerate in a multiple of the Period of Vibration T of the load oscillation.

1. Make a pencil-and-paper estimate of the Period of Vibration T of the load oscillation in the crab's driving motion.
2. Determine the Period of Vibration of the load oscillation in the crab's driving motion of your ADAMS model, for instance by means of an eigenfrequency analysis (Simulation Control → Linear).
3. Simulate the motion of the container assuming a total displacement in the driving direction of the crab of 60 [m]. Check the statement of the crane expert, is he right? Plot the speed of the crab and the speed of the container in the crab's driving direction as a function of the *displacement* of the crab in the driving direction.

The crab motion can be generated by using a Translational Joint Motion. First create such a joint with a random value, after which the function F(time)= can be modified in the Function Builder. By summation of a number of STEP functions you may obtain the desired motion.

4. As a rule, the container will be loaded asymmetrically and we are wondering what the effect of this will be on the container's positioning on the A.G.V. Assume the centre of mass to be 20 % = (1.6, 1.6, 8.0) [ft] off from the geometrical centre (keep the mass moments of inertia at the centre of mass as is). Change the model and simulate the same driving motion. Plot the container's angular rotation around its vertical axle as a function of time and compare this result with the symmetrical situation. Is there a significant difference such that positioning of the container on the AGV will give trouble? Explain!

Assignment 4

In a standard textbook on kinematics and dynamics of mechanical systems [1], the steady state analysis of a regulator is discussed as an example. A regulator is an instrument that tries to keep the speed of an engine constant, irrespective of the load. This can be traced back to James Watt and his steam engines. The literal text of the example is given as an appendix. In this assignment we will eventually make and simulate the complete model of this system in 5 steps.

1. Make a simplified model of the regulator in ADAMS. Only model the *Ground*, *Spindle*, *Ball1*, *Ball2* for the given geometry of the model.
2. Make an estimate of the equilibrium (steady state) at an angular velocity of $\omega = 105$ [rpm]. Check this by means of a simulation in ADAMS. Think of a way to damp out the transient response to get to the steady state.
3. Make the complete model of the regulator in ADAMS. Apparently the distance constraint does not exist in the ADAMS software. Replace this constraint by means of a stiff spring-damper element between the arm and collar. Make a well considered choice for the value of the stiffness and the damping (report!). Simulate this regulator at the steady state angular velocity $\omega = 105$ [rpm]. Check the position of the arms, these should make a 45° angle with the vertical, do they?
4. Next only add the feedback of the engine to your model and carry out a simulation which shows that in the absence of the disturbance the 45° equilibrium will be reached again.
5. Finally add the disturbance to your model and simulate the three different types of engines, take for the vertical spring a stiffness of $k = 1000$ [N/m]. Use another STEP function, as described in assignment 3, for the description of the external couple. Plot the angular velocity and the collar's vertical position and the spindle's angular velocity as a function of time and compare these results with those in [1], fig 12.6.5 and 12.6.6. Discuss this comparison.

Assignment 5

In order to examine the safety of a standard tractor driving along the public road at high speed (> 30 [km/h]) [2], the vehicle has to be submitted to an ISO dual lane change, also known as the moose test. This middle range tractor (AS1) has an undamped rear axle fixed to the body and a rigid front axle attached by a swivel axis to the body at the height of the wheel axle. The steering is realized by letting both front wheels rotate by the same angle around their vertical axes. The total mass is 5850 [kg] and the mass moment of inertia around the vertical or yaw-axis 9255 [kgm²] and around the pitch-axis 7245 [kgm²]. The mass moment of inertia around the roll-axis has not been given in [2] but is estimated at 5235 [kgm²]. The centre of mass of the system is situated 1626 [mm] in front of the rear axle and vertically 1095 [mm] in

relation to the ground (unloaded configuration). The wheel base is 2703 [mm] and the gauge at the front and the rear is 1900 [mm]. The rear tyres are of the 18.4R38 type with an unloaded radius of 789 [mm], a width of 467 [mm] and a total mass, tyre plus rim, of 273 [kg]. The front tyres are of the 16.9R26 type with a 621 [mm] radius, a width of 429 [mm] and a mass of 152 [kg]. The vertical stiffness of both tyres is $2.0 \cdot 10^5$ [N/m].

1. Make a model of the tractor in ADAMS. Model the front swivel axle as a separate body with a negligible mass and mass moment of inertia. For the tyres, we will make use of the **Delft-Tyre** Model. The Tyre Property Files for front and rear tyres are `16r26_new.tir` and `18r38_new.tir` and the Road Data File `FlatRoad.rdf` describes a flat level road. These files can be found on the course website. Let this model drive straight ahead at a low initial speed (< 5 [km/h]) in order to prove that your model is working.
2. Determine the vertical displacement and the pitch angle of the main body in the static equilibrium state. This static equilibrium can best be simulated by letting the tractor drive straight ahead at low speed from the initial state. This also gives the tyres a chance to spin up. Plot this vertical displacement and pitch angle as a function of time. Check these steady state values by a simple pencil-and-paper calculation.
3. Make a pencil-and-paper estimate of the eigenfrequency of the pitch movement and check this by means of a Linear analysis on your model.

Subsequently, we would like to submit the tractor to the moose test. This test, comprehensively described in ISO/TR 3888 [3], includes an overtaking manoeuvre or a fast swerving manoeuvre around an obstacle within a track marked by pylons. You can find the description of the track as an appendix. The easiest way to steer the front wheels is by letting them rotate sine-like. The amplitude and frequency of this motion for a desired vehicle track can be estimated by remembering that the vehicle's yaw-rate $\dot{\alpha}$ is approximately proportional to the steering position of the front wheels φ and the vehicle's forward speed v , and inversely proportional to the wheelbase l , summing up: $\dot{\alpha} \approx (v/l)\varphi$. In general the tractor will slow down as it moves along, therefore to maintain constant speed along the track you should construct a first order control system pushing and pulling the tractor forward. Do not try to drive the wheels, but let the control system work on the centre of mass of the tractor body.

4. Determine, by iterative analysis, the maximum speed v_{max} at which the tractor will safely pass the moose test. For this speed, plot the tractor's track in the horizontal plane. What is your advice on the maximal allowing speed of tractors on the public road?
5. Finally plot, as a function of time, the three tyre forces: normal, lateral and longitudinal, that the right front and rear tires are subjected to during the manoeuvre at maximum speed and compare this with the (average) tyre forces in the static equilibrium position.

Alternative Assignment 5

In order to examine the lateral stability of a bicycle we make a model in ADAMS. The bicycle is modelled by four rigid bodies connected by hinges: the rear wheel, the frame with a rigid rider attached, the fork and the handlebar assembly being the front frame, and the front wheel. For the geometry and parameter values we follow the bicycle benchmark paper from Meijaard *et al.* [4]. This also give us room to compare results. However, in this paper the contact of the wheels with the level ground is modelled by idealized pure-rolling constraints (nonholonomic constraints). Unfortunately, these type of constraints are not present in the

ADAMS software and therefore we will model the wheel contact by compliant tires. For the tires, we will make use of the **Delft-Tyre** Model. The Tyre Property Files for front and rear tires are `bicycle700c.tir` and `bicycle600c.tir` and the Road Data File `FlatRoad.rdf` describes a flat level road. These files can be found on the course website.

1. Make a model of the bicycle in ADAMS. Let this model drive straight ahead at a low initial speed (≈ 1 m/s) in order to prove that your model is working.
2. Simulate the motion of the bicycle for 5 seconds. The initial conditions are an upright position with the steering straight ahead, an initial forward speed of 4.6 m/s, an initial lean angular velocity, for the whole assembly, of 0.5 rad/s, and zero steering rate. Give a clear representation of the motion in a number of graphs of your own choose. Compare your results with those from Figure 4 of the benchmark paper [4].
3. Use your simulation to address one or more of the following questions (or some of your own):
 - (1) Below 4.3 m/s and above 6.0 m/s the bicycle is unstable. However, these instabilities have different characteristic motions. Demonstrate these instabilities by transient analyses on the ADAMS model and characterize the different motions.
 - (2) A folklore in bicycle science is that the wheel reaction forces in the contact point are in the plane of the wheel. True or not?
 - (3) Does the bicycle have any stable motions besides upright straight ahead?
 - (4) By changing mass and geometry parameters, what's the slowest speed for which you can find a stable bicycle?

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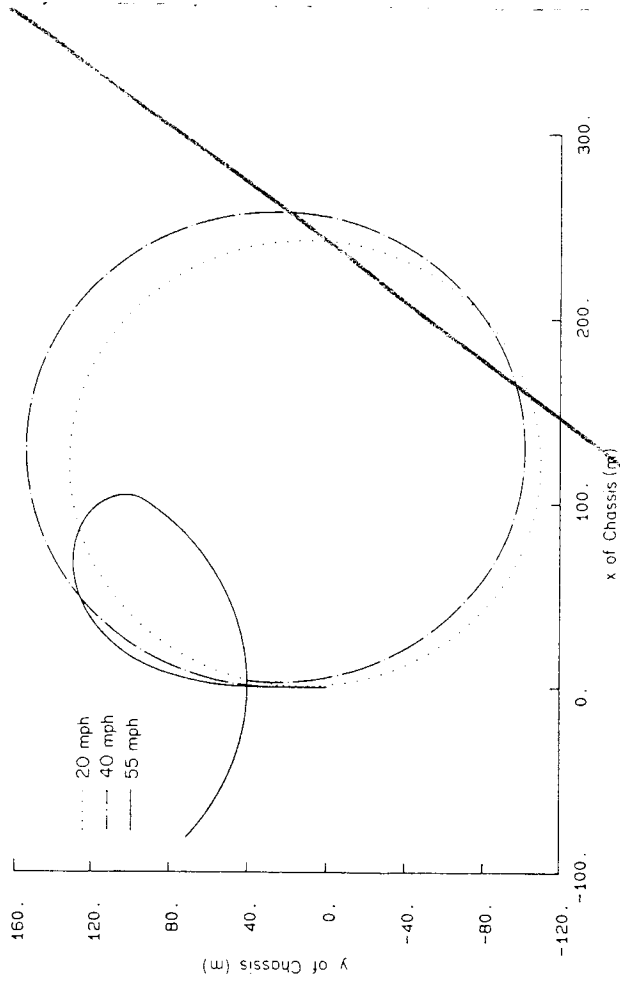


Figure 12.5.11 Chassis trajectory for circular steer input.

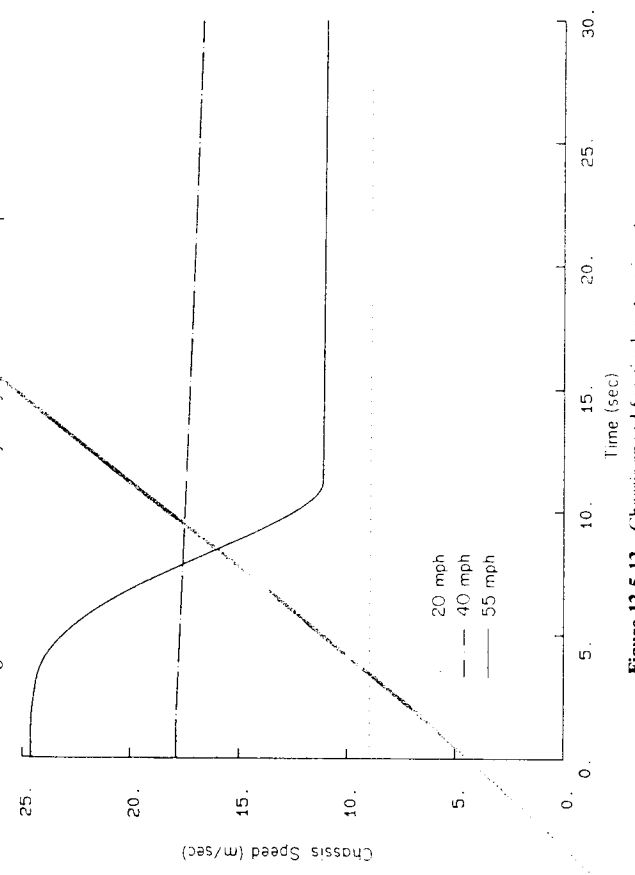


Figure 12.5.12 Chassis speed for circular steer input.

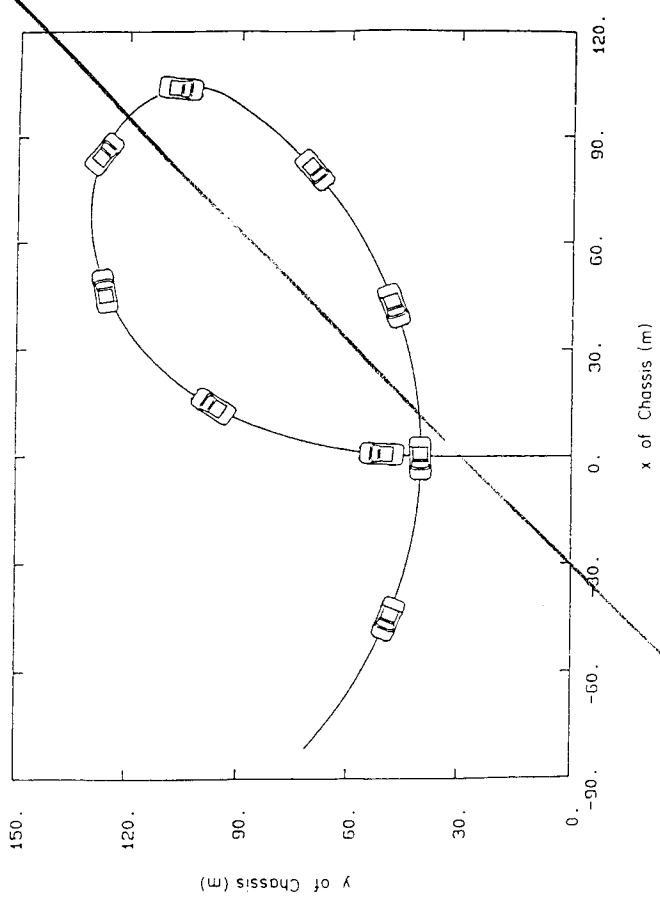


Figure 12.5.13 Vehicle orientation in circular steer maneuver (55 mph).

the speeds of the vehicle in the three different simulations. As expected, considerable speed loss (reduction in kinetic energy) occurs in the 55 mph case, since dissipative friction forces act while the vehicle is slipping laterally.

Figure 12.5.13 shows that the vehicle is not tangent to the path during the maximum slip period in the $y_0 = 55$ mph simulation.

12.6 DYNAMIC ANALYSIS OF A GOVERNOR MECHANISM

12.6.1 Alternative Models

The governor mechanism shown in Fig. 12.6.1 is composed of five bodies, including ground (body 1). Bodies 2 to 5 are shaft, ball 2, and collar, respectively. The arms attached to the balls are connected to the shaft by revolute joints whose axes are parallel, both perpendicular to the shaft. The

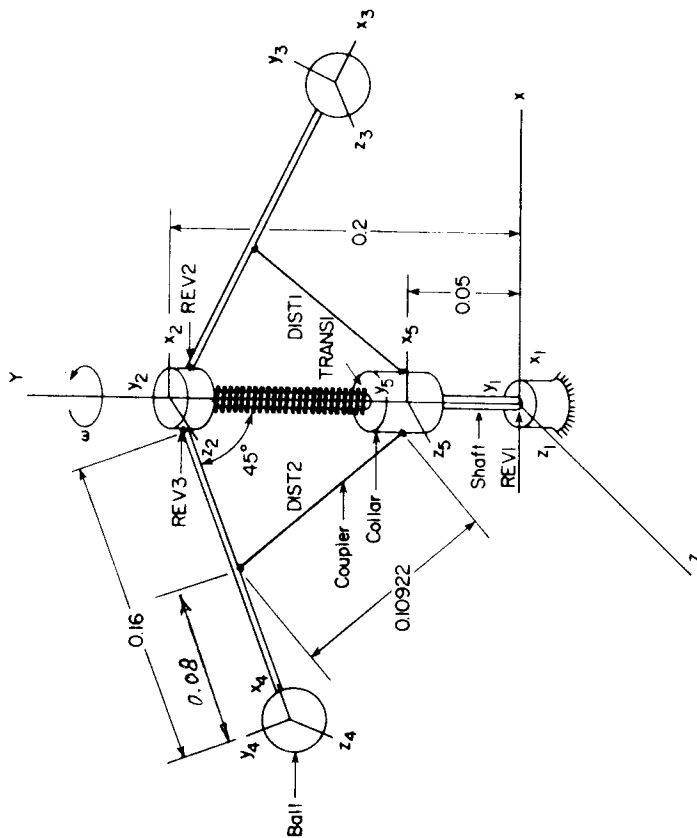


Figure 12.6.1 Governor mechanism.

collar is connected to the shaft by a translational joint. Couplers that connect the ball arms and collar are modeled as distance constraints. A TSDA element is attached between the shaft and collar. The inertia properties of the components of the system are defined in Table 12.6.1.

The intended function of the governor is to maintain a nearly constant

TABLE 12.6.1 Inertia Properties of Governor Mechanism

Body	Mass	I_x	I_y	I_z	$I_{x,y}$	$I_{y,z}$	$I_{z,x}$
Ground	1.0	1.0	1.0	1.0	0.0	0.0	0.0
Spindle	200.0	25.0	50.0	25.0	0.0	0.0	0.0
Ball 1	1.0	0.1	0.1	0.1	0.0	0.0	0.0
Ball 2	1.0	0.1	0.1	0.1	0.0	0.0	0.0
Collar	1.0	0.15	0.125	0.15	0.0	0.0	0.0

angular speed ω of the shaft under a varying resisting torque from the machine being driven. If the angular velocity of the shaft decreases, the balls drop and hence lower the collar. A linkage attached to the collar then opens the fuel feed to the engine, which generates an increased torque and leads to a speed-up of the shaft. As a result, the balls rise toward their nominal position as the shaft angular velocity approaches the desired value.

The elements of the kinematic model are as follows:

Bodies	Constraints	nc = 35
Five bodies		
Distance constraint: DIST1		1
Distance constraint: DIST2		1
Revolute joint: REV1		5
Revolute joint: REV2		5
Revolute joint: REV3		5
Translational joint: TRANS1		5
Ground constraint		6
Normalization constraint		5
Euler parameter		
		nh = 33

DOF = 35 - 33 = 2.

Data for the distance constraints, revolute joints, and translational joint are tabulated in Tables 12.6.2 to 12.6.4. Data for the TSDA element are given in Table 12.6.5. Three different models (1, 2, and 3) are distinguished, with spring constants $k = 1000, 2000,$ and 3000 N/m , respectively.

TABLE 12.6.2 Data for Distance Constraints

DIST1

Point	x'	y'	z'	x''	y''	z''	x'''	y'''	z'''	R
Body										
Ball 1 ③	-0.08	0.0	0.0	-0.08	0.0	1.0	1.08	0.0	0.0	0.0
Collar ⑤	0.0	0.0	0.0	0.0	0.0	1.0	1.0	0.0	0.0	0.0

Distance = 0.10922

DIST2

Point	x'	y'	z'	x''	y''	z''	x'''	y'''	z'''	R
Body										
Ball 2 ④	0.08	0.0	0.0	0.08	0.0	1.0	1.08	0.0	0.0	0.0
Collar ⑤	0.0	0.0	0.0	0.0	0.0	1.0	1.0	0.0	0.0	0.0

Distance = 0.10922

TABLE 12.6.3 Data for Revolute Joints

		REV1								
Body	Point	P			Q			R		
		x'	y'	z'	x'	y'	z'	x'	y'	z'
Ground	①	0.0	0.0	0.0	1.0	0.0	0.0	0.0	0.0	1.0
Spindle	②	0.0	-0.2	0.0	0.0	-1.2	0.0	0.0	0.0	-0.2

		REV2								
Body	Point	P			Q			R		
		x'	y'	z'	x'	y'	z'	x'	y'	z'
Spindle	②	0.0	0.0	0.0	0.0	0.0	1.0	1.0	0.0	0.0
Ball 1	③	-0.16	0.0	0.0	-0.16	0.0	1.0	1.0	0.0	0.0

		REV3								
Body	Point	P			Q			R		
		x'	y'	z'	x'	y'	z'	x'	y'	z'
Spindle	②	0.0	0.0	0.0	0.0	0.0	0.0	1.0	1.0	0.0
Ball 2	④	0.16	0.0	0.0	0.16	0.0	1.0	1.0	0.0	0.0

TABLE 12.6.4 Data for Translational Joint

Body	Point	P			Q			R		
		x'	y'	z'	x'	y'	z'	x'	y'	z'
Spindle	②	0.0	0.0	0.0	1.0	0.0	0.0	0.0	0.0	1.0
Collar	⑤	0.0	0.0	0.0	1.0	0.0	0.0	0.0	0.0	1.0

TABLE 12.6.5 Data for Translational Spring-Damper-Actuator

Body	Point	P			Q			R		
		x'	y'	z'	x'	y'	z'	x'	y'	z'
Spindle	②	0.0	0.0	0.0	0.0	0.0	1.0	1.0	0.0	0.0
Collar	⑤	0.0	0.0	0.0	0.0	0.0	1.0	1.0	0.0	0.0

Spring constant, $k = 1000, 2000, 3000 \text{ N/m}$
 Damping rate, $c = 30 \text{ kg/s}$
 Free length of spring, $L_0 = 0.15 \text{ m}$

12.6.2 Steady-State Analysis

Based on the configuration of Fig. 12.6.1, steady-state motion is determined by selecting the desired angular speed ω of the spindle. Using dynamic force balancing and $\omega = 11.0174 \text{ rad/s}$, at a slope of 45° of the ball arms, the collar is to be stationary. Table 12.6.6 defines the steady-state configuration of the system.

TABLE 12.6.6 Data for Steady-State Configuration

Body	x	y	z	e_0	e_1	e_2	e_3
Ground	①	0.0	0.0	1.0	0.0	0.0	0.0
Spindle	②	0.0	0.2	0.0	1.0	0.0	0.0
Ball 1	③	0.11314	0.08686	0.0	0.9239	0.0	0.0
Ball 2	④	-0.11314	0.08686	0.0	0.9239	0.0	0.0
Collar	⑤	0.0	0.05	0.0	1.0	0.0	0.0

12.6.3 External Torque

An external torque T_e due to the load driven by the shaft is applied to the spindle, as shown in Fig. 12.6.2. To compensate for this torque, which tends to reduce shaft speed, the torque T_s applied to the spindle by the engine as a result of fuel fed by a collar height variation $\Delta\ell$ is modeled as

$$T_s = C \times \Delta\ell \quad (12.6.1)$$

where C is the torque generated due to fuel fed to the engine by a unit $\Delta\ell$, which is spring deformation (vertical movement of collar). Three different values $C = 7500, 12,500, \text{ and } 17,500$, corresponding to increasing engine power, are used to study the system dynamic response.

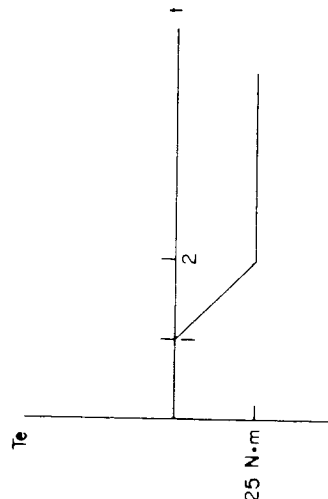


Figure 12.6.2 External torque on spindle.

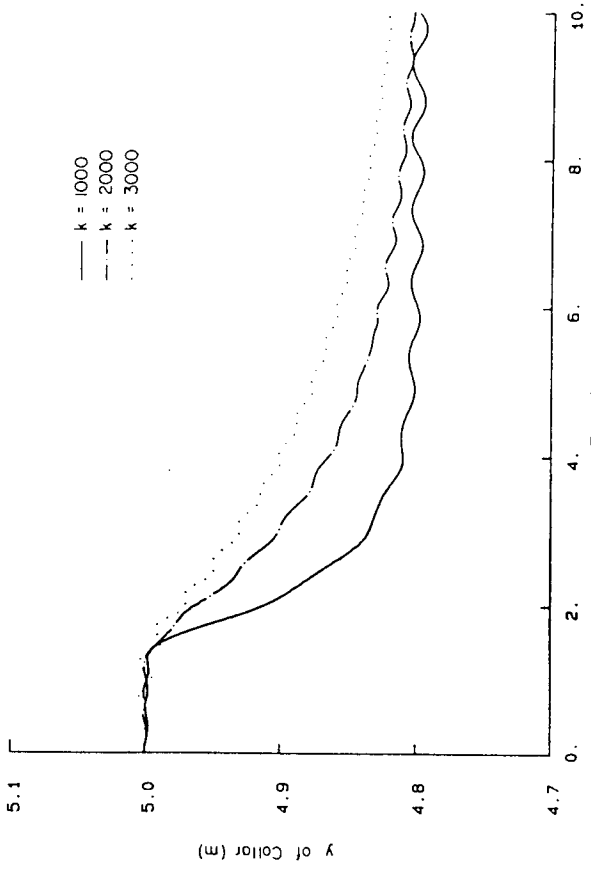


Figure 12.6.3 y of collar versus time ($C = 12,500$).

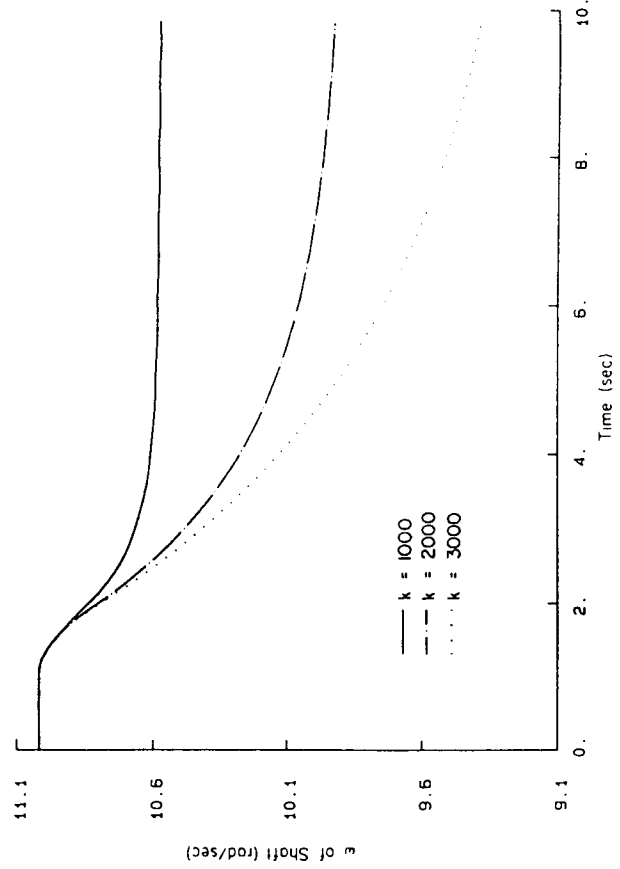


Figure 12.6.4 ω of shaft versus time ($C = 12,500$).

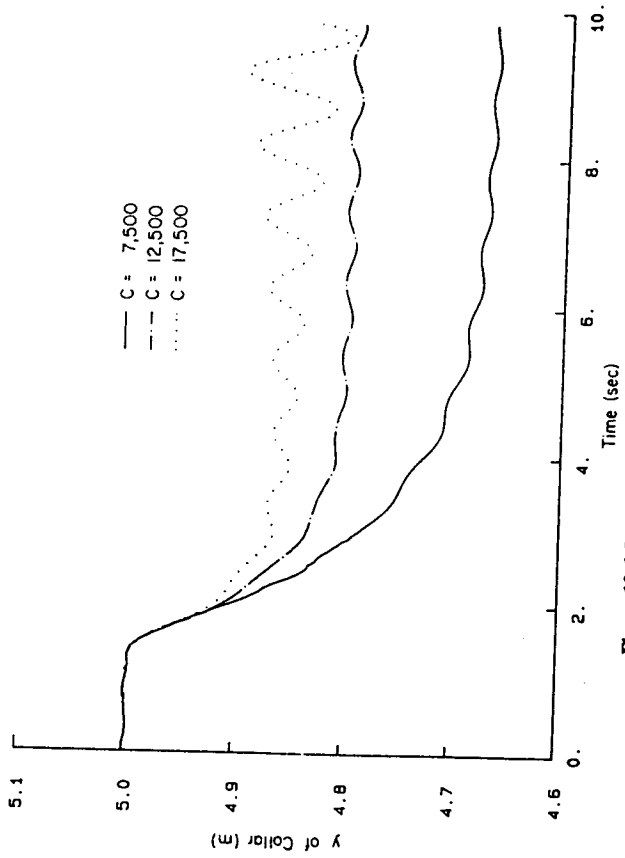


Figure 12.6.5 y of collar versus time ($k = 1000$).

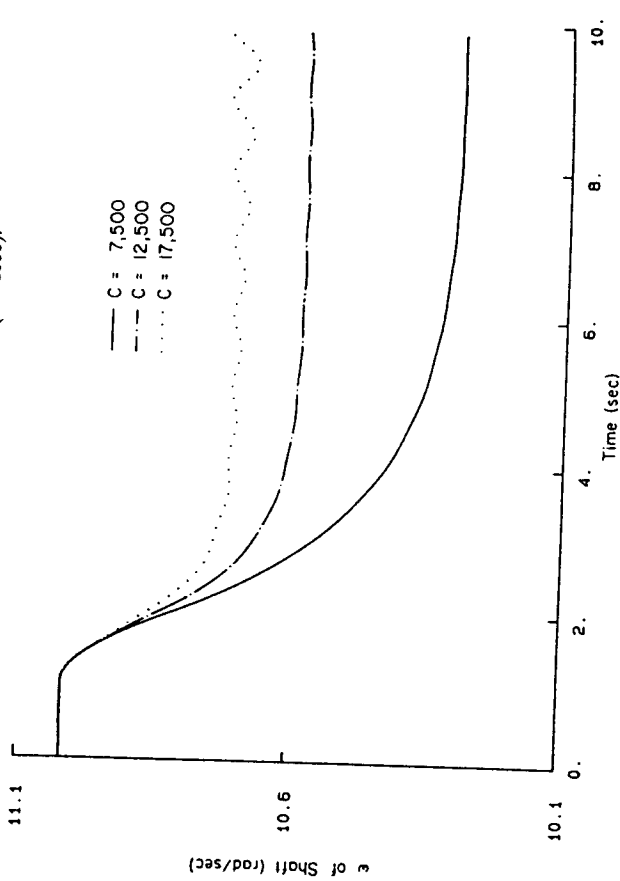


Figure 12.6.6 ω of shaft versus time ($k = 1000$).

12.6.4 Dynamic Analysis

Starting with the given steady-state configuration and $C = 12,500$, dynamic simulations are performed for three different values of the spring rate k . Figures 12.6.3 and 12.6.4 show variations in the vertical position of the collar and angular speed of the shaft. Note that the stiffer the spring, the longer the transition time from the initial steady state to another steady state.

Dynamic responses are plotted in Figs. 12.6.5 and 12.6.6 for different torque feedback rates, with $k = 1000$. As the feedback rate becomes larger (more powerful engine), the system may become unstable.

PROBLEMS

DADS Projects

- 12.1** Using the three body model of DADS Project 10.1, repeat the dynamic analysis of the spatial slider-crank mechanism presented in Section 12.2. Analyze the differences in required driving torque and dynamic response from those presented in Sections 12.2.2 and 12.2.3, due to the elimination of connecting rod inertias.
- 12.2** Using the data in Section 12.4 and the following inertial properties of each connecting rod, carry out a dynamic analysis of the fifteen-body air compressor model of DADS Project 10.2

Mass	$I_{x'x'}$	$I_{y'y'}$	$I_{z'z'}$	$I_{x'y'}$	$I_{y'z'}$	$I_{z'x'}$
0.5	0.04	0.04	0.001	0.0	0.0	0.0

where the center of mass of the connecting rod is at its midpoint and its z' axis is along the rod.

Evaluate the differences in driving torque, reaction forces, and dynamic response from those presented in Section 12.4, due to the inertias of the connecting rods.

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4.2 ISO-Spurwechsel

Der in ISO/TR 3888 [87] definierte doppelte Spurwechsel, auch ISO-Spurwechsel oder -Wedeltest genannt, stellt eine sehr realitätsnahe Closed-Loop-Fahraufgabe für das Zusammenwirken Fahrer-Fahrzeug-Verkehr dar. Er simuliert einen Überholvorgang bzw. das schnelle Ausweichen vor einem Hindernis innerhalb einer vorgegebenen Fahrspur, **Bild 4.24**. Pkw, für die dieser Versuch konzipiert wurde, können die Spurgassen mit bis über 100 km/h durchfahren.

Der vorhandene Fahrereinfluss und dessen teilweise Rückwirkungen auf die ermittelten Größen haben bisher noch keine standardisierten Kennwerte ergeben, weshalb dieses Fahrmanöver nur in einem "Technischen Report" (TR) [ISO/TR 3888, 87] beschrieben ist. Trotzdem wird er als etabliertes Verfahren der Fahrverhaltensuntersuchungen gesehen, entweder als vergleichender Fahrleistungstest - Bestimmung der maximalen Durchfahrungszeit - oder zur subjektiven Beurteilung [Zomotor, 137].

In jüngerer Zeit, in der die Schnittstelle Mensch-Fahrzeug vermehrt erforscht wird, wird der ISO-Spurwechsel zur Ermittlung des Fahrer-Fahrzeug-System beschreibender vergleichender Kenngrößen, die sowohl bezüglich des objektiven Fahrverhaltens als auch des subjektiven Fahrerindrucks korrelieren, in Fahrversuchen sowie Simulationsrechnungen mit Fahrermodellen sehr häufig angewendet [Dibbern et al., 13 u. 14; Laermann et al., 64; Kudritzki, 61; Riedel et al., 102 u. 103], auch bei Lkw und Lastzugkombinationen [Käppler et al., 56; Köfalvi, 58; Elink Schuurman et al., 22]. Allgemein anerkannte Kennwerte wurden daraus bisher noch nicht genormt.

4.2.1 Versuchsdurchführung

Der doppelte Fahrspurwechsel wurde entsprechend den Maßen in ISO/TR 3888 [87] durch Pylone markiert, **Bild 4.24**.

Der Versuch wurde mit gestufter, jeweils konstanter Fahrgeschwindigkeit ab 20 oder 30 km/h bis zur maximalen Fahrgeschwindigkeit - außer LKW - mit mindestens 3 Durchgängen pro Parameter gefahren. Die Verläufe einiger Meßgrößen zeigt beispielsweise **Bild 4.24**.

In der Regel wurden die Versuche mit Fahrer W durchgeführt. Der Fahrereinfluss wird anhand der Fahrzeuge AS4 und LKW dargestellt.

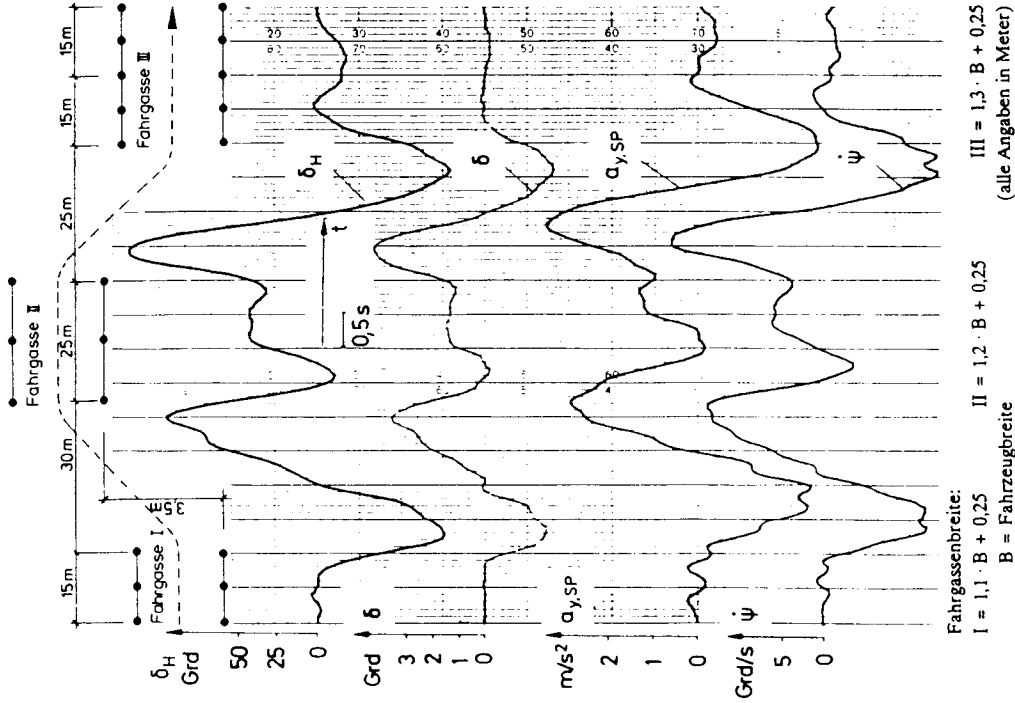


Bild 4.24: Meßgrößenverläufe beim ISO-Spurwechsel AS2, $v = 50 \text{ km/h}$ [Betzler, 4]

Zur Aussage der vom Fahrer aufzuwendenden Lenkaktivität bei Einhaltung der Fahrspassen wird das "Lenkleistungsmaß" [Betzler, 4] herangezogen. Das Lenkleistungsmaß ist der über die Fahrstrecke gemittelte Lenkaufwand, s. Kap. 3.1.1, dividiert durch die Fahrzeit.

$$P_{\delta,H} = \frac{1}{t} \cdot \delta_{HK}$$