Frequency domain approach for the design of heavy-duty vehicle speed controllers

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Abstract: The large parameter variations in heavy-duty trucks make it difficult to find appropriate settings for fixed structure speed controllers. Satisfying the design specifications with experience based trial-and-error control design methods is very time-consuming. This paper discusses the modelling and model verification of a vehicular driveline and the model based, systematic design and tuning of speed controllers, using frequency domain techniques. Experiments with a real truck show that the performance and robustness of the closed loop system is within the specifications. The focus is on the cruise control, but the approach is also applicable to other speed controllers.

Keywords: closed-loop identification; speed control; control design; PI controllers.


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1 Introduction

Heavy-duty vehicles like truck-trailer combinations are equipped with often more than 50 control systems, e.g., for braking and for stability. Approximately ten of these controllers serve to control the vehicle and engine speed. Well known examples are the cruise controller and the engine and vehicle speed limiters. Commonly, these controller are of the PID-type, possibly extended with filters. They have to be tuned to satisfy a priori specified driveability performance and stability requirements. Currently, tuning is most often done by experience-based trial-and-error. This has the disadvantage that stability or performance robustness for other vehicle configurations can not be guaranteed unless a large number of experiments is carried out. Other disadvantages are that no insight is obtained in possible improvements and that it is very time consuming.

To overcome these disadvantages, it is necessary to adopt another tuning method. Very little can be found in literature on tuning PID controllers for automotive applications. In practice, often Ziegler-Nichols related methods (Ziegler and Nichols, 1942) are used. A recent example can be found in Olsson (2004) where these methods are used to automatise engine speed control tuning. These methods were developed for

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Frans Veldpaus received his Master Degree and Doctor Degree from the Eindhoven University of Technology (TU/e) in the Netherlands in 1966 and 1973 respectively. Since 1966, he works in the Department of Mechanical Engineering of the TU/e, first in the area of structural dynamics and solid mechanics and next in the field of control of mechanical systems. His main research activities concern modelling and control of automotive systems, i.e., of active wheel suspensions and power trains in general and of automotive components like clutches and continuously variable transmissions in particular.
applications in the process industry. Their main advantages are that they are simple to use and that no explicit model knowledge is required. However, they implicitly assume that the underlying system is more than critically damped. Therefore they are not suitable for application to vehicular drivelines since these generally are weakly damped and can show strong resonances. The frequency and magnitude of these resonances vary largely with vehicle mass and gear ratio. When these dynamics are neglected, the performance of the closed-loop system can deteriorate and stability can be hampered. Finally, using Ziegler-Nichols related methods it is not possible to tune the system to satisfy predefined performance specifications, and little can be said about the robustness of the resulting closed-loop system.

The problems related to mechanical resonances are also encountered in motion control of mechanical systems like robots. In this field, it is common to use frequency domain techniques for controller tuning (Steinbuch and Norg, 1998). These techniques can only be used if the relevant behaviour of the system can be described with acceptable accuracy by a linear model. For these systems, they offer the possibility to tune for performance and stability robustness. The required models are obtained by applying first principles of physics, by experimental identification or by a combination of these methods. This paper discusses a tuning approach using a combination of these methods: a nonlinear model is derived, linearised in one operating point and validated in experiments. Next, frequency domain methods are applied for the design and tuning of speed controllers. The main focus will be on cruise control tuning, but the proposed approach is also applicable to other speed controllers.

The remainder of this paper is organised as follows. In Section 2, first a nonlinear model for a heavy-duty vehicle is derived and linearised around a stationary operating point. Step response measurements and frequency response function measurements on a real truck are used to determine the parameters of this linear model and to validate it. The controller structure is given in Section 3. There also the approach to tune the controller, using a loop shaping technique (Franklin and Powell, 2002), is discussed. In Section 4, experimental results are given. Finally, conclusions and recommendations are given in Section 5.

2 Modelling and identification

2.1 System description

Figure 1 shows an overview of the driveline of a truck. The torque at the output shaft of the engine is transmitted via the clutch to the gearbox, reduced by the gearbox and transmitted to the final reduction and the differential by the propulsion shaft. Two drive shafts transmit the torque from the differential to the rear wheels. The weight of this truck can vary between 7 tonnes (without trailer) and 40 tonnes (with a fully loaded trailer).

The to-be-controlled quantity is the vehicle speed $v$. Here, the emphasis is on cruise control in straightforward driving on horizontal roads. The cruise control has to operate for vehicle speeds between 30 km/h and 90 km/h. Neither engine braking nor braking by the friction brakes or the retarder are considered.
Figure 1  Overview of the truck without trailer

The available measurement for the cruise controller is the output of an inductive cog wheel sensor at the output shaft of the gearbox. This output is transmitted to the controller via the CAN-bus of the truck. The vehicle speed, estimated from the output signal of this sensor, is seen as the vehicle speed $v$.

2.2 The nonlinear vehicle model

Figure 2 gives a schematic representation of the driveline. It is assumed that the clutch is and remains closed whenever the cruise controller is active and that all deformations in the engine, the clutch, the gearbox and the connecting shafts between these parts may be neglected. The selected transmission ratio of the gearbox is denoted by $r_g$ so $\omega_g = r_g \omega_e$, where $\omega_e$ and $r_g$ are the angular velocities of the engine shaft and the output shaft of the gearbox, respectively. It is assumed that the propulsion shaft (between the gearbox and the differential) and the drive shafts (between the differential and the rear wheels) behave as linear, deformable bodies. Because only straight-on driving is considered here, the two drive shafts may be combined into one equivalent shaft. The stiffness of the propulsion shaft is large compared to the stiffness of the equivalent drive shaft. Besides, the relevant moment of inertia of the combination of the final drive and the differential is very small, meaning that the stiffness and damping of the propulsion shaft and the drive shaft may be combined into one linear torsion spring with stiffness $k_{eq}$ in parallel with a viscous damper with damper constant $b_{eq}$, taking into account the transmission ratio $r_f$ of the differential. The angular velocities of the output shaft of the differential and of the wheels are denoted by $\omega_d$ and $\omega_w$, so $\omega_w = r_f \omega_d$, where $r_f = r_g r_d$ is the total ratio from engine speed $\omega_e$ to drive shaft input speed $\omega_d$. The torsion $\varepsilon$ of the equivalent drive shaft follows from $\dot{\varepsilon} = \omega_d - \omega_e$. 
The rotational kinetic energy \( U_d \) of the subsystem, consisting of the engine, clutch, gearbox, differential and their connecting shafts, is given by
\[
U_d = \frac{1}{2} J_d \omega_d^2,
\]
where the equivalent moment of inertia \( J_d \) of this subsystem is a quadratic function of the selected gear ratio. The torques on this subsystem are the engine torque \( T_e \), the torque \( T_{eq} \) in the equivalent drive shaft and a loss torque \( L_d \). The virtual work \( \delta A \) of these torques for a variation \( \delta \phi_d \) of the rotation of the outgoing shaft of the differential is given by
\[
\delta A = (T_e - L_d) \frac{\delta \phi_d}{r_d} - T_{eq} \delta \phi_d.
\]
It is assumed that in stationary as well as in transient situations, the engine torque can be determined from the stationary engine map and is given by
\[
T_e = T_{\text{map}}(\omega_e, \Phi) \quad (1)
\]
where \( \Phi \) is the fuel mass flow. In control terms, this quantity is the output of the cruise controller and the input \( u \) of the to-be-controlled system, so \( u \equiv \Phi \). As mentioned before, the equivalent drive shaft is modelled as a linear system with stiffness \( k_{eq} \) in parallel to a damper \( b_{eq} \), so
\[
T_{eq} = k_{eq} \dot{\varepsilon} + b_{eq} \ddot{\varepsilon}; \quad \dot{\varepsilon} = \omega_d - \omega_e. \quad (2)
\]
The loss torque \( L_d \) is more difficult to model. Based on a large series of experiments with gearboxes (van Dongen, 1983) it is assumed that \( L_d \) depends on the ingoing torque \( T_e \) and the angular velocity \( \omega_e \) of the gearbox and is given by
\[
L_d = (1 - \eta_d) T_e + r_d^2 b_d \omega_e. \quad (3)
\]
In general, both the efficiency \( \eta_d \) and the damper coefficient \( b_d \) will depend on the selected gear ratio. This ratio is constant if the cruise controller is active, so this dependency is not a problem here.
Application of the equations of Lagrange on the given relations for the kinetic energy $U_d$, the virtual work $\delta A$ and the torques $T_{eq}$ and $L_d$ results in the equation of motion for the considered subsystem, i.e.,

$$J_d \omega_d = \eta_d T_{eq} - (b_d + b_{eq}) \omega_d + b_{eq} \omega_n - k_d \varepsilon.$$

(4)

The torques on the driving rear wheels are the torque $T_{eq}$ in the equivalent drive shaft, a loss torque $L_{w} = b_{w} \omega_{w}$, a torque $R_{w} F_{w}$ due to the horizontal, forward force $F_{w}$ in the tyre-road contact and a torque $t N_{w}$ due to the normal force $N_{w}$ in the tyre-road contact. Here $R_{w}$ is the radius of the rear wheels. It is assumed that the variations of this radius may be neglected. In general the distance $t$ between the point of application of the normal force $N_{w}$ and the vertical through the wheel centre is small enough to neglect the torque $t N_{w}$. Therefore, the equation of motion for the rear wheels becomes

$$J_{w} \omega_{w} = b_{w} \omega_{w} - (b_{w} + b_{eq}) \omega_{w} + k_{eq} \varepsilon - R_{w} F_{w}.$$

(5)

where $J_{w}$ is the total moment of inertia of the rear wheels.

The vehicle is driven by the horizontal tyre force $F_{w}$. The other forces of interest on the vehicle are the air drag force $F_{\text{air}}$, the rolling resistance force $F_{rr}$ and possibly a disturbance force $F_{\text{dist}}$ due to, for instance, road slope, wind gusts and road unevenness. It is assumed that the disturbance force is constant, that the air drag force is proportional to the square of the vehicle speed $v$, so $F_{\text{air}} = c_{a} v^{2}$ with a constant drag factor $c_{a}$, and that the rolling resistance force $F_{rr}$ is proportional to the total weight $mg$ of the vehicle, so $F_{rr} = c_{r} mg$ where $m$ is the total mass and $c_{r}$ is a constant. In the sequel, the disturbance force is combined with the rolling resistance force into one constant external force $F_{0}$. With these forces, the equation of motion for the vehicle can be shown to be

$$m \ddot{v} = F_{w} - c_{a} v^{2} - F_{0}; \quad F_{0} = c_{r} mg + F_{\text{dist}}.$$

(6)

A main problem in deriving a vehicle model for the design of speed controllers is raised by the constitutive equation for the tyre force $F_{w}$. It is assumed that (Pacejka, 2002):

$$F_{w} = \mu(\chi) N_{w}.$$

(7)

where $N_{w}$ is the normal force from the road on the tyre and $\mu$ is a nonlinear function of the wheel slip $\chi$, defined by Mitschke (1995):

$$\chi = \frac{R_{w} \omega_{w}}{v} - 1.$$

(8)

The normal force is not constant. Due to the longitudinal load transfer this force depends linearly on the vehicle acceleration $\dot{v}$, so

$$N_{w} = N_{w0} - \lambda m \dot{v}.$$ 

(9)

where $N_{w0}$ is the normal force if the vehicle is at rest. The load transfer coefficient $\lambda$ only depends on the geometry and mass distribution of the truck and the trailer and is constant as long as the configuration and loading condition of the vehicle are not changed.
Elimination of $N_w$ yields a relation for the tyre force $F_w$ as a function of $\chi$, $v$ and $F_{\text{dist}}$:

$$\{1 + \lambda\mu(\chi)\}F_w = \mu(\chi)N_w + \lambda\mu(\chi) \cdot (c_m g + c_n v^2 + F_{\text{dist}}).$$  \hspace{1cm} (10)

### 2.3 Linearisation of the vehicle model

The obtained nonlinear model is too complex to serve as a cruise control design model. However, since the controller has to realise a constant cruise speed $v_0$, it is obvious to linearise the model around a stationary operating point, characterised by this speed, the constant external force $F_0$ and the selected gear. Substitution of $\dot{v} = 0$ and $\dot{\omega}_e = \dot{\omega}_d = \dot{\chi} = \dot{\epsilon} = 0$ yields the following relations for the stationary quantities

$$\mu(\chi_0)N_0 = F_{\text{w},0} = F_0 + c_n v_0^2; \quad R_{w,0}\omega_{\text{w},0} = (1 + \chi_0) v_0$$  \hspace{1cm} (11)

$$k_{w,0} = R_{w,0}F_{\text{w},0} + b_n\omega_{\text{w},0}; \quad \omega_{d,0} = r_{\omega} \omega_{e,0} = \omega_{\text{w},0} + \dot{\epsilon}_0$$  \hspace{1cm} (12)

$$\eta_{\theta} T_{e,0} = r_{e} (R_{\text{w},0}F_{\text{w},0} + b_n\omega_{\text{w},0} + b_d\omega_{d,0}); \quad T_{\text{map}}(\omega_{e,0}, \Phi_0) = T_{e,0}. \hspace{1cm} (13)

It is assumed that the tyre function $\mu$ is differentiable with smooth first derivative $\gamma$, i.e.,

$$\gamma(\chi) = \frac{d\mu(\chi)}{d\chi}. \hspace{1cm} (14)$$

Then it is possible to determine $F_{\text{w},0}$, $\chi_0$ and $\omega_{e,0}$ from equation (11), $\omega_{d,0}$ from equation (12) and $T_{e,0}$ from equation (13). Furthermore, it is assumed that the engine function $T_{\text{map}}$ is differentiable with smooth first partial derivatives $-\Gamma$ and $\Lambda$, i.e.,

$$\Gamma(\omega_e, \Phi) = -\frac{\partial}{\partial \omega_e} T_{\text{map}}(\omega_e, \Phi); \quad \Lambda(\omega_e, \Phi) = \frac{\partial}{\partial \Phi} T_{\text{map}}(\omega_e, \Phi). \hspace{1cm} (15)$$

Then the stationary fuel mass flow $\Phi_0$, i.e., the stationary input $u_0 \equiv \Phi_0$ can be solved from equation (13).

Denoting deviations from the stationary values by a prefix $\delta$, so $\delta \Phi = \Phi - \Phi_0$, etc., it is seen that

$$\delta T_e = -\Gamma_0 \delta \omega_e + \lambda_0 \delta \Phi = \frac{\Gamma_0}{r_{\delta}} \delta \omega_d + \lambda_0 \delta \Phi$$

$$\delta \chi = \frac{R_{\text{w}}}{v_0} \delta \omega_d - \frac{1 + \chi_0}{v_0} \delta v; \quad \delta F_w = b_{n,0} R_{\text{w}} \delta \omega_d - b_{d,0} \delta v$$

where the damping-like quantities $b_{n,0}$ and $b_{d,0}$ are given by

$$b_{n,0} = \frac{\gamma_0}{1 + \lambda \mu_0} \cdot \frac{N_w}{v_0}; \quad b_{d,0} = (1 + \chi_0) b_{n,0} - 2 \frac{\lambda \mu_0}{1 + \lambda \mu_0} c_n v_0.$$
With these results, a minimal state space description of the linearised model is given by

\[
\delta \dot{x} = A \delta x + B \delta u \tag{16}
\]

\[
\delta y = C \delta x \tag{17}
\]

where \( u \equiv \Phi \) is the input (the fuel mass flow) and \( y \equiv \omega_d \) is the measured quantity (the angular velocity of the outgoing shaft of the differential). The entries of the state \( x \) are the deformation \( \varepsilon \), the angular speed at the differential \( \omega_d \), the wheel angular speed \( \omega_w \) and the vehicle speed \( v \), so

\[
x = [\varepsilon \quad \omega_d \quad \omega_w \quad v]^T. \tag{18}
\]

The system matrix \( A \), input matrix \( B \) and output matrix \( C \) follow from

\[
A = \begin{bmatrix}
0 & 1 & -1 & 0 \\
-\frac{k_x}{\tau} & -\frac{(k_x + k_\mu)^2 + \eta J_d}{\tau^2} & \frac{k_x}{\tau} & 0 \\
\frac{k_x}{\tau} & \frac{k_x}{\tau} & -\frac{k_x + k_\mu}{\tau} & \frac{R \omega_d}{\tau} \\
0 & 0 & \frac{R \omega_d}{m} & -\frac{k_x + 2k_\mu}{m} \\
\end{bmatrix} \tag{19}
\]

\[
B = \frac{J_d}{\tau} \quad C = \begin{bmatrix} 0 & 1 & 0 \end{bmatrix}. \tag{20}
\]

### 2.4 Influence parameters

Figure 3 shows the Bode plot of the transfer function from fuel mass flow to vehicle speed in a typical operating point of the vehicle. This plot is used to illustrate the influence of the most important model parameters on the essential characteristics of the system. The frequency range of interest is split in four regions: a very low frequency region, in which the transfer function is constant, a low frequency region in which the rigid body mode of the truck dominates the dynamics, a high frequency region in which a resonance occurs, and a very high frequency region in which the transfer is governed by the dynamics of the engine inertia. The most important characteristic for control purposes is the resonance peak, which is characterised by a zero/pole pair. The selected gear and vehicle mass mainly determine the pole location. The resonance frequency depends on the selected gear and increases with increasing shaft stiffness and with decreasing moment of inertia \( J_d \). The ‘damping’ \( b_{t,0} \) is the main parameter for the damping of the anti-resonance.
2.5 Modelling, based on step response measurements

The approximate realisation theory (de Schutter, 2000) is used to estimate a linear model from responses $\delta y(t) = \delta \omega(t)$ to small, stepwise perturbations $\delta u(t) = \Delta u \epsilon(t - t_0)$ of the fuel mass flow. Here, $\Delta u$ is the step amplitude whereas $\epsilon(t - t_0) = 0$ for $t < t_0$ and $\epsilon(t - t_0) = 1$ for $t > t_0$. For a linear, constant system, the shifted, normalised step response $\delta y_{\text{norm}}(t) = \delta y(t - t_0)/\Delta u$ should be the same for all sufficiently small amplitudes. Because stepwise perturbations excite the lower frequencies of a system more than the higher frequencies, it is to be expected that the obtained model will be less accurate for high-frequency dynamics.

All measurements are performed on a truck with a 12 gear automated manual transmission and with a dSPACE MicroAutoBox system. This system communicates via the CAN-bus with the electronic control unit of the engine.

The controller design models should cover the complete operating range of interest. Therefore, the step response measurements have been performed at various cruise speeds between the minimum and maximum speed and, at each of these speeds, for two gear ratios in which the desired cruise speed can be attained. The vehicle weight varies between the two extremes, i.e., between 7 tonnes and 40 tonnes. Three different step amplitudes have been used for each measurement in order to check linearity of the system. As an example, Figure 4 shows typical normalised responses for a 40 tonne truck for input step amplitudes of 10%, 20% and 30% of the maximum fuel mass flow $\Phi_{\text{max}}$.

From this figure it can be concluded that after 1 sec the slopes of the normalised responses are approximately the same. However, the amplitude of the initial oscillations seems to decrease for higher step amplitudes. The reason is that backlash in the driveline becomes important for the considered large amplitudes. By zooming in on the initial part of the response, it is seen that a time delay of approximately 70 ms occurs in all measurements. This is partly caused by the sampling on the CAN-bus and, to a larger extent, by a delay in the torque production in the engine.
The measurement noise (mainly harmonic, due to sensor misalignment) is fairly large and has a great influence on the linear model, obtained with the approximate realisation method. Therefore, the measurements are filtered offline with a low pass filter without phase loss. A low cut-off frequency will lead to loss of relevant data, whereas the noise will not be filtered effectively with a high cut-off frequency. The time span of the response measurements also has a large influence on the resulting model. A longer time span will result in a better description of the low-frequency behaviour but also in a less accurate description of the high frequency behaviour. Another factor of importance is the order of the to-be-estimated model. Here, the filter cut-off frequency, the time span of the measurements and the model order are compromised in such a way that the resulting model is capable of capturing the low frequency dynamics, which are dominated by the rigid body mode of the truck.

2.6 Modelling, based on frequency response measurements

For frequency domain identification, sensitivity measurements (Ljung, 1999) are performed. During the measurements, a weakly tuned PI-controller is used to keep the vehicle speed near the desired speed \( r \equiv v_0 \), see Figure 5. The perturbation \( d \), i.e., the excitation of the plant, contains the frequencies that are supposed to be relevant for the identification. The transfer function from \( d \) to the disturbed plant input \( u \), i.e., the sensitivity function \( S \), is estimated from

\[
S(s) = \frac{P_{ds}(s)}{P_{dd}(s)}
\]

(21)

where \( P_{ds} \) is the cross-spectral density between \( d \) and \( u \) and \( P_{dd} \) is the autospectrum of \( d \). Since the transfer function \( C \) of the controller is known it is possible to reconstruct an estimate \( \hat{P} \) for \( P \) from

\[
\hat{P}(s) = \frac{1 - \hat{S}(s)}{C(s)\hat{S}(s)}
\]

(22)
Experiments have been performed with low-pass filtered white noise inputs $d$ containing frequencies up to 40 Hz. The measurements have been performed in the same operating points as in the previous subsection. The Frequency Response Function (FRF) is calculated using an FFT algorithm with 1024 FFT points. A Hanning window of 1024 points is used with 512 points overlap between the FFT blocks. With a measurement time of 300 s and a sample frequency of 100 Hz, the lowest estimated frequency is approximately 0.1 Hz.

2.7 Model comparison and validation

The step response measurements and the FRF measurements are also used to identify the parameters of the linearised vehicle model. Figure 6 shows the results of the linearised model, the step response based model and the FRF based model for a 40 tonne truck in a typical operating point. A low-pass filter and a delay are added to the linearised model to fit the measured FRF for high frequencies. The need for this low-pass filter can be explained by the engine dynamics. Based on Figure 6 and on the earlier remarks, the following observations, valid for the entire operating range, can be made:

- The tuned linearised model predicts the behaviour of the truck very well, so the model structure seems to be correct.
- The step response based models are reliable for low frequencies but not for high frequencies.
- The models based on the FRF measurements are reliable for high frequencies, but not for low frequencies. This is indicated also by the coherence functions (not given here), which show that the FRF estimates are reliable for frequencies higher than 0.5 Hz. It should be noted that there are just a few data points for low frequencies, i.e., at 0.1 Hz and 0.2 Hz.
- The time delay is clearly visible in the FRF measurements as an increasing phase lag. By plotting the phase on a linear frequency scale the time delay is estimated at 60 ms, i.e., approximately the same as observed from the step responses.

For the design and tuning of the cruise controller, a combination of the step response based models and the FRF estimates is constructed: the step response based models are used for the low frequency region, whereas the FRF estimates are used for the high frequency region. The Bode plots of the models for all operating points are given in Figure 7. The physical model is very useful to evaluate the effect of configuration changes.
3 Controller tuning

3.1 Controller structure

All speed controllers in the truck that is used for the experiments, are PI-controllers in series with a first order low-pass filter:

$$C(s) = \left( K_p + \frac{K_i}{s} \right) \frac{1}{\tau s + 1}$$

(23)

where $K_p$ is the proportional gain, $K_i$ the integral gain and $\tau$ the time constant of the filter.
Additional features are an integral anti-windup and the possibility to use two sets of PI-gains, called low gain and high gain. The value for $\tau$ is the same in both sets.

Here, the low gain will be tuned to achieve stability robustness for the whole operating range at an acceptable performance, whereas the high gain will be used to increase the performance in operating points with high vehicle weights and high vehicle speeds.

### 3.2 Performance and stability specifications

The performance specifications are related to the driveability of the vehicle. They are expressed in terms of step response related quantities of Figure 8:

- the overshoot $M_p$ is the maximum overshoot of the final value, divided by this final value
- the rise time $t_r$ is the time it takes the system to reach its final value for the first time.

**Figure 8** Definition of overshoot $M_p$ and rise time $t_r$.

In Franklin and Powell (2002) these quantities are related to the natural frequency $\omega_n$ and the specific damping $\zeta$ for an undercritically damped ($\zeta < 1$) second order system with transfer function

$$P(s) = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2}.$$  \hspace{1cm} (24)

For this system it can be shown that (Franklin and Powell, 2002):

$$t_r = \frac{1.8}{\omega_n}; \quad M_p = e^{(-\zeta\sqrt{1-\zeta})}.$$  \hspace{1cm} (25)

These relations can be used as a rough approximation for other than second order systems. After replacing $\omega_n$ by the desired bandwidth $\omega_{bw}$, the relations in equation (25) will be used to specify the performance criteria for the closed loop system.
The classical quantities to assess stability robustness are the gain margin $GM$ and the phase margin $PM$. The gain margin is the factor by which the open loop transfer function $C(s)P(s)$ can be multiplied before the Nyquist curve of this function intersects the point $(-1, 0)$. The phase margin is the amount by which the phase of this function can be decreased before the Nyquist curve intersects the point $(-1, 0)$. For $PM \leq 70$ deg, the relative damping $\zeta$ may be approximated by (Franklin and Powell, 2002):

$$\zeta \approx \frac{PM}{100}$$

Equation (26)

with $PM$ in degrees. This relation is derived for second order systems with a transfer function as given by equation (24) and with unity feedback, i.e., $C(s) = 1$. Nevertheless, it is used here in combination with equation (25) to determine a desired $PM_{des}$ from a specified value $MP$ for the allowable overshoot. The desired gain margin $GM_{des}$ is initially set to 2, as recommended in Aström and Hägglund (1995).

### 3.3 Tuning approach

The given specifications have to be achieved for all operating points. This is done by first tuning the low gain of the controller, such that the specifications are satisfied for all operating points. After that, the high gain is tuned for a smaller set of operating points, namely for vehicle speeds above 70 km/h and gears 10–12.

The approach taken here is to first identify a worst-case operating point. Then loop shaping on the linearised model for this point is used to find controller parameters $K_p$ and $\tau$ such that the specifications are satisfied. Next, the value for $K_I$ is maximised since higher values of $K_I$ result in a better load disturbance attenuation (Aström and Hägglund, 1995). This maximisation is done considering all operating points, since using the worst-case point only gives no guarantee that the desired phase margin is achieved for all operating points.

An analysis with the physical model shows that the worst-case operating point is found for the truck with the highest vehicle weight, driving with the lowest vehicle speed in the lowest gear. Here this means a 40 tonnes truck driving at 30 km/h in gear 7. The proportional gain $K_p$ is chosen such that the desired bandwidth is reached for the worst-case, according to:

$$K_p = \frac{1}{|\bar{P}_{\text{const}}(j\omega_{\text{bc}})|}.$$  

Equation (27)

Next, the value for $\tau$ is chosen such that the desired gain margin is achieved for the worst-case model. This guarantees stability for all operating points. Finally, $K_I$ is maximised under the constraint that the desired phase margin is satisfied for all operating points.

The same procedure is taken for the high gain parameter set of the cruise controller, but now with the model of the truck driving 70 km/h in gear 10 as the worst-case.

### 3.4 Tuning results

Figure 9 shows the resulting Nyquist plots for all considered operating points, using the low gain setting of the controller. All open loop transfers satisfy the specified gain and
phase margin. It can be concluded that the outlined approach results in controller settings that achieve the specifications for all operating points.

**Figure 9** Nyquist plots for all operating points and the controller with the low gain setting

![Nyquist plots](image)

**4 Experimental validation**

First, the stability of the worst-case operating point with the low gain setting for the controller is cursorily looked at. Figure 10 shows the response to a tip-in disturbance in which the driver is pressing the accelerator pedal for approximately 0.6 s. The weakly damped oscillation in this response can be caused by, for instance, backlash and/or friction in the driveline. These non-linearities are known to cause limit cycles (Hensen, 2002). Several solutions to this problem have been proposed, including nonlinear control laws or adding a load observer (Lagerberg and Egardt, 2002; Nordin and Gutman, 2002). However, since in this paper the controller structure is fixed, the approach taken here is to reduce the high-frequency control action by increasing the gain margin. To achieve this, the value for $\tau$ is increased. Experimentally, it turned out that this strategy yields good results, shown in Figure 10 by the dashed line.

**Figure 10** Vehicle speed and fuel mass flow after a stepwise disturbance in the fuel input for $GM = 2$ (solid) and increased $GM$ (dashed): (a) fuel injection and (b) tachograph speed

![Vehicle speed and fuel mass flow](image)
The consequence of a higher $GM$ is that the bandwidth has to be lowered to allow integral action. In a test in which the setpoint is suddenly changed from 30 km/h to 40 km/h, the overshoot is within the specifications for all relevant operating points. An example is shown in Figure 11. Due to the large operating range for which the low gain applies, the value for the integral gain is rather low, which results in a rather sluggish settling behaviour. However, for the driver of the truck, this effect is barely noticeable.

Figure 11 Vehicle speed response for a changing setpoint

5 Conclusions and recommendations

The presented modelling methods are complementary in the sense that the step response based method results in reliable models for low frequencies, that the FRF measurements lead to a reliable model for high frequencies and that the linearised model is useful for interpretation of experimental results, evaluation of the influence of parameter changes, robustness analyses, etc.

The presented frequency domain tuning method offers the possibility to tune the speed controllers in heavy-duty trucks in a structured way. The use of stability and performance criteria as tuning parameters is faster and more robust than trial-and-error or Ziegler-Nichols related methods. Experimental results show that the controllers resulting from the tuning method meet the specifications.

Given the structure of the controller, with this method the maximum performance is obtained. If higher performance over all operating points is required then it is recommended to implement controllers with another structure. One could think of controllers which adapt their parameters to the selected gear and/or to (estimates of) the vehicle weight. This offers the opportunity to increase the integral gain and hence to improve the disturbance attenuation.

In future work an effort will be made to automatise the tuning process by means of an optimisation algorithm. This will eliminate the loop shaping by hand.
Frequency domain approach

References


Note

1 The bandwidth is defined here as the 0 dB crossover frequency of the open loop transfer function $C(s)P(s)$. 