

**Modeling and simulation of a driveline with
an automatic gearbox**

**Master's Thesis
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Abstract

In this master's thesis project a dynamical model of a truck's driveline with an automatic transmission is built up and implemented in the modeling language Modelica. The model is built in a modular way to facilitate reuse and make the model more well-arranged. The truck model has been developed from a previously created model that describes the driveline of a truck with manual transmission.

The purpose of this project is to develop a truck model that can be used to simulate speed signals from different parts of a driveline, to enable tests of an automatic gearbox system in a lab instead of tests in an ordinary truck. To fulfill such tests, speed signals from different parts of a truck's driveline have to be generated by the model. In other words, speed signals that are sensed on a real truck by the control units have to be simulated, i.e. the mechanical parts in the driveline of a truck must be replaced by electrical signals that simulate the behavior of the real components. This is done to let the control units believe they receive speed signals generated from a real truck. The problem is to create a model that calculates these speed signals with dynamical behavior to achieve realistic values that agree with the reality.

To be able to run tests of the automatic gearbox system, i.e. achieve automatically performed shifts, the simulated speed signals have to correspond to each other within the shift in process and even between the shift courses. Otherwise the control unit of the gearbox system would believe that something is wrong and degrade its functionality.

The main work has been put into modeling of the engine, the converter and the gearbox. These three driveline parts have been modeled completely without inheriting previously developed components. The engine component describes the behavior of a RQV control and the converter module models two different states, first a fluid coupling and then a lockup clutch. The first-mentioned mode acts in reality hydraulically and is modeled by empirical formulas describing the converter's ability to absorb torque. In the lockup clutch mode the converter provides a direct drive from the engine to the transmission gearing. The gearbox module is modeled by adjusting the gear ratio signal sent from the gearbox management system.

By using the simulated speed signals that are generated by the developed truck model that was built in this project, has it been possible to test the control unit of the automatic gearbox system in the test lab. This means that several tests can be performed in a lab instead of being forced to run the tests in a real truck. This will save a lot of time in the future. The model enables driving simulations from stand still up to the higher speed domains. With the accelerator pedal the engine speed is controlled. When the control unit of the automatic gearbox system reaches an intended gear shifting point, automatic up- or downshift will be performed.

Preface

This master thesis is the last part of my education in electrical engineering at the Royal Institute of Technology and makes my degree therefore completed. The work has been carried out at Scania CV AB, at the RESA (Vehicle Electrical Architecture and Chassis System Software) group belonging to the department of Truck Development. The department is located at Technical Center in Södertälje.

First of all I would like to thank my supervisor at Scania, Johnny Arne, for his guidance and the help of this thesis. Furthermore, I would like to thank the group manager, Peter Madsen, who gave me a hint about this interesting thesis and let me working with this. Another person at Scania I would express my gratitude to is Robert Sjödin for help and hints.

The report is intended for both people at Scania and at KTH.

Notation

Engine

$w_{reference}$	Reference engine speed	[rad/s]
w_c	Crankshaft speed	[rad/s]
T_{engine}	Maximum engine torque	[Nm]
$q_{acc_pedal_pos}$	Accelerator pedal position	[1]
$\Delta w_{present}$	Present engine speed difference	[rpm]
$\Delta w_{allowed}$	Allowed engine speed difference	[rpm]
c	Percent value of maximum available engine torque	[1]
$T_{available_engine}$	Available engine torque	[Nm]

Converter

T_{pump}	Pump torque	[Nm]
$T_{absorbed}$	Absorbed torque	[Nm]
w_{pump}	Pump speed	[rad/s]
$w_{turbine}$	Turbine speed	[rad/s]
SR	Speed ratio	[1]
TR	Torque ratio	[1]
T_c	Transferred torque	[Nm]

Gearbox

T_{g_in}	Input shaft torque of the gearbox	[Nm]
T_{g_out}	Output shaft torque of the gearbox	[Nm]
w_{g_in}	Input shaft speed of the gearbox	[rad/s]
w_{g_out}	Output shaft speed of the gearbox	[rad/s]
n_{gear_ratio}	Adjusted gear ratio	[1]

Retarder

$T_{retarder}$	Applied braking retarder torque	[Nm]
$T_{propeller}$	Propeller shaft torque	[Nm]
$w_{propeller}$	Propeller shaft speed	[rad/s]

Final drive

$T_{final_drive_l}$	Left drive-shaft torque	[Nm]
$T_{final_drive_r}$	Right drive-shaft torque	[Nm]
n_{final_drive}	Gear ratio of the final drive	[1]
$w_{final_drive_l}$	Left drive-shaft speed	[rad/s]
$w_{final_drive_r}$	Right drive-shaft speed	[rad/s]

Brake

F_{b_n}	Normal force	[N]
k	Constant	[1]
p	Brake pressure	[N/m ²]
p_c	Collaring pressure	[N/m ²]
T_b	Braking torque	[Nm]
c_{geo}	Geometric constant	[m]
m	Friction coefficient	[1]
w_w	Wheel shaft speed	[rad/s]

Wheel

F_x	Tire force	[N]
r	Rolling radius	[m]
T_w	Torque acting on the wheel hub	[Nm]
J_w	Wheel and tire inertia	[kgm ²]
w_w	Angular velocity of wheel	[rad/s]
F_z	Normal force	[N]
C_r	Rolling resistance coefficient	[1]
C_{r_iso}	Rolling resistance coefficient	[1]
C_a	Rolling resistance coefficient	[h/(km ²)]
C_b	Rolling resistance coefficient	[h/km]
F_{rr}	Rolling resistance force	[N]

Chassis

F_{slope}	Gravity force due to the road slope	[N]
m	Mass of the vehicle	[kg]
g	Gravity on earth	[m/s ²]
a	Road slope	[rad]
$slope$	Road slope	[%]
F_{ar}	Air resistance force	[N]
c_D	Aerodynamic constant	[1]
A	Front area truck	[m ²]

ρ	Air density	[kg/m ³]
v_d	Difference between vehicle and wind velocity	[m/s]
v	Truck velocity	[m/s]
\dot{v}	Acceleration of the truck	[m/s ²]
F_{fa}	Front axle normal force	[N]
F_{ra}	Rear axle normal force	[N]
n_{axles}	Number of rear axles	[1]
l_{fr}	Axle distance	[m]
l_{fm}	Distance from front axle to mass center point	[m]
h_g	Height to working point of gravity force	[m]
h_a	Height to the working point of air drag	[m]
$F_{normal_front_wheel}$	Front wheel normal force	[N]
$F_{normal_rear_wheel}$	Rear wheel normal force	[N]
n_{front_wheels}	Number of front wheels	[1]
n_{rear_wheels}	Number of rear wheels	[1]

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1 Introduction

1.1 Background

In a previously performed master thesis project at Scania a modular truck model has been created. This model describes the driveline of a truck. The model is built up by submodels, for example by an engine, a clutch, a gearbox, wheels and a chassis. The most complex submodel is the wheel and tire component, which is modeled as flexible to generate a dynamic tire force. The entire model simulates speed signals from different parts of the driveline, which are sent to control units located in the test lab.

To be able to run tests of a truck's driveline in a laboratory instead of in a real truck, some speed signals have to be simulated. The reason to this is that no mechanical components in the driveline, such as engine, converter, gearbox, retarder etc, are implemented in the laboratory in its real shape. Instead the laboratory only consists of sensors, electrical equipment and electronic control units of different subsystems etc. Therefore the mechanical parts in the driveline of a truck must be replaced by electrical signals that simulate the behavior of these real components.

The truck model that is going to be modeled will use parts from the previously developed driveline model. More information about the model can be read in [5]. First and foremost the developed wheel and the tire model will be used in this master thesis project.

The truck industry has continuously been under development. In the beginning the industry was almost totally built up by mechanics. But during the last decades there has been more and more computer technology involved in the trucks. And nowadays there are a lot of large and complex computer systems. The computers are used in the control units that communicate with each other.

The use of computer simulations within the industry is quickly increasing. The simulations are typically used to optimize products and to reduce costs and time in the product development process. Whereas in the past were enough to simulate each subsystems separately, the trend today is an increasing demand for simulations of complex physical systems composed of subsystems from different areas, such as mechanical, electrical, hydraulic and control systems.

In the truck development it becomes more and more important to design and test products by computer simulations before building prototypes. Simulation is also a good help when systems should be tested in different versions. With simulations one can make close predictions of vehicle performance. Simulations can be helpful for tuning parameters and settings and to improve new parts of a vehicle. The main benefits are lower expenses, shorter development progresses, less time needed for running tests and thus hopefully higher quality of the products. Simulations also increase the opportunities to test different functions and versions by very simple changes.

During the last years an increasing numbers of electronic control units have been settled in the Scania trucks. Several of these share information, such as the engine management system, the

gearbox management system and the braking control. The trucks are equipped with a CAN¹-bus that connects all control units of the truck, i.e. the ECUs² communicate with each other through the CAN network. This network protocol is a common standard for serial communication, frequently used in the automotive industry because of its reliability and robustness. The network is divided into three different buses called red, yellow and green bus. The red bus is considered as the most critical and the units connected to the bus are those whose information is of highest priority in the point of driveability, i.e. only systems according to the driveline are connected to this bus. If an error arises on that bus then the truck is put out of operation and the driver has to stop immediately. The yellow bus is applied for important but not critical systems such as the lightning and the instrument cluster. If an error occurs on the bus this cause the driver to visit a workshop. To the green bus are units that handle the driver's comfort connected, such as the audio system and the air condition. A gateway enables data to be transferred between the three buses.

To be able to communicate on the bus, each node connected to the bus needs a bus controller, to interface each node on the bus. The system also needs a communication protocol that is standard for how messages are built up and how these are passed through the bus. The CAN communication works in this following way: one node transmits a message to all other nodes. A message is identified by the information it contains. On the basis of the received identifier each node decides whether it should process the message or not. The identifier also determines the priority the message holds for the access to the bus. The main parts of a message are the identifier field and the data field. More information about CAN can be read in appendix A.

When developing a single control unit, several tests have to be done to verify the functionality of the system. The test can either be done directly in a truck or in a test lab. Running the tests in a computer is more economical and saves a lot of time compared to real tests. Some other advantages of testing in a lab are the larger working space and possibilities to test different functions and versions. Other advantages are that the test environment is repeatable and is not changing from day to day. Further, in a truck serious accidents can occur if some control units are not working properly. A disadvantage of tests performed in a lab can be that the driving situations are only a simulation of a real road manner, which means it is difficult to reach all natural influencing factors.

RESA is the unit of Scania that is responsible for the electrical architecture of the trucks. The group cooperates with other groups about subjects according to different parts of the electrical system and its architecture. RESA is responsible for two types of testing of the electrical systems:

- Integration tests performed in the integration lab.
- Truck tests.

Tests on control units are performed in a laboratory, where the functionality of the networked ECUs is checked so that they behave as expected in the written test specifications. The following three tests are done:

- CAN communication tests.
- Robustness tests of the CAN-communication.
- User functions tests.

¹ CAN stands for Controller Area Network.

² ECU stands for Electronic Control Unit.

The CAN-communication test is performed to verify that the ECUs are sending CAN messages as specified. Each ECU is supposed to send a specific number of messages on the bus with a certain time interval. Robustness tests of the CAN-communication are done to determine whether the systems manage to resume correct communication after a disturbance in the CAN-conditions. Even the bus capacity, i.e. bus load during abnormal conditions or when error frames are sent on the bus, is controlled in these tests. The user functions tests involve hundreds of test with different purposes. A user function can be very simply to test or more complicated with several ECUs involved.

Almost all systems have its specific control unit. This makes it easier and more efficient to update functions and for example develop more efficient engines that need less fuel. Some examples of systems that use control units for their functionality is:

- The ABS-brake, that makes the wheels rolling during a panic brake.
- The engine.
- The gearbox.

The automatic gearbox system has not been tested in the lab before, since it has not been implemented there. The ECU of this system needs information about the engine speed, the turbine speed and the propeller shaft speed. These signals are measured along the driveline with sensors. But when the gearbox system shall be tested in a lab, those three signals have to be simulated in real time, because the system will be run without a real gearbox. The relationship between them needs to be realistic and they have to react dynamical, to avoid the ECU to send error messages. This leads to a need for a mathematical model of the driveline of a truck, which simulates the speeds of the driveline in real-time.

1.2 The thesis

The purpose of this thesis is to create a dynamical model of the driveline of a truck that describes the motion in longitudinal direction. If the model can be built up in a modular way, this is preferable to non-modular model, thus this will increase the possibilities for updates and to make changes in the model. The driveline includes an automatic gearbox system that never been simulated before at Scania. This management system, i.e. the ECU, manages three different parts of the driveline: a converter, a gearbox and a retarder. Further, all components in the driveline that connect the engine with the wheels have to be modeled.

The model has to fulfill real-time simulation and communicate with the existing control units in the integration lab. The simulation model has to show dynamical response and behave very close to the reality. Otherwise, if the speed signals not behave as in a real truck, then the ECUs connected to the model believe something is wrong and degrade its functionality. Further, error frames would be sent through the bus.

It is desirable that the whole gear shifting range is possible to test. This means the goal is to be able to simulate the model in the same speed range as a real truck can be driven, i.e. from standstill to about 90 km/h. Further, the driver of the simulation shall be able to control the vehicle speed by the accelerator pedal, the brake pedal and the retarder hand level. These are all implemented in the integration lab. To perform an automatic gear shifting the ECU of the automatic gearbox system has to achieve a good shifting point, i.e. suitable speed values from the driveline have to be detected. Then an automatic gear shifting will be performed.

2 Measurements

In the beginning of this master thesis project several measurements on speed signals from a truck called “Lloydén” were performed. These measurements were all performed at Scania’s test track. The purposes of these measurements were to find out how the different speed signals from the driveline of truck with an automatic gearbox are affecting each other. This means that the relation between the speed of the input and output shaft has to be known during the whole driving, i.e. during a shift and between two shifts. Another problem that has to be solved was to find the connection between the engine speed and the turbine speed of the converter. Since the converter operates hydraulically and works as a fluid coupling, there are no obvious relations between the two speed signals. This behavior can be seen in figure 3 below, where upshifts to the first and second gear are displayed.

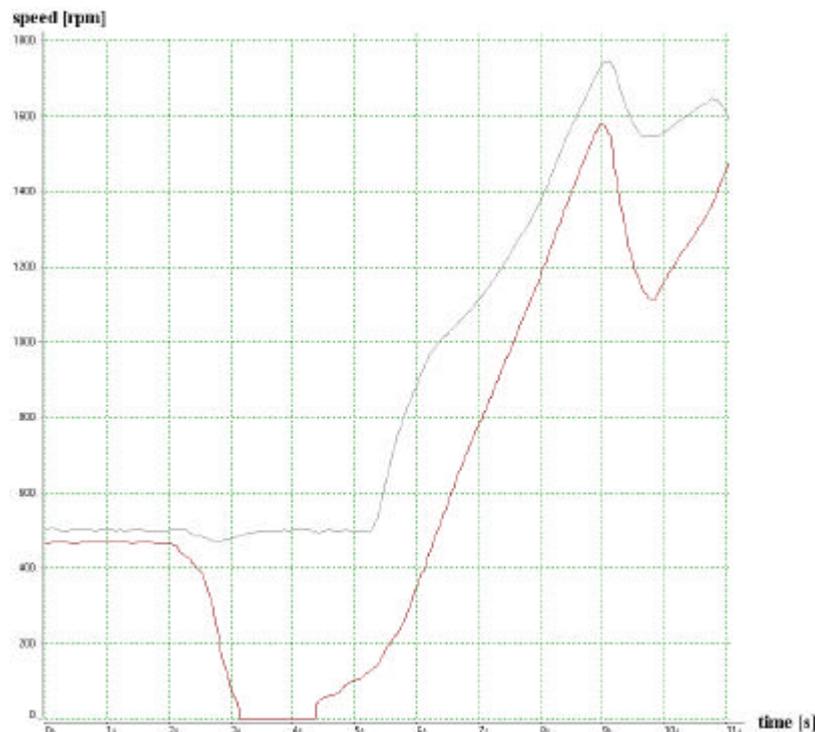


Figure 3. The upper curve shows the engine speed and the lower curve the turbine speed

Figure 4 represents turbine speed during a typical upshift. The locally maximum points for the turbine speed are all occurred within a shift in process. By analysing the time from the shift initialisation point to the locally turbine maximum speed, the behavior of the mechanical gear ratio can be determined. During a shift, changes in engine speed are reflected by changes in turbine speed. When an upshift occurs, turbine speed decreases reflecting the change in the transmission gear ratio. After the shift is completed the vehicle continues to accelerate, and the turbine speed will begin to increase. When the turbine speed reaches an intended rpm, i.e. the shift initiation point, the ECU commands an automatic upshift to begin. The modeling of the gearbox module is described in section 6.3.

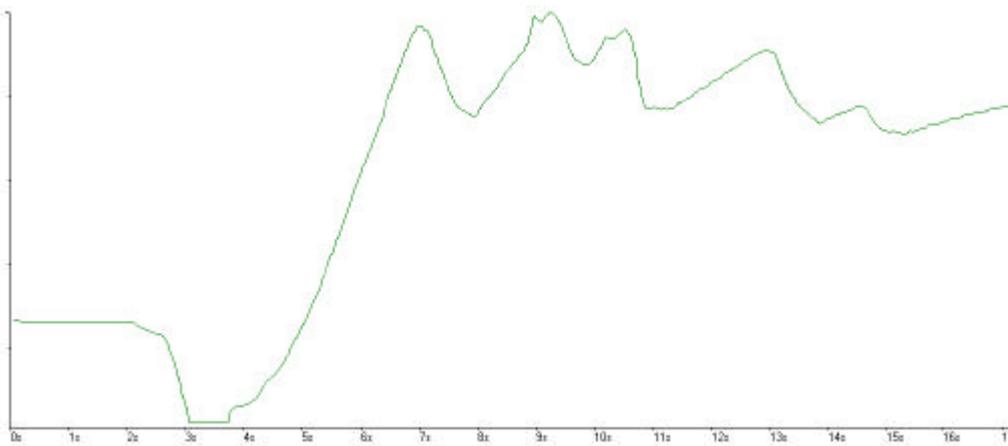


Figure 4. Turbine speed during an upshift

The upper signal in figure 5 shows the gear ratio signal and the lower describes the shift in process signal, i.e. the time needed to perform a complete shift. It can be realized that the gear ratio signal is changing in steps, from one gear ratio level to another. These changes are performed at the very end of the shift in process.

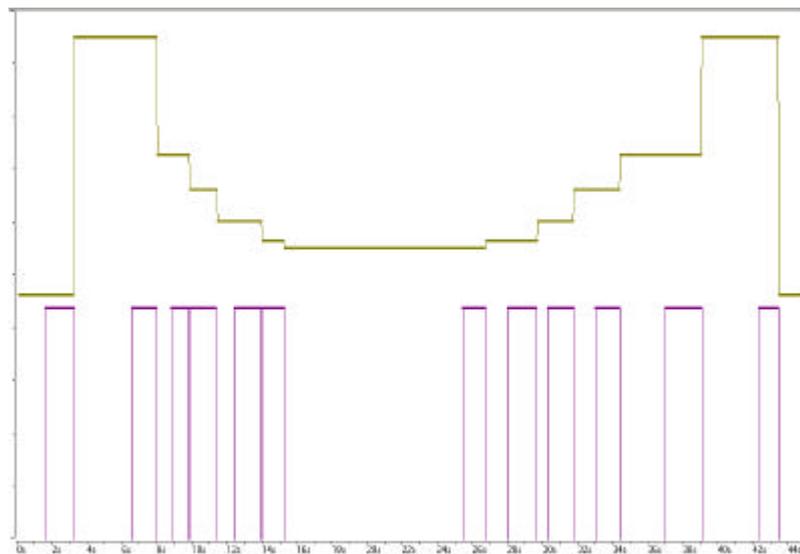


Figure 5. The gear ratio signal and the shift in process signal

3 Modeling

One of the most difficult problem that an engineer who works with modeling deals with, is the question about how to translate a physical phenomenon into a set of equations that it describes. It is usually difficult to describe phenomenon totally, so one usually strives for a set of equations that describes the physical system approximately and adequately.

In this section the applied modeling and simulation environment are described. The purposes of modeling languages and the advantages of building modular models are also described. Further, the way the modeling software handles the causality of models is even treated. In chapter 3.3 the interfaces between submodels are further discussed.

3.1 Modeling and simulation environment

The following programs have been used for modeling, c code generation, compiling, simulation and checking the CAN traffic:

Dymola¹

Matlab³

Simulink³

Microsoft Visual C++

Control Desk⁴

CANoe⁵

The purpose of a modeling language is to describe the behavior of small pieces of a larger system. A modeling language should support reuse of previous work and help to handle the complexity of systems when they further are enlarged. In the market today several simulation software packages are available. With few exceptions, all simulation packages are not working well in different areas in a reasonable way.

Since the lab already was equipped with hardware from dSPACE, this simulation interface was for natural reasons used for this project. Real-Time Workshop has been applied for generation of code for the dSPACE hardware. Real-Time Workshop automatically generates source code from Simulink block diagrams. Together with dSPACE's Real-Time interface, these tools provide an easy transition from the block diagram to dSPACE's real-time hardware. A program called Control Desk is finally used to run the real-time simulation.

In Control Desk one can create an instrument panel where the values of the variables in the model can be displayed in real-time during simulation. Therefore a virtual instrument panel can be created, with ordinary gauges, numerical displays, spin buttons and sliders. Parameters can also be defined as tunable, which means the values can be changed directly.

¹ Dymola stands for Dynamic Modeling Laboratory and is a trademark of the Swedish company Dynasim AB.

² Matlab is a registered trademark of MathWorks, Inc.

³ Simulink is a registered trademark of MathWorks, Inc.

⁴ ControlDesk is a program delivered by the German company dSPACE.

⁵ CANoe stands for CAN Open Environment is a program developed by the German company Vector-Informatik. The program is used to log traffic on CAN buses.

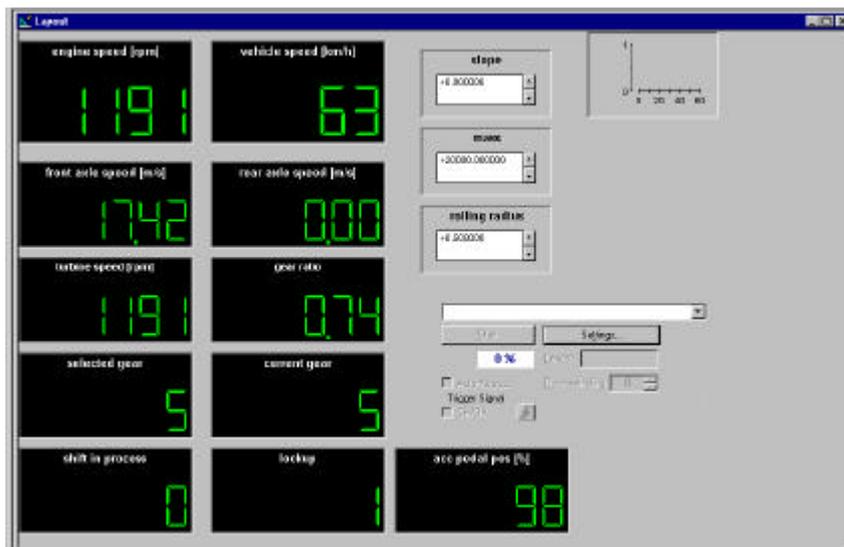


Figure 1. Display of a real-time simulation in Control Desk

3.2 Modelica

Modelica is a language for modeling of physical system, designed to support libraries of standard components and to simplify model exchange. It is a quite modern language built on acausal modeling with mathematical equations and object-oriented construction.

In Modelica equations are used for modeling of physical phenomenon. No particular variables need to be solved for manually, instead the software that handles the Modelica language is doing this automatically. Modelica can be used to solve a lot of problems that can be expressed in terms of differential-algebraic equations (DAEs) that describes the behavior of continuous variables. Modelica also support other types of equations such as ordinary differential equations (ODE).

The use of object-oriented construction means that systems are build up by objects, i.e. components in model libraries that together create an entire model. Models are hierarchically decomposed into submodels. Furthermore, a component can have properties from several other components. This makes it easier to reuse and exchange models, submodels and model libraries, and this is a main benefit with an object-oriented language. Another advantage of an object-oriented language instead of block-oriented is that there is no need to rewrite the equations in explicit form.

In Modelica the physical phenomena are modeled with general equations. Physical modeling is based on the relationship between physical quantities. For a driveline system, quantities such as torque and angular velocity are of great interest. These quantities are predefined in Modelica.

3.2.1 Acausal modeling

An advantage of Modelica in comparison to other modeling languages is that it supports acausal modeling, which means that there is no sense in what way the equations are read. Efficient graph-theoretical algorithms are used to determine which variable to solve in each

equation and to find minimal systems of equations. In many other modeling languages it is common that a variable is assigned a value, which makes it impossible to change the direction of the flow. The use of acausal modeling allows easier reuse of component models.

Since the language builds on non-causal modeling with ordinary differential and algebraic equations, there is a need of sorting the equations of the entire model before a system can be simulated. To translate the equations of a model the Modelica translator is used. The translator automatically organizes the equations into state space form. Therefore no variables have to be solved for manually.

3.2.2 Connectors

The model is build up by two different types of mechanical components, rotational and translational. All components in the driveline from the engine to the wheels are based on rotational mechanics. On the other hand, the connections between the wheels and the chassis module are instead based on translational mechanics. For rotational components torque is considered as intensity and angular velocity as flow, and in a similar way for translational components are forces respective velocities used instead.

Connections between submodels are conveniently described by defining connectors that model physical couplings as they are found in real systems. In Modelica there are specific objects that represent this connection between the different components. These objects are for mechanical components called flanges. The ingoing connection is usually called flange_a and the outgoing flange_b. Between two connections the value of the angle shall be the same. The torque is defined as a flow and thus sums up to zero in a connection node, i.e. outgoing torque from one component is equal to ingoing torque of next component. The equations describing the components are thus describing the relations between intensity and flow.

3.3 Dymola

The simulation model was built by using the modeling software Dymola, which stands for Dynamic Modeling Laboratory. With Dymola one can construct a model of a physical and mechanical process.

The modeling environment Dymola, delivered by the Swedish company Dynasim AB, is a hierarchical object-oriented tool for modeling and simulation of different kind of physical systems. The usual need for manual conversion of equations to block diagrams is removed by use of an automatic formula manipulation. The models are divided into several levels with models, classes and equations. Equations are used in the lowest level.

The basic idea behind the Dymola modeling language was to use general equations, objects and connections to allow the modeling work more physically behaved instead of based from a pure mathematical perspective. Dymola is based on the modeling language Modelica. Modelica allows integration and reuse of code developed in different modeling and simulation environments.

Tools and model libraries are available in Dymola by the Standard Modelica Library. The libraries are available in many different engineering domains as for electronics, rotational and

translational mechanics, thermodynamics, control etc. Further, inside these libraries several standard components are implemented. There is also possible to create own component libraries. The intention is to let the user use available standard components and then self develop components to create a complete model. When all components have been built up they can be dragged to the entire model platform and connected together.

In Dymola there is a Modelica translator that can be used to translate the equations of the model and generate c-code of the implemented Dymola model. By this operation the model can be imported into a Simulink model. Further, through this procedure the code is converted to a MEX⁴-file format and stored in a Simulink block, called S-function⁵ (further described in section 3.4).

3.4 Simulink

Simulink is an interactive environment integrated in Matlab for modeling, simulation and analyzing dynamic systems. Since Simulink is integrated with Matlab, the program is normally well known for many Matlab users. Simulink provides a graphical user interface for constructing block diagram models via drag and drop operations. Models can be grouped into hierarchies to create a simplified view of components or subsystems. Therefore a model built up in Simulink consists of blocks that correspond to subsystems of the model. In each block there are mathematical relationships that describe the physical behavior of the system. If the subsystems affect each other, information between the blocks has to be exchanged in order to evaluate the relationships.

A system using block diagrams is described by quantities that are known and quantities that are unknown. Block diagram is build up by components called blocks, which uses the known quantities so calculate the unknown quantities. Therefore all blocks in a Simulink model have the following characteristics: an input (u), an output (y) and a state (x).



Figure 2. A block diagram in Simulink

The signal flow between components is predetermined and limited into one direction. Every signal represents a variable. Furthermore, the unknown variables have to be expressed as functions of input signals given to the blocks. Therefore the calculated variables in the blocks must be given as output signals, so they further can be connected to other blocks.

Simulink provides a large set of modeling tools that can be used to develop detailed block diagrams of systems. However, if a truck model is going to be built up in a modular way then all components in the model have to be represented by blocks. Further, as mentioned previously, the signal flow through each block must be known when implementing the relationships. This means equations for the entire model have to be known when modeling each block. Therefore to model each component independent of each other is difficult.

⁴ MEX is a compiling utility in Matlab.

⁵ S-function stands for system-function and is a programming language description in Simulink.

The benefit with Simulink is the easy to use design and since the software is integrated with Matlab it makes it very powerful with a lot of useful analysis and design applications. Another benefit is the easy way to create plots of signals. Further, the ability to easy and quick simulate a model built in Simulink is another good thing with this modeling program. The simulation involves the numerical integration of sets of ordinary differential equations (ODEs). Simulink provides a number of solvers for simulation of such equations.

As mentioned previously in section 3.1 is Simulink in this project used as an interface with the dSPACE hardware. Further, Simulink is applied to import the Dymola model through a S-function block. S-function is a computer language description of a Simulink block. S-function can be written in Matlab, C, C++, Ada, or Fortran. S-functions built by c-code are compiled as MEX-files using the “mex” utility. The S-function is further included in a block that are drawn with one input and one output port, regardless of the number of inputs and outputs of the subsystem. This means that the entire Modelica model is included into the Simulink model by the created S-function block.

4 The electronic control units

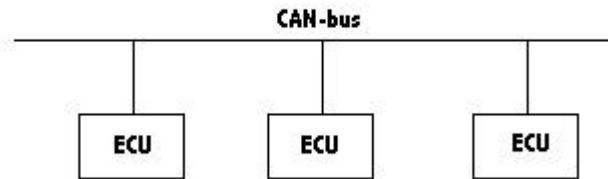


Figure 3. The ECUs connected to the CAN-bus

An electronic control unit, ECU, is a microcomputer that can be seen as the brain of a control system. As mentioned in the introduction part, the ECUs in Scania's test lab are connected together by three different CAN-buses. To be able to have a communication between the ECUs and the truck model, an I/O interface has to be implemented, that can deliver CAN-messages as input signals to the model. Further, the I/O interface has to be able to generate speed signals from the model. The hardware needed for simulation is described in section 5.

In this project the ECU of the automatic gearbox system is the one of primary interest, since it receives simulated speed signals from the created truck model, which are the engine speed, the turbine speed and the output shaft speed. Through these signals the ECU is able to determine shift sequences, shift timing, and clutch apply and release pressures. There is also a lot of other information that are sent to the ECU, such as: shift selector position, throttle position, oil level, retarder temperature, engine water temperature and oil sump temperature. Then the ECU compiles and processes these data.

Further the ECU is programmed to provide the most suitable operating characteristics for variations in load, terrain or environment, and to adjust for clutch wear. Signals processed by the ECU make it possible for the microcomputer to determine the characteristics of a shift in process. The ECU is constantly adjusting for changes in operating conditions, applying closed loop control and using adaptive logic. More information about the behavior of the control unit of the gearbox system can be read in [9].

5 Hardware needed for real-time simulation

To be able to except off-line simulations also run simulations in real-time and communicate with the control units, some hardware equipment has to be connected. To execute the model in real-time the state variables have to be updated before the sample time has elapsed. This demands a fast computer. The properties of the applied numerical integrator are described in appendix E.

An I/O interface is used to implement the connection between the Simulink model and the hardware system. In this apply a special computer system, called Autobox, is used. The system, i.e. the computer and the I/O interfaces are delivered by the German company dSPACE, and is particularly intended for vehicle simulations. A dSPACE PCMCIA card is utilized to connect a dSPACE hardware with the computer

The computer uses a Power PC Processor with a clock frequency of 480 MHz. The I/O is able to generate a lot of different electrical pulse signals that normally are generated by sensors in the vehicles. During simulation of the model the following signals are utilized from the I/O:

- PWM signals, used to sense the engine speed, the turbine speed in the converter and the propeller shaft speed.
- Sinusoidal signals, used to sense the speed of the wheels. Sinus pulse trains are also used to sense the speed, the position and the rotational direction of the engine speed.

The I/O is only able to generate totally 4 sinusoidal signals, which therefore is enough to simulate the speed of four different wheels.

Since the model also needs information from control units in different parts of the driveline, the I/O is also used to deliver CAN signals to the model. Simulink's block libraries are enhanced by a specific dSPACE block library for the I/O hardware support. The blocks represent the CAN messages that are input signals to the simulation model. The CAN messages are implemented in Simulink as blocksets called RTI CAN. This means that the CAN configuration is carried out by block diagram in Simulink. These blocks provide convenient code generation for the I/O board. Further, several CAN signals are defined in each CAN message that is composed to a block in Simulink. The CAN messages are imported from the CAN database. Configuration information, such as baud rate, identifier, message length, is entered in these blocks.

6 The driveline parts

This chapter explains all modeled driveline components individually. Further, this chapter describes the different forces and torques acting on different parts of the driveline in a truck. Further all of the speed signals from the engine to the wheels are determined. In longitudinal direction specific forces affect the vehicle. Aerodynamic and gravity forces that act on a vehicle are put in relation to the traction and braking forces applied by the driver.

The driveline is a fundamental part of a vehicle and can be seen as the system that transfers energy from the combustion inside the engine to kinetic energy of the vehicle. In general, the purpose of the driveline is to reach as efficient transformation as possible, which leads to higher performance and lower fuel consumption.

A curved road model applies side forces to the vehicle, but these properties will not be treated in this study, because such simulations are here of less interest. Therefore the developed model only describes the truck movements in longitudinal direction. To fulfill this, a model including the driveline and the chassis is needed. The driveline consists of all components that connect the engine with the tires. A general view of the driveline can be seen in figure 6. In the chassis model the truck resistances due to road slope and wind resistance are included, also the distribution of normal forces acting on the wheel axles are a part of the chassis component.

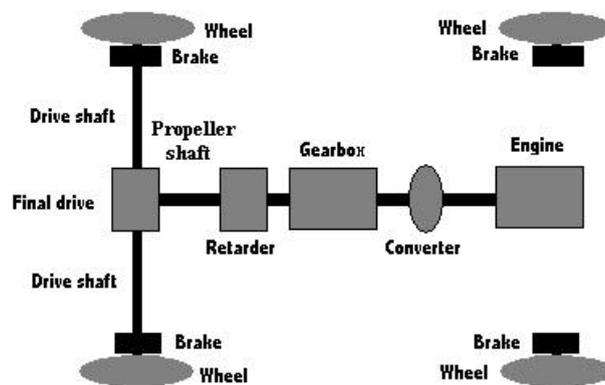


Figure 6. Schematic picture of the modeled driveline

In this master's thesis project an automotive driveline has been modeled by using the modeling environment Dymola. All components, mentioned later on in this section, have been modeled individual and then the entire model has been built up by these components.

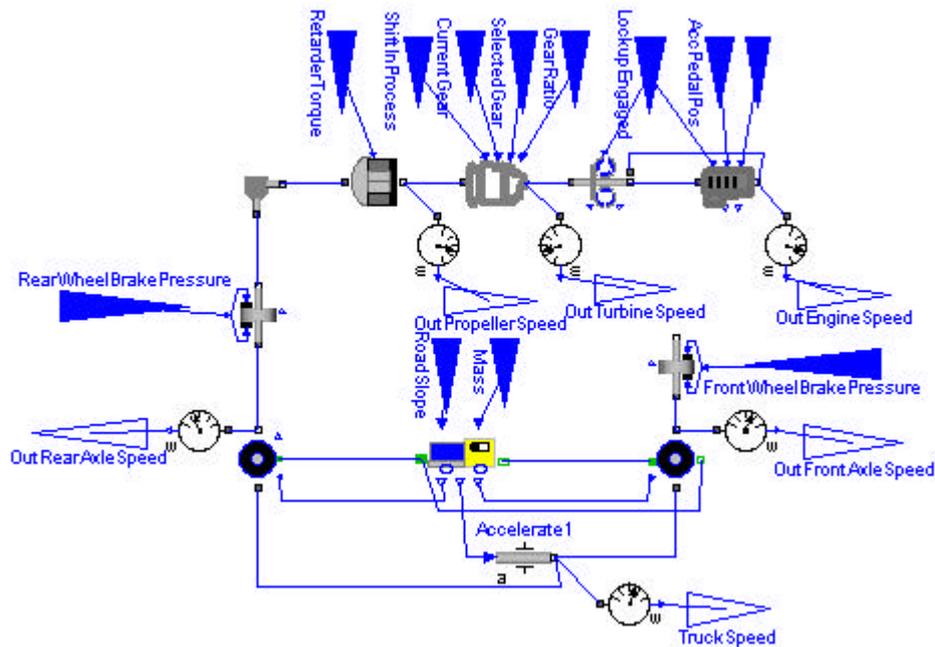


Figure 7. The entire Dymola model

Figure 7 above shows the entire Dymola model with all included components and how these are connected to each other. The quantities that are transferred between the rotational components are torques and angular velocities, and between translational components are forces and velocities transferred. The dark filled triangles describe input signals to the model and the white non-filled triangles represent output signals.

The following CAN messages are used as input signals to the model:

- The lockup engaged signal.
- The shift in process signal.
- The current gear.
- The selected gear.
- The actual gear ratio.
- The engine start signal.
- The accelerator pedal position.
- The brake torque of the retarder.
- The brake pressure acting on each wheel.

Other variables that are input signals to the Modelica model are the road slope and the vehicle mass. These two mentioned input signals are defined as tunable variables during the simulation. There is of course of less interest to change the mass of the vehicle during simulation, but between simulations there can be useful. The advantage of having variables defined as tunable is to get away from restarting a new simulation or even worse reload the model again when a parameter is assigned a new value.

To calculate the acceleration respectively the deceleration of the truck all forces acting in longitudinal direction are taken into account. This involves forces generated by the tires, and counteracting forces as rolling resistance, air drag and road slope. By adding up all the forces, the change in the vehicle speed can be calculated by using Newton's II law.

The output signals from the model consist of speeds from different parts of the driveline. These signals are needed because they are sensed in a real truck by speed sensors. The electrical pulse signals that are output signals to the model will make the ECU believing the signals are generated from a real driveline. The simulated speed signals can be seen in figure 8. If the pulse signals not are relative to each other as expected by the ECU, i.e. within tight ranges, the ECU will begin to generate error frames on the bus. The following signals are sent to the ECU of the automatic gearbox system:

- The engine speed, w_e .
- The turbine speed, w_t .
- The propeller shaft speed, w_p .

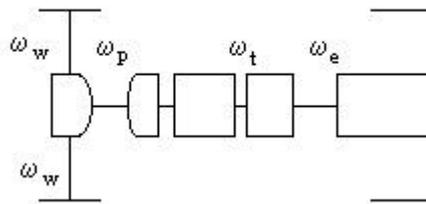


Figure 8. Different speed signals from the driveline

The figure above shows the simulated signals that are sent to different control units in the test lab. Besides the three simulated PWM-signals sent to the gearbox system, are the wheel speeds, w_w , sent to the brake system and the propeller shaft speed is delivered to the tachograph.

The driver of the simulation shall be able to affect the speed of the simulated vehicle by the ordinary accelerator pedal and the brake pedal. Further an external brake can be used to assist in stopping the vehicle. This brake utility is called retarder and is controlled by a hand level selector. The retarder is acting directly on the propeller shaft. By the gear level the driver can put the vehicle into eight different gear states (R, N, D, 1, 2, 3, 4 and 5). Condition 1, 2, 3, 4 and 5 act as the driving gear (D) but with the difference that the number defines the highest possible gear number to activate. Therefore in drive mode all forward gears can be activated, i.e. gear number one to six. In this mode the gear shifting is performed automatically by the ECU of the gearbox system.

6.1 Engine

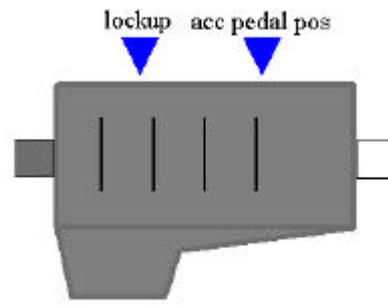


Figure 9. The engine component

The engine is the power source of a driveline. The engine produces different magnitude of torque depending on the speed of the engine. Low engine speed delivers less torque. For trucks the maximum torque can be achieved when the engine speed is between 1000-1400 rpm. The output power of an engine can be the same for two different engine speeds and torques. The relationship between the torque and the angular velocity of the engine is determined by the fuel injection system. It can be mentioned that it is more fuel economical to drive at low engine speed and high torque settings than at higher engine speed and lower torque.

The magnitude of torque out of the engine is determined from a motor curve describing the relationship between engine torque and engine speed, given from [10]. It is not easy to find a function that can be valid for describing the curve in the whole speed range with acceptable accuracy. Therefore the function has to be divided into several parts. By this operation the available engine torque can be described by nine functions valid in different speed ranges.

$$T_{engine} = f_i(w_c), i = 1, 2, \dots, 9, w_c = [0 \quad 2400] \quad (6.1)$$

Where:

T_{engine} : maximum engine torque.

w_c : crankshaft speed.

The equation in (6.1) is only useful when the crankshaft speed of the engine is higher than the turbine speed, otherwise the engine is no longer driving the rest of the transmission, and further no fuel is injected to the cylinders. Since the amount of injected fuel can be considered as proportional to the available torque the engine can produce, results no fuel injection in that no positive torque can be transmitted. An inner friction is always acting on the engine that decreases the amount of possible delivered torque. This inner friction is already included in the engine torque function above, i.e. equation (6.1) describes the real amount of torque that can be utilized. However, when the turbine speed becomes higher than the engine speed this equation cannot be used, since the function above is only defined for positive torques. Therefore another function describing the losses in the engine has to be considered. Such function is treated and calculated later on in this section.

Figure 10 at next page is a rough sketch of the maximum engine curve, i.e. the maximum delivered torque from the engine.

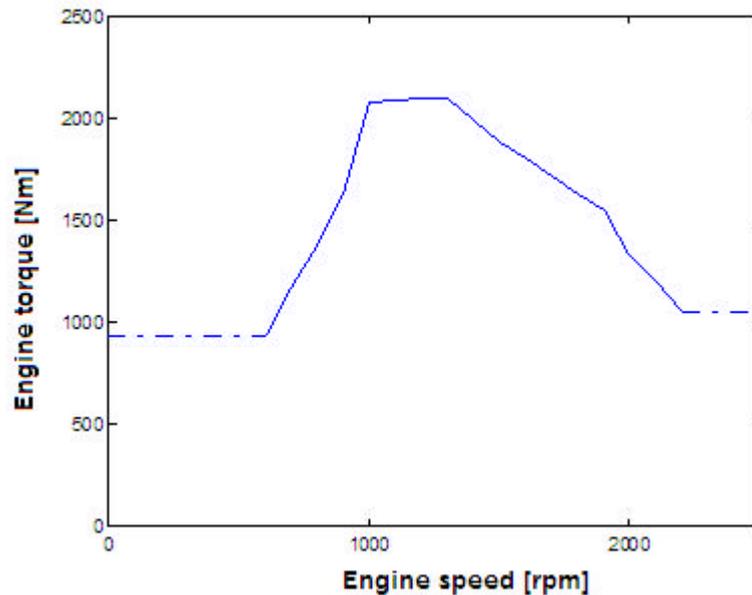


Figure 10. Schematic picture of the engine torque as function of engine speed, adjusted from [10]

The driver controls the engine with the accelerator pedal. The purpose of the engine management system seen from the driver can be to control the injection system so that the engine delivers a demanded engine torque or speed to the rest of the driveline.

The driver pushes the accelerator pedal that is mechanical connected to a sensor. The output signal of this sensor is analog and received by an electronic control unit, called the coordinator⁶, where the signal is further processed. Then the signal is normalized to a percent value of maximum depressed pedal position, and sent to the engine management system. The percent value is then converted to a reference engine speed that the system tries to control the real engine speed against. Calculations in the control unit result in a requested amount of injected fuel as with large accuracy are delivered to the individual cylinders.

The reference engine speed can easily be described by a linear function of the accelerator pedal position. When the truck stands still and the engine is idling then the engine speed is about 500 rpm. Therefore the reference speed cannot be below 500 rpm and further the maximum speed is limited to 2400 rpm. This leads to the following equation:

$$\mathbf{w}_{reference} = 500 + 0.01 * 1900 * \mathbf{q}_{acc_pedal_pos} \quad (6.2)$$

Where:

$\mathbf{w}_{reference}$: reference engine speed.

$\mathbf{q}_{acc_pedal_pos}$: accelerator pedal position.

⁶ The coordinator is a control unit that coordinates the ECUs and gateways messages between the different buses.

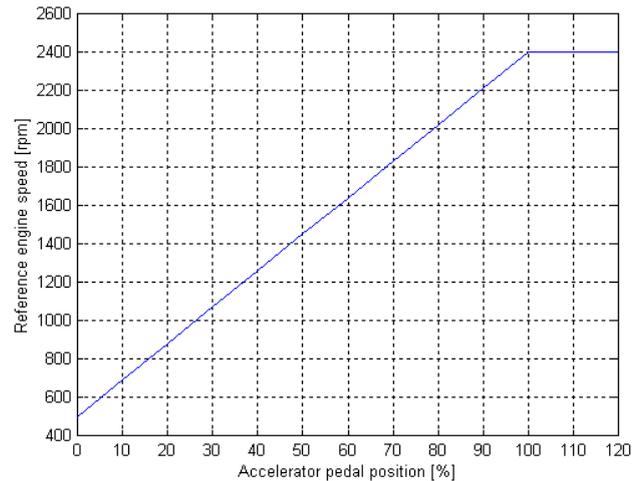


Figure 11. Reference engine speed as function of accelerator pedal position

The figure above shows the behavior of the reference engine speed as function of the accelerator pedal position. When the reference engine speed and the actual engine speed are known, it is easy to determine the control error. The error, i.e. the speed difference between demanded and actual speed, is handled by a speed regulator called RQV in the engine management system (further discussed later on in this section). The speed difference can be described by:

$$\Delta w_{present} = w_{reference} - w_c \quad (6.3)$$

Where:

$\Delta w_{present}$: present engine speed difference.

$w_{reference}$: reference engine speed.

w_c : crankshaft speed.

6.1.1 Shuffle torque

The inner friction of an engine is an important factor when the maximum attained engine torque is going to be determined. At higher engine speed the friction can be as high as one fifth of the engine torque. This friction torque is already included in the maximum engine torque curve (see figure 10) during the ordinary condition when the engine delivers driving torque to the transmission. However when the turbine speed becomes higher than the engine speed, which can appear in a downhill slope then the engine is no longer driving the transmission. The engine is thus instead driven by the transmission, and the torque out of the engine can therefore be considered as negative. During such conditions a friction torque called shuffle torque are calculated, as then represents the engine torque delivered from the engine. This is the reason to why a shuffle torque function also has to be determined.

The function for the shuffle torque is calculated from a number of measurement points, reference [11]. When these points are connected together through a curve, they nearly form a linear function. This means that the shuffle torque is considered as a linear function valid from the idling cycle (500 rpm) up to the maximum engine speed.

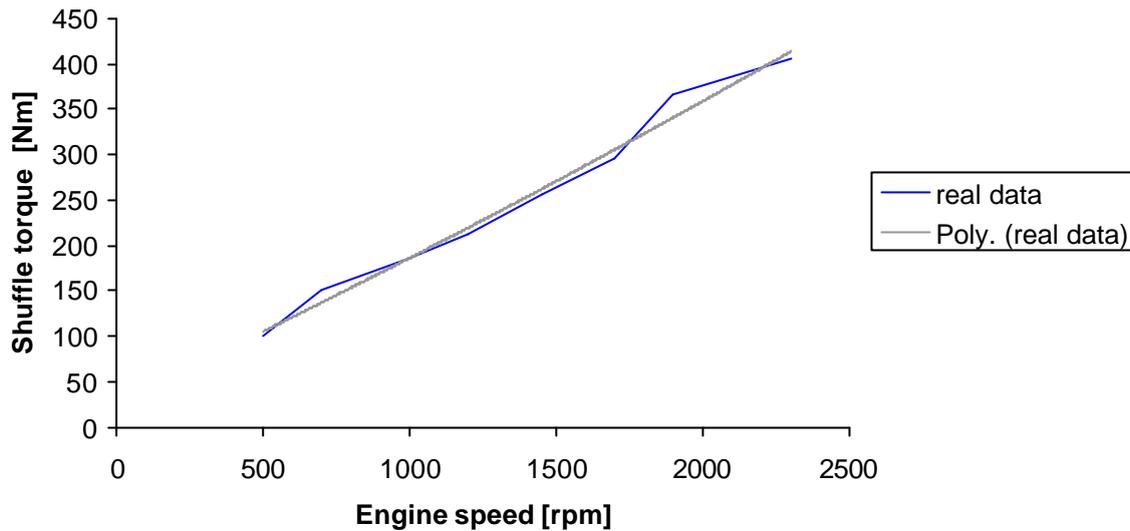


Figure 12. Shuffle torque as function of engine speed. The real data is taken from [11] and the gray line is an adjusted function describing the shuffle torque

6.1.2 RQV control

Two different control strategies are often used to express the signal from the accelerator pedal. The first is called “torque control” or RQ-regulator, and can be used to attain a given engine power at a given engine speed and pedal position. Hence, the regulator gives an output torque of the engine that is proportional to the position of the accelerator pedal. The second method is called “speed control” or RQV⁷-regulator, and interprets the given pedal position as a required engine speed. Traditional RQV control is a proportional control that calculates the engine torque from knowledge of engine speed and accelerator pedal position. The input to the RQV controller is the difference between the demanded speed and the present speed of the engine.

In this project the signal from the accelerator pedal is modeled with RQV control. The reason to this is that trucks with automatic gearboxes from Allison usually use this engine speed control. By using RQV the interval of the pedal position decreases, since the regulator helps to keep a reference speed even if the resistances increase, for example when driving in an uphill slope. However, it is not always appreciated to keep a demanded speed during all driving cases, thus the pedal position would act as a reference speed to a cruise control. To avoid that another reflection is done, where the change in injected amount of fuel as a function of increased load, is assigned a predetermined decrease of engine speed.

⁷ RQV is a description of the German “Regler Menge Vestell”.

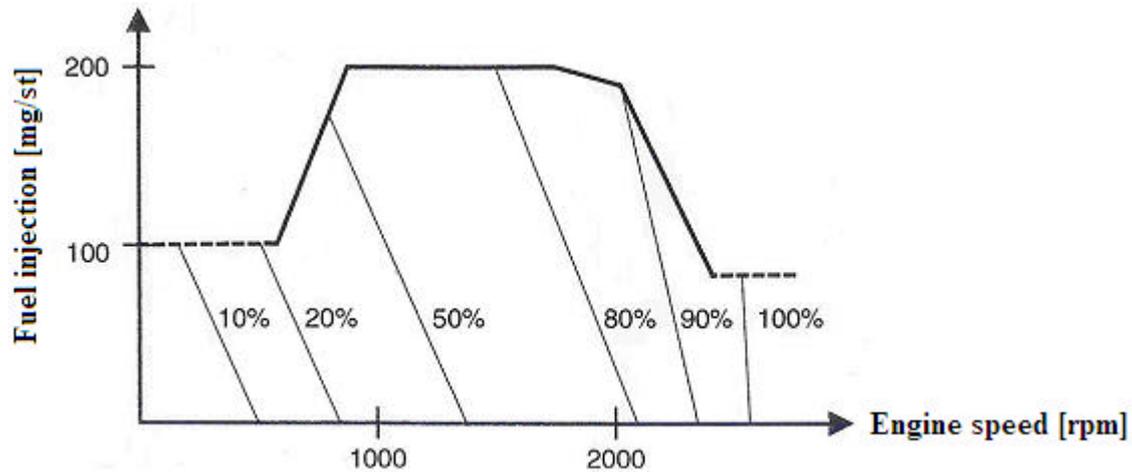


Figure 13. Schematic sketch of the principle of a RQV control, given from [4]

Figure 13 shows the relationship between the amount of injected fuel and the engine speed at different present pedal positions. It can be realized that a decrease of engine speed due to increases in load, results in a large increase of injected fuel and therefore also in a higher engine torque. The leaning lines are called “droop lines” and describe the changes in the amount of fuel in relation to variations in the engine speed. Note that the droop lines are leaning less at higher engine speed, which is implemented to avoid the full-load curve from falling down too much at the highest engine speed. The droop values as a function of accelerator pedal positions and injected amount of fuel can be seen in the following table.

Droop values [rpm/mg/st]											
Acc pedal pos [%]	0	11	21	32	42	54	68	83	98	100	
Injected amount of fuel [mg/st]	0	6.0	6.0	6.0	6.0	6.0	6.0	6.0	6.0	5.0	2.2
	50	4.5	4.5	4.5	4.5	4.5	4.5	4.5	4.5	3.8	1.7
	325	4.5	4.5	4.5	4.5	4.5	4.5	4.5	4.5	3.8	1.7

Table 1. Droop values for different accelerator pedal positions, given from [4]

The droop values together with the data according to figure 13 can be used to calculate an allowed speed difference between the reference speed and the actual engine speed at different pedal positions. This is done by multiplying the droop values with the amount of injected fuel into the cylinders.

$$\Delta w_{allowed} = f(q_{acc_pedal_pos}) \quad (6.4)$$

Where:

$\Delta w_{allowed}$: allowed engine speed difference.

$q_{acc_pedal_pos}$: accelerator pedal position.

A function of the estimated allowed engine speed difference can be seen in figure 14. Above the curve the speed difference is larger than the defined speed difference where the RQV starts to control the fuel injection. In such case maximum amount of fuel is injected. On the other hand at values below the curve the controller reduces the injected fuel such that the certain engine speed is obtained, i.e. the control system controls the fuel injection according to the droop lines in figure 13. More about the RQV control can be read in [4].

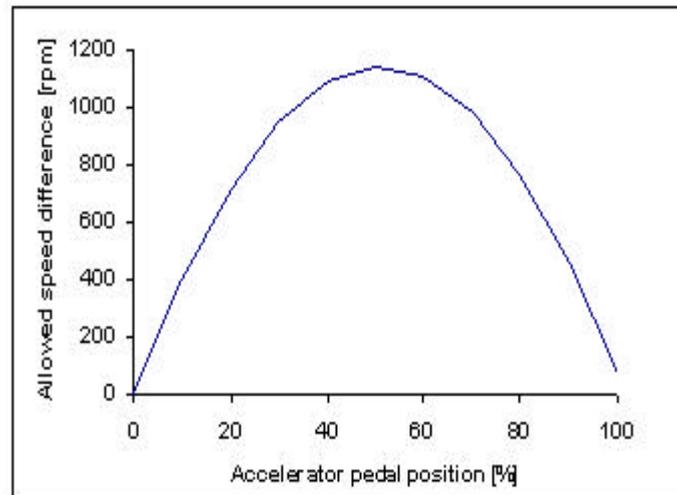


Figure 14. Allowed speed difference at different pedal positions. Calculated from the droop values and the fuel injection

Therefore if the present speed difference is smaller than the allowed speed difference, then the ratio between them determines the grade of the engine's maximum torque that can be delivered. However when the present engine speed is higher than the reference speed, i.e. one demands a lower engine speed than the engine delivers, then no fuel injection is done and therefore is no torque either transferred from the engine. The percentage of the maximum available engine torque can therefore be described by:

$$c = \begin{cases} 0 & , \mathbf{w}_{reference} < \mathbf{w}_c \\ 1 & , \Delta \mathbf{w}_{present} > \Delta \mathbf{w}_{allowed} \\ \frac{\Delta \mathbf{w}_{present}}{\Delta \mathbf{w}_{allowed}} & , otherwise \end{cases} \quad (6.5)$$

Where:

c : percent value of maximum available engine torque

$\Delta \mathbf{w}_{present}$: present difference between reference engine speed and actual engine speed.

$\Delta \mathbf{w}_{allowed}$: allowed speed difference to start RQV control.

When the percent value is determined, then to calculate the torque that can be produced by the engine is straightforward. The torque delivered by the engine can thus be expressed by:

$$T_{available} = c \cdot T_{engine} \quad (6.6)$$

Where:

$T_{available}$: available engine torque.

c : percent value of maximum engine torque.

T_{engine} : maximum engine torque.

6.2 Converter

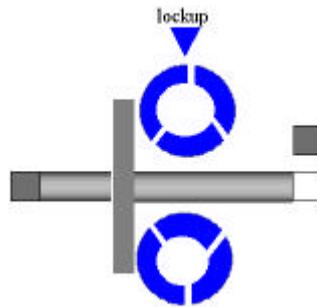


Figure 15. The converter component

Just like manual transmission trucks, trucks with automatic transmissions need a way to let the engine turn while the wheels and gears in the transmission come to a stop. In manual transmission the engine is connected to the gearbox by a clutch, which is able to connect and disconnect the engine with the rest of the driveline. On the other hand vehicles with an automatic transmission have no clutch that disconnects the transmission from the engine. Instead of a clutch a device called torque converter is used. Thus no clutch pedal is needed for gear shifting. This means that the converter connects the engine and the gearbox module.

The torque converter operates hydraulically and transfers torque from the engine to the transmission. It consists of a vaned converter pump, a vaned turbine, a vaned stator, and a lockup clutch and torsional damper assembly. Torque converter works as a type of fluid coupling, which allows the engine to rotate fairly independent of the transmission. When the engine is turning slowly, as the case when the vehicle is idling, the amount of torque passed through the converter is very small.

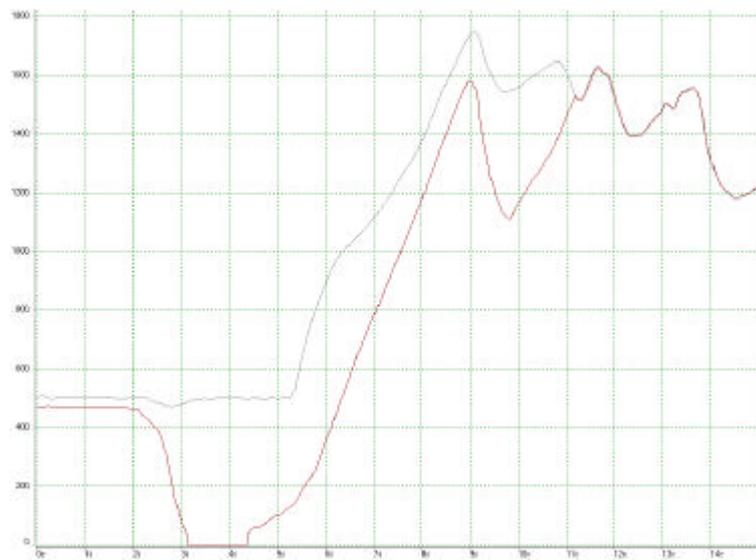


Figure 16. The upper curve shows the engine speed and the lower curve the turbine speed expressed in rpm during a measurement performed at the test track

The turbine speed is almost independent of the engine speed, as the measurements in figure 16 above shows, during the first 11 seconds. This is the case when the converter works as a fluid coupling and the lockup clutch mode is inactivated. Fluid coupling can transfer a large amount of torque from converter pump to the turbine.

When the accelerator pedal is more pressed down by the driver, the engine speeds up and pumps more fluid into the torque converter causing more torque to be transmitted to the rest of the driveline. Modern torque converters can multiply torque from the engine by two or three times. This can only appear when the engine speed is much higher than the speed of the transmission.

The charging pump pressurizes the transmission fluid. The pump rotates with the engine to provide a constant flow of pressurized fluid for the transmission hydraulic system. The pump inside a torque converter is a type of a centrifugal pump. As it spins, the fluid is flung to the outside and then a vacuum is built up that draws more fluid in at the center. The fluid then enters the blades of the turbine, which is connected to the transmission. This means the turbine causes the transmission to spin.

6.2.1 Fluid coupling drive

To determine the performance of a torque converter knowledge about the flow of the fluid has to be known in every single moment, i.e. during all possible relations between the engine and turbine speed. It is difficult to physically determine the ability of the converter to transfer torque, since the connection between the crankshaft of the engine and the input shaft of the gearbox is represented by a fluid coupling. Instead of trying to describe the behavior of the converter physically, measurements on the converter performance are used instead. These measurements, delivered by the producer Allison, gave information about the converter's capability to absorb and transmit torque in different speed ranges. To describe the performance of the converter, variables based on speeds and torques are utilized, namely speed ratios and torque ratios. The relation between the turbine and the engine speed determines the speed ratio, i.e.:

$$SR = \frac{w_{turbine}}{w_{pump}} \quad (6.5)$$

Where:

- SR : speed ratio.
- w_{pump} : pump speed.
- $w_{turbine}$: turbine speed.

The torque ratio is discussed later on in this section.

Absorption characteristics are used to determine the torque that can be absorbed by the converter. The amount of absorption is affected by the current engine speed and the speed ratio. The following equation is based on the absorption characteristics given by Allison, reference [9]:

$$T_{absorbed} = \left(\frac{w_{pump}}{a_1 \cdot (SR)^6 - a_2 \cdot (SR)^5 + a_3 \cdot (SR)^4 - a_4 \cdot (SR)^3 + a_5 \cdot (SR)^2 - a_6 \cdot (SR) + a_7} \right)^2 \quad (6.6)$$

This equation is only valid when the engine speed is higher than the turbine speed, i.e. the speed ratio is below one. The converter is not able to absorb more torque than the engine can

produce. Therefore, the intersection between the available torque from the engine and the absorbed torque by the converter gives the current engine speed. In other words, the engine speed is determined by iterations, i.e. the value of the engine speed is changed by iterations until the following equation is fulfilled.

$$T_{available} = T_{absorbed}$$

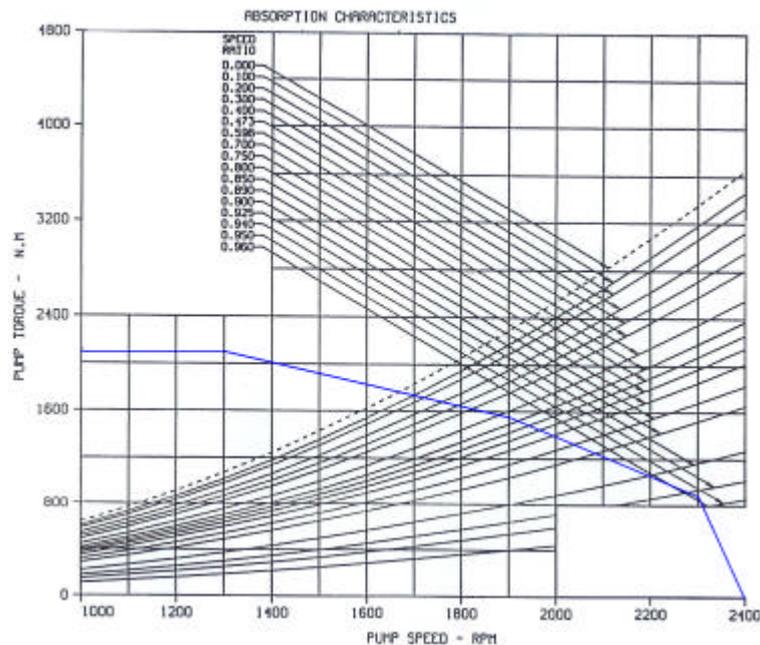


Figure 17. The blue line is a rough sketch of the available engine torque and the black curved lines describe the converter's capability to absorb torque, [9]

The figure above shows the converter's capability to absorb torque for different gear ratios. A high value of the speed ratio leads to a low grade of absorption. In other words, to absorb maximum torque the speed difference between the engine and the turbine must be large.

In the case when the engine speed instead becomes lower than the turbine speed, the definition (6.5) gives a speed ratio above one, which means the transmission drives the engine. Since the speed ratio is only defined between zero and one, the absorption characteristics cannot be used directly.

A consequence of a situation with a higher turbine speed than engine speed is that the torque absorption then has a different characteristic. The reason to this is the opposite direction of the fluid flow that passes through the turbine. As a result of the opposite flow, the turbine speed becomes lower because the vanes of the propeller are then acting in a non-optimum angle. Therefore the torque absorption becomes lower, and the value can by estimation be set to 0.2 times the ordinary absorption. Further, the absorbed torque has to be considered as negative since counteracting forces from the transmission influences the engine. This leads to that the absorbed engine torque can be expressed by:

$$T_{absorbed} = (-0.2) \cdot \left(\frac{\mathbf{w}_{turbine}}{a_1 \cdot \left(\frac{1}{SR}\right)^6 - a_2 \cdot \left(\frac{1}{SR}\right)^5 + a_3 \cdot \left(\frac{1}{SR}\right)^4 - a_4 \cdot \left(\frac{1}{SR}\right)^3 + a_5 \left(\frac{1}{SR}\right)^2 - a_6 \cdot \left(\frac{1}{SR}\right) + a_7} \right)^2 \quad (6.7)$$

Where:

a_1 to a_7 : constants.

In the equation above, it can be realized that the speed ratio has been inverted and the engine speed from equation (6.6) is replaced by the turbine speed.

The blades of the converter elements are curved so that hydraulic fluid charged into the converter circulates from the pump element to the turbine, then to the stator and back to the pump. This interaction allows engine torque to be transmitted from one member to the other through a fluid medium and provides the mean of engine torque multiplying. The amount of torque multiplication depends on each converter's blade design. Blade geometry of a converter is also affecting its capacity to absorb engine torque.

The torque ratio is calculated from the speed ratio on the basis of absorption characteristics from Allison, see [9]. Torque ratio can be described as the torque multiplication factor of the converter. Since the value of the speed ratio is between zero and one, the torque ratio becomes inside the interval [1.9 0.99]. The following equation has been adjusted from a table of data including torque ratios and speed ratios:

$$TR = 3.8863 \cdot SR^5 - 5.4121 \cdot SR^4 + 0.6544 \cdot SR^3 + 1.1961 \cdot SR^2 - 1.2085 \cdot SR + 1.8993 \quad (6.8)$$

Where:

TR : torque ratio of the converter.

The absorbed engine torque together with the present torque ratio determine the amount of torque transferred to the gearbox, according to this equation:

$$T_c = TR \cdot T_{absorbed} \quad (6.9)$$

Where:

T_c : transferred torque by the converter.

6.2.2 Lockup operation

Some torque converters, as for Allison WTEC automatic gearbox system, have an internal, hydraulically operated "lockup clutch" in them. The lockup clutch provides a direct drive from the engine to the transmission gearing. Therefore, at some preset point the transmission will cause this converter lockup clutch to engage in order to mechanically lock the input turbine and the output turbine together. Therefore, with the lockup clutch engaged, the converter pump, turbine and turbine shaft all rotates at exactly the same speed as the engine. This mechanical lockup clutch eliminates converter slippage and improves the efficiency of

the vehicle a bit as kinetic energy losses are overcome. This leads to that the fuel consumption is reduced.

Automatic gearboxes equipped with a lockup clutch are designed to extend the operating speed range of the converter. At a particular operating point the lockup is engaged which means that all slippage between converter input and output members are eliminated. This means maximum power and speed from the engine flywheel can be transmitted. During lockup operation the converter's torque ratio and speed ratio are not applicable. Instead, the lockup clutch transmits full pump torque and speed to the converter turbine. This means that no torque multiplication is done, i.e. the torque ratio can be considered as the value one. Therefore the physical behavior during lockup can be described by the following formulas:

$$W_{turbine} = W_{pump} \quad (6.10)$$

$$T_c = T_{pump} \quad (6.11)$$

Where:

W_{pump} : engine speed.

$W_{turbine}$: turbine speed.

T_{pump} : absorbed torque in the converter.

T_c : transferred torque from the converter.

The lockup clutch engages in response to electronic signals from the ECU. As the turbine speed approaches the engine speed the ECU energizes a lockup solenoid in the control module. When the lockup clutch again is disengaged the converter returns to a fluid coupling drive, that appears when the turbine decreases below a predetermined turbine speed.

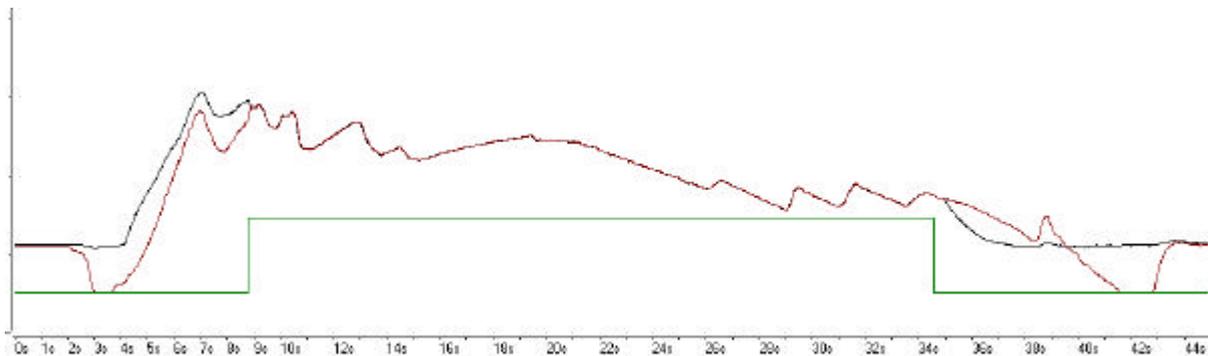


Figure 18. Engine speed and turbine speed together with the torque converter lockup signal

Figure 18 shows the engine speed (the upper curve) and turbine speed (the middle curve) during a normal truck driving. The lowest signal represents the torque converter lockup signal that is used as an input to the converter component. This signal becomes high (1) when the lockup is engaged and low (0) when disengaged. In the engage mode the turbine speed almost exactly follows the engine speed.

If an ABS control system is implemented and the system activates the ABS system then the torque converter lockup is automatically set to disengaged, i.e. converter returns to a fluid coupling drive.

6.3 Gearbox

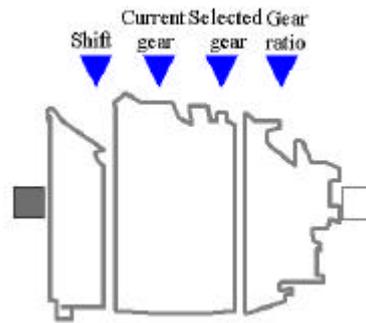


Figure 19. The gearbox component

It is today often desired to have an automatic gear shifting system on heavy trucks. Automatic gearshifts lead to better driving comfort and result even in less wear.

An automatic transmission uses planetary gear sets to create the different gear ratios. Transmission fluid flows to different clutches to achieve range shifts. Solenoids and valves are used to control the flow and pressure of the hydraulic fluid inside the system. Seven solenoids are used to open and close the circuits that apply and release the clutches. For example the first gear is activated when clutch C1 and C5 is applied. The points at which shifting occurs is dependent upon predetermined speeds and other operating conditions.

The automatic gearbox system that has been used during this project has six forward gears, one reverse gear and a neutral. Every gear number has its own gear ratio, which can be seen in table 2. The ratio of the gearbox depends on the chosen gear, the lower gear the higher ratio. The gear ratio signal is called *Actual gear ratio* in the CAN database for CAN signals. Figure 20 shows the actual gear ratio signal during an upshift to the sixth gear number.

gear number	gear ratio
1	3.51
2	1.906
3	1.428
4	1
5	0.737
6	0.639
N	0
R	4.801

Table 2. Gear ratios for Allison WTEC III. Given by measurements performed at Scania's test track.

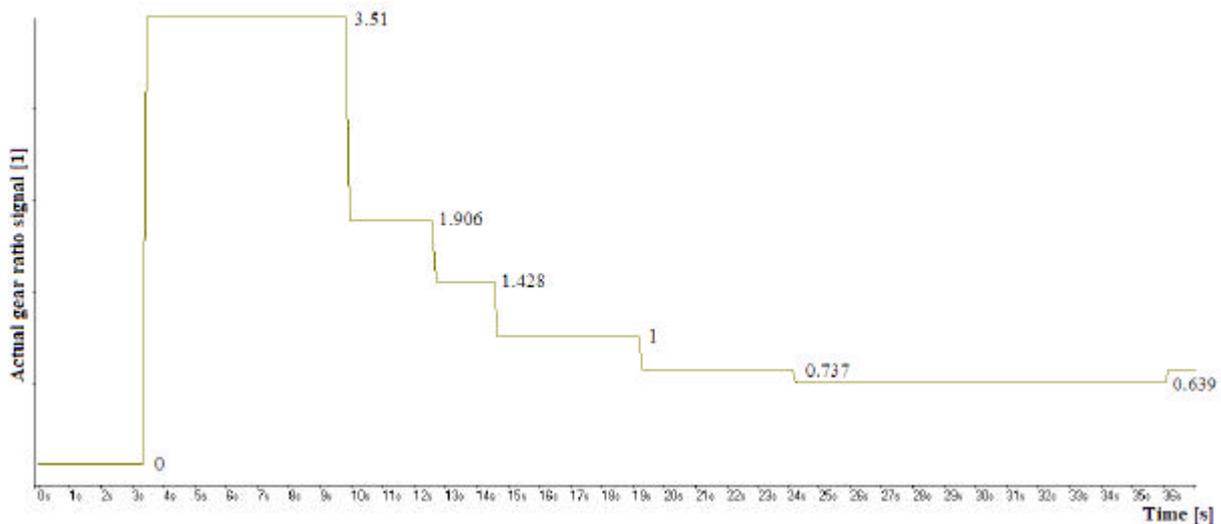


Figure 20. The behavior of the gear ratio signal during an upshift

By taking a look at the behavior of the gear ratio signal can it be established that the signal is changing quickly from one gear level to another at the very end of the shifting process. Since the gear ratio is the factor that provides the relation between the input and output shaft speeds of the gearbox, this discrete behavior of the signal is not acceptable to use when modeling the gearbox. Thus the signal would give an input shaft speed that not corresponds correctly to the output shaft speed, because the input speed will be affected by an entire gear ratio step in a moment. This makes it impossible to use the actual gear ratio signal directly, which means the signal has to be modified so it slowly is changed at the later part of shift in process. The problem is therefore to determine during what time interval within the shift in process the gear ratio signal has to be adjusted. Outside this shifting state the actual gear ratio signal can be used, because it then has a constant value that follows the gear ratio of each gear number.

Measurements on the gear shifting time have therefore been of interest. They are performed on the test track using a computer with a self-programmed Capi-program in CANalyzer⁸ to store the times from each gear shifting. The results from the measurements showed that the gear shifting time was almost the same for all gear numbers. The exceptional case was the first upshift where the shift was active during a longer time interval. The shift in process state can be seen as the time from that the ECU starts to prepare the gear shifting until the shift is completed.

By also looking at behavior of the engine and turbine speed, conclusions about the different parts in the shifting procedure could be done. From that it could be established that the filling phase of the disc was about 0.5 s and the rest of the shifting time, just over half a second, is needed to engage the next gear. It turned out that the turbine speed during the filling phase of the shift in process continues with nearly the same slope as before the shift. This means the turbine speed continues to increase almost linear while the output shaft speed increases. The behavior can be seen more clearly in figure 21.

⁸ CANalyzer is a universal tool to record data traffics for bus systems and is delivered by the German company Vector. By Capi-program one can write own programs included in CANalyzer.

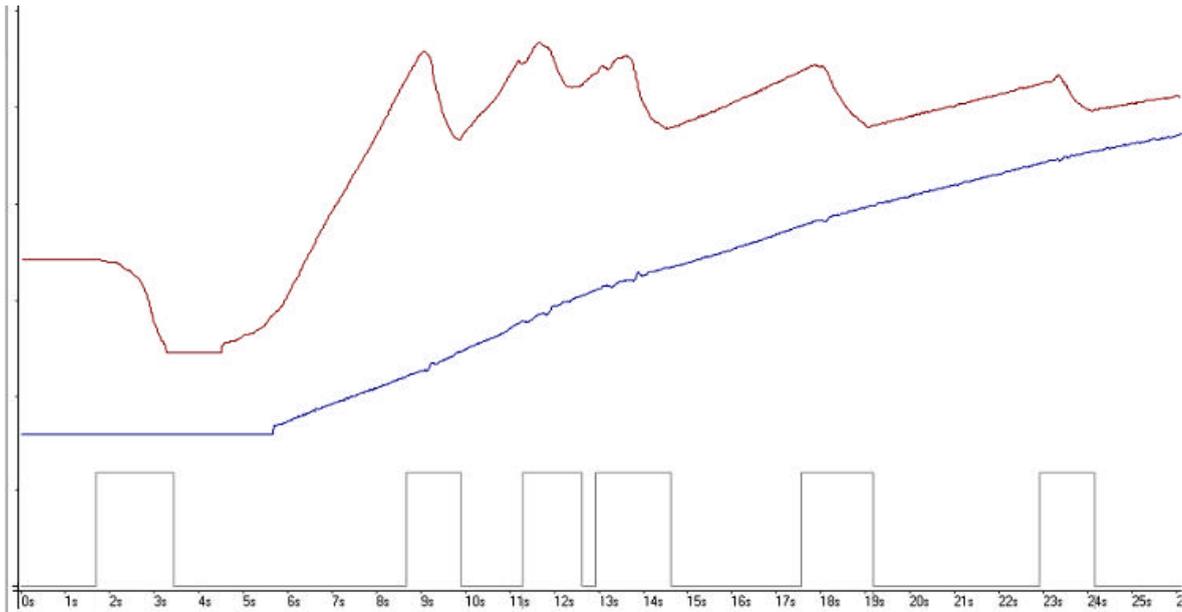


Figure 21. The upper curve represents the turbine speed, the middle curve the output shaft speed and the last signal describes the shift in process signal

Further, the turbine speed is calculated as the product between the gear ratio and the output shaft speed, which means that the turbine speed increases while the output speed is increasing in the first half of the shift process, since the actual gear ratio signal is modeled to be constant. This means the gear ratio signal can be modeled by a 0.5 s dead-zone in the first half of the shift in process.

The measurements gave also information about the second part of the shift process, where it could be established that the turbine speed starts to decrease and decreases until the shift in process reaches the end. This means the gear ratio signal can be modeled by a linear transition to the next gear ratio level durable over about 0.5 s. The adjusted gear ratio signal is therefore ramped up to the next gear ratio during an upshift and ramped down during a downshift.

The current and selected gear signals are used to get the right gear ratio value from which the new signal will be ramped up from and to the value it will reach. These signals are sent from the control unit of the gearbox system and distributed on the CAN network.

When a forward gear range has been selected, the transmission starts in the lowest gear of the range, and as conditions permit, automatic upshifts will appear until the highest gear in the selected range is in use. Therefore to achieve a direct up- or downshift in two steps is impossible, i.e. for example an upshift from gear two to gear four that can be done in a manual gearbox. Hence, to model the gear shifting it is enough to establish a transition between two gear levels and not by arbitrary gear ratio levels. Further, above engine idle neutral-to-range shifts are normally not inhibited. The same limit holds for directional forward to reverse changes that are typically not permitted if an appreciable output shaft speed is detected.

The function of the gearbox is to adjust the gear ratio from the engine to the rest of the driveline to reach a better working range. The torque multiplication in the gearbox module is determined by the gear ratio. The relationship between the input and output shaft speeds is described by:

$$W_{g_in} = n_{gear_ratio} \cdot W_{g_out} \quad (6.12)$$

Where:

- W_{g_in} : input shaft speed of the gearbox.
 W_{g_out} : output shaft speed of the gearbox.
 n_{gear_ratio} : adjusted gear ratio of the gearbox.

The output modules of the gearbox contain components that transmit power and torque to the vehicle driveline. The torque out of the gearbox can be described by:

$$T_{g_out} = n_{gear_ratio} \cdot T_{g_in} \quad (6.13)$$

Where:

- T_{g_in} : input shaft torque of the gearbox.
 T_{g_out} : output shaft torque of the gearbox.

6.3.1 Ratio tests

During each shift the ECU performs tests to check the transmission range and to verify that a shift is being made properly. These tests are the range verification test, the off-going ratio test and the oncoming ratio test.

Range verification is continuously done outside the shift in process. The verification verifies that the current range is attained in range commanded by the ECU. The test checks the current gear ratio, i.e. compares the turbine and the output shaft speed. This speed ratio is then compared to the speed ratio of the range the ECU has commanded, that is stored in memory. If the ratios do not match, a diagnostic code is logged.

The off-going ratio test is performed during the beginning of the shift in process. Within a predetermined time after the shift is commanded, the ECU determines the ratio between the turbine speed and output shaft speed. The speed ratio is then compared to the ratio of the previous range. If the previous range is still present after a period of time, the ECU assumes the off-going clutch did not release. After two attempts with the same condition, a diagnostic code is logged and the transmission is set to the previous range. The off-going ratio test is active between the shift initiation point and the turbine speed pull down detected point.

The ongoing ratio test is performed near the end of the shift in process. This test checks the turbine speed and the output shaft speed to determine if the transmission is in range commanded by the ECU. When the ratio does not match, the ECU assumes the oncoming clutch did not engage and will log a diagnostic code.

6.4 Retarder

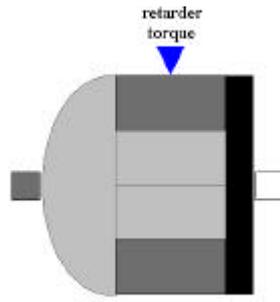


Figure 22. The retarder component

The retarder is an external brake frequently used in trucks. The purpose of the retarder is to assist in stopping the vehicle, assist in controlling vehicle downhill speed on a steep grade (expressed in percent), extend the life of the vehicle service brakes, and enhance vehicle control. Since the retarder is acting on the propeller shaft, the brake is just affecting the driven wheels.

The duty cycle of the retarder is determined by the requested percent retardation, transmission output shaft speed, and the transmission range attained. The ECU controls the retarder capacity. When output shaft speed is within the calibrated range, retarder capacity is allowed its maximum value. When the speed is above or below a calibrated value, the ECU modulates the retarder to a lower capacity. The retarder is deactivated at a calibrated low output shaft speed to make a smooth transition to the service brakes. To avoid large heat load generated by the retarder, the capacity is reduced when the output shaft speed is above same value. The estimated braking torque the retarder produces can be determined from two CAN messages that give information about the actual retarder percent torque and the maximum retarder torque. These two values are used for calculating the braking torque generated by the retarder that is taken as an input to the model as signal *retarder torque*. This gives the following expression for the retarder component:

$$T_{propeller} = T_{g_out} - T_{retarder} \quad (6.14)$$

Where:

T_{g_out} : output shaft torque of the gearbox.

$T_{retarder}$: applied braking retarder torque.

$T_{propeller}$: propeller shaft torque.

The actual amount of retarding torque generated by the retarder is dependent upon two factors:

- The maximum available retarder capacity, which is determined by the transmission and retarder configuration.
- The fraction of maximum retarder capacity which is desired. This represents the driver demand.

The desired retarder capacity is determined by a hand level that allows the driver to select a non-active position or six different levels of retardation. The hand level varies a 5 V signal which is sent to the ECU, and interprets the signal as an amount of requested retardation.

As a result of that the retarder only can apply torque on the propeller shaft, the angular velocity at the left and the right side of the component is the same, consequently:

$$\mathbf{w}_{propeller} = \mathbf{w}_{g_out} \quad (6.15)$$

Where:

\mathbf{w}_{g_out} : output shaft speed of the gearbox.

$\mathbf{w}_{propeller}$: angular velocity of the propeller shaft.

The retarder is automatically disengaged when the anti-lock brake system (ABS) of the vehicle is activated.

6.5 Propeller shaft

The propeller shaft does only exist in rear driven vehicles with the engine in front. This shaft transfers energy from the front end of the vehicle to the rear. The propeller shaft is actually elastic, but to facilitate the model and save simulation time is the shaft considered as stiff.

This means the following equation is valid:

$$T_{propeller} = T_{retarder} \quad (6.16)$$

Where:

$T_{retarder}$: torque driving the propeller shaft.

$T_{propeller}$: torque out of the propeller shaft.

Since the propeller shaft is considered as rigid, the angular velocity is the same along the whole shaft. Therefore the following holds:

$$\dot{\mathbf{w}}_{propeller} = \dot{\mathbf{w}}_{retarder} \quad (6.17)$$

This assumption leads to that no object for the propeller shaft has to be created, and the retarder module can be connected directly to the final gear.

6.6 Final drive

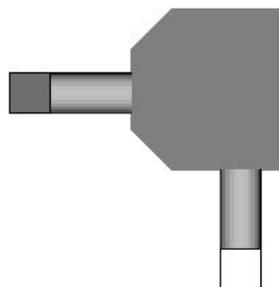


Figure 23. The final drive component

The final drive is a differential gear that allows the driven wheels to rotate with different velocities. This is necessary when the vehicle is driving in curves. The final gear connects the propeller shaft with the drive shafts. The component is modeled without any energy losses in the gear, i.e. the model is considered as ideal.

The ideal final gear has the following properties:

$$T_{final_drive_r} = T_{propeller} \cdot \frac{n_{final_drive}}{2} \quad (6.18)$$

$$T_{final_drive_l} = T_{final_drive_r} \quad (6.19)$$

Where:

$T_{final_drive_l}$: torque acting on the left drive shaft.

$T_{final_drive_r}$: torque acting on the right drive shaft.

n_{final_drive} : gear ratio of the final drive.

The gear ratio gives also the equation for the angular velocity:

$$\frac{\omega_{propeller}}{n_{final_drive}} = \frac{(\omega_{final_drive_l} + \omega_{final_drive_r})}{2} \quad (6.20)$$

Where:

$\omega_{propeller}$: angular velocity of the propeller shaft.

$\omega_{final_drive_l}$: angular velocity of the left final drive output.

$\omega_{final_drive_r}$: angular velocity of the right final drive output.

6.7 Drive shaft

The drive shaft represents the connection between the final gear and the wheels. This shaft is like the propeller shaft, assumed to be stiff without flexibility. This is done to save simulation time. Therefore no model for the drive shaft is created and the following can be expressed:

$$T_{drive_shaft_l} = T_{final_drive_l} \quad (6.21)$$

$$T_{drive_shaft_r} = T_{final_drive_r} \quad (6.22)$$

Where:

$T_{drive_shaft_l}$: torque acting on the left drive shaft.

$T_{drive_shaft_r}$: torque acting on the right drive shaft.

Since the propeller shaft is considered as rigid the angular velocity is the same along the whole shaft, the following equations can be written:

$$\dot{\omega}_{drive_shaft_l} = \dot{\omega}_{final_shaft_l} \quad (6.23)$$

$$\dot{\mathbf{W}}_{drive_shaft_r} = \dot{\mathbf{W}}_{final_shaft_r} \quad (6.24)$$

Where:

$\dot{\mathbf{W}}_{drive_shaft_l}$: angular velocity of the left drive shaft.

$\dot{\mathbf{W}}_{drive_shaft_r}$: angular velocity of the right drive shaft.

6.8 Brakes

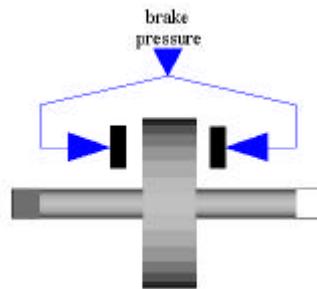


Figure 24. The brake component

To be able to stop the vehicle, the simulation model is equipped with brakes. The brake component is reused from a previously performed master thesis project, reference [5]. Further, the module is built up by a standard component from the available Dymola libraries. The brake has been parameterised to attain the characteristics of a real brake. One brake module is located at each wheel. Since the final shaft is considered as rigid, the rear brakes can be connected directly to the final drive.

The wheel brake model describes a disc brake consisting of brake pads. The component is used to get a braking torque on the wheel hub. This torque arises from the friction that comes up between the brake pads and the wheel. The size of the braking torque is depending on the applied braking pressure. To generate a braking torque, the brake pads have to be pressed against the wheel discs with a normal force. Therefore the braking pressure is used as an input to the model. This pressure is estimated in the control unit of the brake system. Unfortunately the pressure is not sent out on the CAN network and therefore not either available as an ordinary CAN message. The problem is solved by reading the pressure directly from the control unit of the brake system, through a connected breakout box. Then the internal brake pressure can be sent to the truck model by using an extra CAN input interface to the dSPACE hardware.

To generate a braking torque the braking pressure has to become higher than a collaring pressure, this depends on the physics of a brake system. To model the effects of a collaring pressure a dead-zone for the normal force is implemented, which can be described by:

$$F_{b-n} = \begin{cases} 0 & \text{if } p < p_c \\ k \cdot (p - p_c) & \text{if } p > p_c \end{cases} \quad (6.25)$$

Where:

k : constant.

p : brake pressure.
 p_c : collaring pressure.

This means that the normal force is considered as zero when the braking pressure is below the collaring pressure and above this value the force is increasing linear. As a consequence of this, the torque increases linear when the pressure has been reached. The figure below shows the behavior of the braking torque.

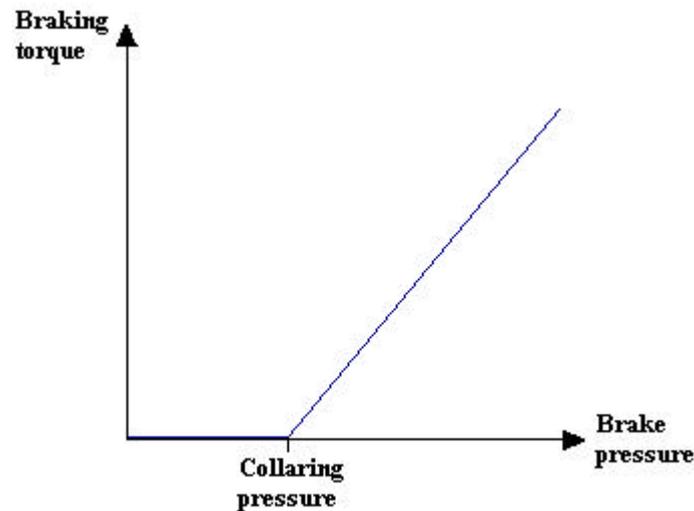


Figure 25. Braking torque as a function of applied braking pressure

The brake model is using a component from the Modelica Standard Library called Friction Base. By this component it is possible to simulate the model both with locked and non-locked wheels. The friction base component is further discussed in appendix C.

When the absolute angular velocity is not zero, the friction torque is a function of the normal force, of a velocity dependent friction coefficient, and of a geometry constant of the braking discs which takes into account the geometry and the assumptions on friction distributions. For the brake system with non-locked wheels the braking torque is given by:

$$T_b = c_{geo} \cdot \mathbf{m}(\mathbf{w}_w) \cdot F_{b_n} \quad (6.26)$$

Where:

c_{geo} : a geometric constant of the braking disc.
 \mathbf{m}_b : frictional coefficient between the brake discs and the wheel hub.
 \mathbf{w}_w : angular velocity of the wheel.
 F_{b_n} : normal force that presses the wheel pads against the wheel.

When the wheels are locked the braking torque is adjusted so that the angular acceleration of the wheel is zero.

6.9 Wheels

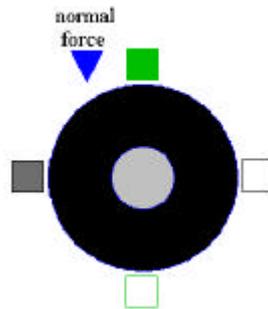


Figure 26. The wheel component

The wheels are the part of the driveline that enforces the contact with the roadway surface. In the contact area the motion is transformed from rotational to translational. The model contains both the wheel and the tire.

To get the component reacting dynamically, the model has to be modeled by an inertia. The tire is assumed to be rigidly connected to the wheel, actually to the wheel rim. This assumption leads to that the inertia of wheel and the tire can be brought together and treated as one inertia.

Since the rear wheels in the model are located to the final shaft through the brake model, the torque acting on the rear wheel hub is the same as the final shaft torque minus possible braking torque. In other words:

$$T_{w_l} = T_{drive_shaft_l} - T_b \quad (6.27)$$

$$T_{w_r} = T_{drive_shaft_r} - T_b \quad (6.28)$$

Where:

T_{w_l} : torque acting on the left rear wheel hub.

T_{w_r} : torque acting on the right rear wheel hub.

The front wheels are not driven by any drive shaft. Therefore the torque acting on the front wheel hub is zero when the front brakes are not engaged. Otherwise, when the front wheel brakes are engaged, then a braking torque is applied that determines the torque acting on the wheel hub:

$$T_w = T_{front_wheel_brake} \quad (6.29)$$

Where:

$T_{front_wheel_brake}$: braking torque generated by the front wheel brake.

The forces and torques acting on the wheel can be seen in figure 27. These quantities are influencing the angular velocity of the wheel.

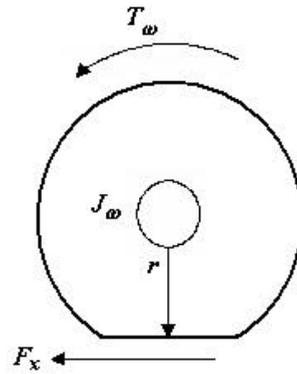


Figure 27. Forces and torques acting on a wheel

From the figure above it can be seen that the rolling radius is the distance from the wheel hub to the contact area against the ground. Therefore a loaded wheel has a shorter radius than an unloaded as the tire slightly gets flattened in the contact zone.

The movement of the wheels in longitudinal direction can be described by an ordinary rotating equation:

$$F_x \cdot r = T_w - J_w \cdot \dot{\omega}_w \quad (6.30)$$

Where:

- F_x : tire force.
- r : rolling radius.
- T_w : torque acting on the wheel hub.
- J_w : wheel and tire inertia.
- $\dot{\omega}_w$: angular velocity of wheel.

To solve equation (6.30) the tire force has to be known. Due to non-linear circumstances the tire forces and tire models are a difficult area within the dynamics of vehicles. Therefore the theory about the tire force is introduced and briefly treated in section 7.

The rolling resistance force acting in an opposite direction to the tire force. Therefore the traction force delivered to the chassis component can be described by:

$$F_{traction} = F_x - F_{rr} \quad (6.31)$$

Where:

- $F_{traction}$: traction force .
- F_{rr} : rolling resistance force.

6.9.1 Rolling resistance

Rolling resistance is caused by elastic deformation of a tire. Primary source of this resistance is the deformation of the pneumatic tires that is generated as they pass over the roadway

surface. The circular shape of a tire will be flattened out in the contact area, if the tire is loaded. As the wheel is rolling new deformation zones will continuously arise at the same moment as deformed tire elements will get back their ordinary shape. The outcome of this deformation are energy losses that usually are called rolling resistance.

Tire manufacturer Michelin has suggested the following formula for determining the rolling resistance, given from [6]:

$$C_r = C_{r_iso} + C_a \cdot ((v \cdot 3.6)^2 - 80^2) + C_b \cdot ((v \cdot 3.6) - 80) \quad (6.32)$$

Where:

C_r, C_{r_iso}, C_a, C_b : coefficients of rolling resistance.
 v : truck velocity.

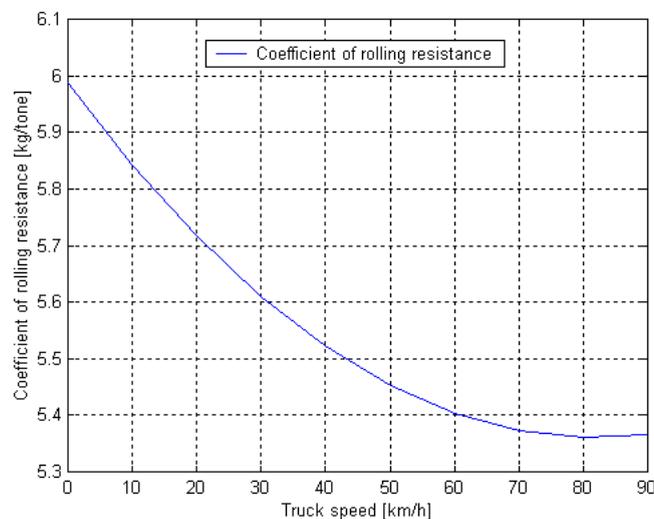


Figure 28. Coefficient of rolling resistance as a function of truck speed

The coefficient of rolling resistance, C_r , is dependent on many variables like tire pressure, tire temperature, vehicle speed, road conditions and wheel adjustments.

The rolling resistance is proportional to the normal force acting on the wheel. This means that the rolling resistance force is not the same for all wheels on a truck. Therefore to calculate the rolling resistance force, the normal forces acting on each wheel have to be known. In section 6.10.3 these calculations are done. Further the normal forces acting on left and right wheels are assumed to be equal. The simplification is only done for forces acting on the same axle, which means that the front and the rear normal forces are not assumed to be equal. The rolling resistance can be expressed by (taken from [6]):

$$F_{rr} = \frac{C_r}{1000} \cdot F_z \cdot \text{sign}(w_w) \quad (6.33)$$

Where:

F_{rr} : rolling resistance force.
 F_z : normal force acting on the wheel.

w_w : angular velocity of the wheel.

From equation (6.33) it can be seen that the rolling resistance is equal to zero when a wheel not rotates. Another result of the tire deformation is that the acting point of the normal force is dislocated due to the higher pressure in the front part of the wheel. Therefore the acting point of the normal force is moved against the front edge of the wheel. Figure 29 shows how the resistance force and the dislocated normal force cause a torque on the wheel hub, defined by reference [8]. As the figure shows are these two generated torques acting in opposite directions and according to this reference can they be assigned the same value.

$$F_{rr} \cdot r = F_z \cdot e \quad (6.34)$$

Where:

e : distance to the dislocated acting point of the normal force.

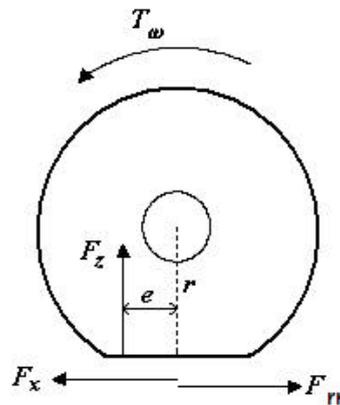


Figure 29. Forces and torques acting on the wheels

Equation (6.33) and (6.34) gives the information that the rolling resistance is not explicitly affecting the rotational speed of the wheel. However the resistance force is acting in a direction that slows down the speed of the truck. This will cause a higher slip value and hence also increase the tire force. So, the result of the rolling resistance is therefore a part of the tire force, which according to equation (6.30) affects the rotational speed of the wheels.

6.10 Chassis

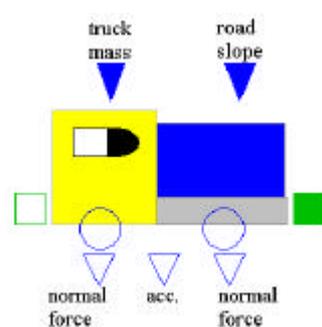


Figure 30. The chassis component

A vehicle can move along only if the sum of all resistances is overcome. Resistances are forces that either balance out the driving force (uniform movement) or are smaller than the driving force (accelerated movement).

In the chassis component all forces from the wheels are added up. This means that the tire forces generated by the rear wheels are giving a positive contribution to the forces that affect the vehicle speed, when the truck moves forward. The front wheels give also a force contribution to the chassis, but this part is negative when the truck accelerates, since the force is affected by rolling resistance and of a possible braking torque.

The chassis are beside forces from the wheels affected by gravitational force, air drag resistance and forces due to the road slope.

To be able to simulate the truck movements on an arbitrary road a signal called *road_slope* is used as an input to this component. By this signal it is possible to change to road topography in real time by changing the road slope. The slope is expressed in percent, positive for uphill and negative for downhill. The generated force as a result of the road slope:

$$F_{slope} = m \cdot g \cdot \sin(\mathbf{a}), \text{ where } \mathbf{a} \text{ is expressed by: } \mathbf{a} = (\arctan(\text{slope} / 100)) \quad (6.35)$$

Where:

- F_{slope} : gravity force due to the road slope.
- m : mass of the vehicle.
- g : gravity on earth.
- $slope$: slope of the road.

Therefore, when driving in an uphill slope the generated gravity force is positive, which means the force acts in a counteracting direction compared to the vehicle movement.

Finally, in the chassis model, the velocity of the vehicle is determined from an ordinary differential equation based on Newton's second law.

6.10.1 Air drag

Air resistance occurs as aerodynamic phenomena affecting the chassis body. The braking force as a result of the air resistance is usually described as a function proportional to the squared velocity. In [7] the air resistance is described by:

$$F_{ar} = c_D \cdot A \cdot \frac{\mathbf{r}}{2} \cdot v_d^2 \quad (6.36)$$

Where:

- c_D : a dimensionless aerodynamic constant.
- A : frontal area of the vehicle.
- \mathbf{r} : mass density of the air.
- v_d : difference between the vehicle velocity and the wind velocity.

In this model the wind velocity is neglected because during normal wind condition this velocity is small compared to the vehicle speed. Further, there is nearly impossible to exactly know the wind direction all the time, i.e. have knowledge of the incoming wind angle to the truck. Therefore the following simplified equation is used:

$$F_{ar} = c_D \cdot A \cdot \frac{\rho}{2} \cdot v^2 \quad (6.37)$$

Where:

v : velocity of the truck.

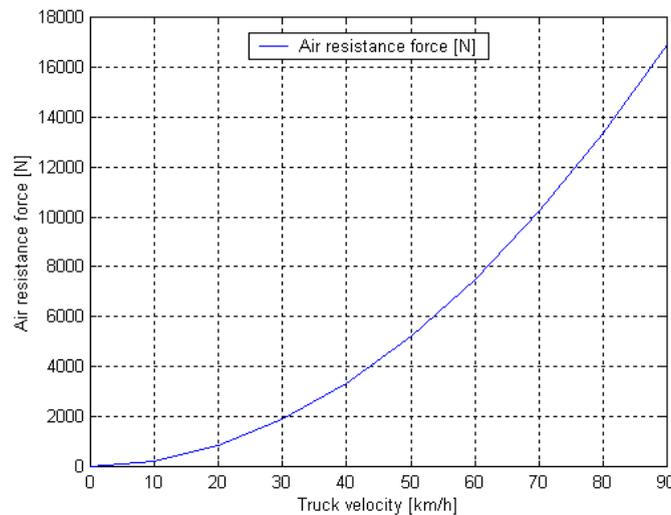


Figure 31. Air resistance as a function of truck velocity

6.10.2 Acceleration of the truck

To calculate the change in the vehicle speed, this can either be done based on energy calculations, i.e. looking at the generated net energy, or by solving a differential equation where all forces are summarized that for a given mass results in a change in the velocity. The later solving method is used in this project, thanks to the very convenient handling of differential equations in Modelica. Therefore the vehicle velocity is calculated from the differential equation based on Newton's II law that is described by:

$$m \cdot \dot{v} = \sum F_{driving_tot} - F_{slope} - F_{ar} \quad (6.38)$$

Where:

m : mass of the truck.

\dot{v} : acceleration of the truck.

$\sum F_{driving_tot}$: driving force generated by all wheels.

F_{slope} : generated longitudinal force as a result of road slope.

F_{ar} : air resistance force.

The velocity of the truck is determined by an ordinary integration of the acceleration over the time. This is done by a component called *accelerate* given from the Modelica Standard Library.

6.10.3 Axle forces

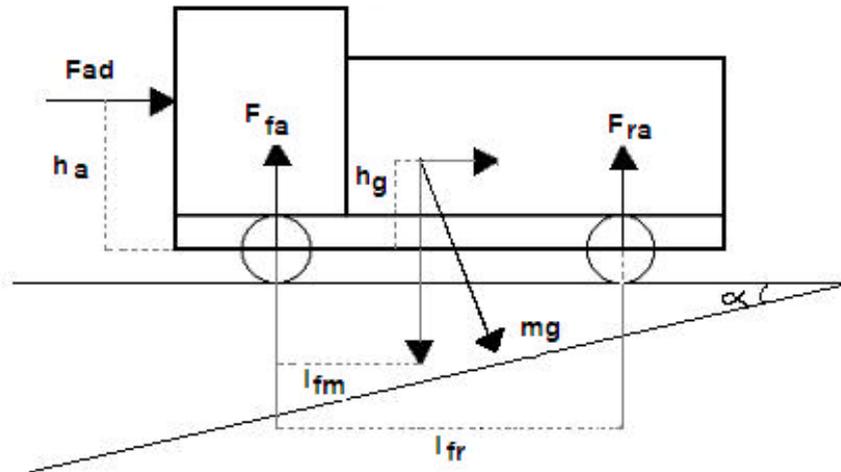


Figure 32. Forces acting on the chassis

Moment theorems are used to distribute the weight of the truck to the front and rear axles. To be able to calculate the tire force and the rolling resistance force, the load on each tire has to be given. The normal forces are determined by torque balance equations.

$$F_{fa} \cdot l_{fr} = (l_{fr} - l_{fm}) \cdot m \cdot g \cdot \cos(\mathbf{a}) - h_g \cdot m \cdot g \cdot \sin(\mathbf{a}) - F_{ad} \cdot h_a \quad (6.39)$$

$$F_{ra} \cdot l_{fr} = l_{fm} \cdot m \cdot g \cdot \cos(\mathbf{a}) + h_g \cdot m \cdot g \cdot \sin(\mathbf{a}) + F_{ad} \cdot h_a \quad (6.40)$$

Where:

- F_{fa} : normal force acting on the front wheels.
- F_{ra} : normal force acting on the rear wheels.
- F_{ad} : air drag force acting on the frontal area of the vehicle.
- l_{fr} : distance from front axle to rear axle.
- l_{fm} : distance from front axle to mass center point.
- h_g : vertical distance to working point of the gravity force.
- h_a : height to the working point of the air drag force.
- m : truck mass.
- g : gravity on earth.
- \mathbf{a} : road slope.

When a truck has more than one rear axle, the normal force acting on each axle becomes lower since this gives another weight distribution. Then the normal forces acting on the rear axles are assumed to be equal. Therefore from equations (6.39) and (6.40) can the front and rear axle normal forces be evaluated:

$$F_{front_axle} = \frac{(l_{fr} - l_{fm}) \cdot m \cdot g \cdot \cos(\mathbf{a}) - h_g \cdot m \cdot g \cdot \sin(\mathbf{a}) - F_{ad} \cdot h_a}{l_{fr}} \quad (6.41)$$

$$F_{rear_axle} = \frac{l_{fm} \cdot m \cdot g \cdot \cos(\mathbf{a}) + h_g \cdot m \cdot g \cdot \sin(\mathbf{a}) + F_{ad} \cdot h_a}{l_{fr} \cdot n_{axles}} \quad (6.42)$$

Where:

n_{axles} : number of rear axles.

To calculate the normal force acting on each individual wheel the normal force has to be divided by the number of wheels located at each axle. Therefore:

$$F_{normal_front_wheel} = \frac{F_{fa}}{n_{front_wheels}} \quad (6.43)$$

$$F_{normal_rear_wheel} = \frac{F_{ra}}{n_{rear_wheels}} \quad (6.44)$$

Where:

$F_{normal_front_wheel}$: normal force acting of each front wheel.

$F_{normal_rear_wheel}$: normal force acting of each rear wheel.

n_{front_wheels} : number of wheels per front axle.

n_{rear_wheels} : number of wheels per rear axle.

The normal forces acting on each wheel are sent to the wheel module where the rolling resistance can be calculated.

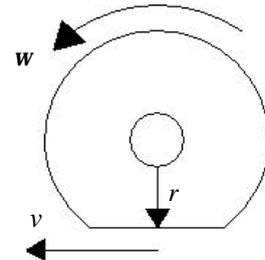
7 Modeling of tires

In this section, the theory behind the tire modeling will be treated. First the longitudinal tire force is introduced and then the traditional tire modeling and its limits are discussed. In the end of this section the implemented tire model is briefly discussed. This model considers the tire as flexible. Since the model was developed during an earlier master thesis project and not in this project, all mathematical proofs according to the model will not be treated here. These can be read in [5].

7.1 Tire model for slip

To be able to change the velocity of the vehicle, forces have to be generated by the tires. To generate a force in longitudinal direction the wheel must rotate with a speed that differs from the vehicle. The speed difference is usually described by a variable called slip. The vehicle organization SAE uses the following expression for slip:

$$s = \frac{r \cdot \omega - v}{v}$$



(7.1)

From the definition above, the slip variable can be established as dimensionless. The slip is further not defined at stand still, as then singularity arises. In vehicle simulations this can give rise to problems, when simulations have to run from stand still.

During braking the wheel rotates slower and the rubber elements that first hit the ground will be stretched more and more. The slip value is, according to the definition in equation (7.1), negative during braking since the translational velocity is higher than the corresponding rotational velocity. Further, a free rolling wheel has a tire slip value equal to zero. Therefore during braking the slip value is varying in the interval from 0 to -1 (locked wheel). A free rolling wheel, i.e. non-driven wheel, rotates with the same speed as the truck in steady state. Thus the slip value is equal to zero.

The longitudinal tire force that is generated as a result of the friction characteristics between the tire and the ground can be seen in figure 33 as a function of the slip. The function is non-linear, but can for small slip values be considered as almost linear. When the wheels start to spin, the tire force is reduced and is getting nearer a steady state value.

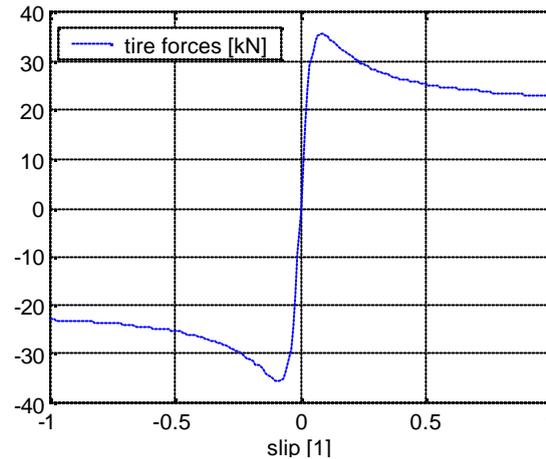


Figure 33. Slip curve, describing the generated tire force as function of slip, given from [5]

The longitudinal force arises as the wheel has an angular velocity that differs from a free rolling wheel. Since sliding normally do not occur, the speed difference between the wheel and the truck will make the rubber in the tire tread stretched. As mentioned previously, during braking the wheel rotates slower and the rubber elements that first hit the surface will be stretched more and more. Further, when the wheels rotate faster than the vehicle then the tire treads at the end of the contact area will be most stretched.

This leads to that a force, due to the stiffness of the rubber, will be generated in the tire. Figure 34 shows the stretching phenomena of the tire elements in the contact area, where the vertical lines illustrate how much the rubber locally is stretched. It can be seen that the tire elements get more and more stretched towards the rear end of the contact area. This behavior leads to that the force generated in the tire is not uniformly distributed over the contact area.

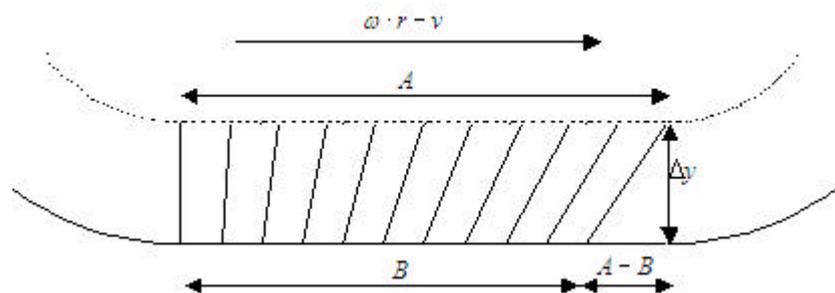


Figure 34. Principle sketch of the stretching behavior of a tire, given from [5]

For a large slip value, i.e. when the speed difference between the wheels and the vehicle is too big, the forces at the rear end of the contact zone will exceed the static frictional forces that keep the tires stuck with the road. This leads to that the rear parts of the tire in the contact area will begin to slide. The sliding parts will still generate a tire force but the magnitude of that is reduced. The reason to this, is that kinetic frictional forces normally are smaller than static frictional forces.

A non-driven wheel rotates in steady state with the same speed as the vehicle. This leads to, as mentioned before, that a free rolling wheel has a slip value equal to zero. Due to this, no parts of the tire are stretched and therefore no tire forces are generated.

7.2 Traditional tire modeling

In traditional tire models the longitudinal force, i.e. the tire force, is usually described as a function depending on both the longitudinal slip and the normal force acting on the wheel. This function can be expressed by:

$$F = F(s) = N \cdot \mathbf{m}(s) \quad (7.2)$$

Where:

N : normal force acting on the wheel.
 $\mathbf{m}(s)$: friction coefficient as a function of slip.

Traditional tire models are often used in simulations to first and foremost simulate the vehicle during braking. Since equation (7.2) is built up from the slip definition, it is not possible to calculate the tire force at stand still, due to the singularity. However, all other simulation cases are possible to run. Scania has in previously performed simulations used a traditional tire model, called Magic formula. More information about the Magic formula can be seen in appendix D. In these simulations three different tire types are used with various friction coefficients. These data has been used to calculate the longitudinal tire force.

7.3 Low speed problems

With the traditional tire model and a time discrete solver, like Eulers forward method, non-acceptable behavior arises at low speed. This can be visualized by considering an example with one mass and one wheel. Further, the slip will be assumed to be small and can according to figure 33 be treated as linear. This means the tire force can be described by:

$$F_x = c \cdot \mathbf{s} \quad (7.3)$$

Where:

c : a constant, defining the slope in the tire force function.

To describe the motion of the wheel and the vehicle, two dynamical equations have to be expressed:

$$\dot{\mathbf{w}}(t) = \frac{T_w(t)}{J} - \frac{r}{J} \cdot F_x(\mathbf{s}) \quad (7.4)$$

$$\dot{v}(t) = \frac{F_x(\mathbf{s})}{m} \quad (7.5)$$

By using expression (7.4) and (7.5) together with the slip definition the following equations can be stated:

$$\dot{\mathbf{w}}(t) = \frac{T_w(t)}{J} - \frac{r}{J} \cdot c \cdot \frac{r \cdot \mathbf{w} - v}{v} \quad (7.6)$$

$$\dot{v}(t) = \frac{c}{m} \cdot \frac{r \cdot \mathbf{w} - v}{v}$$

To determine the stability of this non-linear system, it has to be expressed with a Taylor expansion of first order around steady state points. Natural choices of steady state point can be:

$$v = v_0 \quad (7.7)$$

$$\mathbf{w} = \mathbf{w}_0 = \frac{v_0}{r} \quad (7.8)$$

$$T_w = T_{w0} = 0 \quad (7.9)$$

With the steady state points the Taylor expansion can be written:

$$\dot{\mathbf{w}}(t) \approx -\frac{c \cdot r^2}{J \cdot v_0} \cdot \mathbf{w}(t) + \frac{c \cdot r}{J \cdot v_0} \cdot v(t) + \frac{1}{J} \cdot T_w \quad (7.10)$$

$$\dot{v}(t) \approx \frac{c \cdot r^2}{J \cdot v_0} \cdot \mathbf{w}(t) - \frac{c}{m \cdot v_0} \cdot v(t) \quad (7.11)$$

The approximated system is linear and can therefore be Laplace transformed, to be able to determine the natural frequencies of the system:

$$s_1 = 0 \quad (7.12)$$

$$s_2 = -\frac{c}{v_0} \cdot \left(\frac{r^2}{J} + \frac{1}{m} \right) \quad (7.13)$$

The expressions above show that the system is stable for all possible linearisation points. On the other hand, the second natural frequency is increasing when the system is linearised at a small value of the vehicle velocity. By using Eulers forward method the natural frequencies of the time continuous system will be transformed from the S-plane to the Z-plane by:

$$z = 1 + s \cdot T \quad (7.14)$$

Where:

z : time discrete natural frequency.

T : fixed step time of the solver.

For a time discrete solver the poles in the Z-plane must be located to the interior of the unit circle, otherwise the solution to the differential equation will not be stable. Therefore by checking the location of the poles system, can it be established that the pole will move to the left when the velocity is reduced. For normal truck parameters and a sampling frequency of 1 kHz the natural frequency for high velocities will be located in the right half plane of the unit circle.

However, for velocities about 30 km/h the discrete pole will cross the imaginary axle, and for velocities below 15 km/h, the natural frequency will enter the outside of the unit circle, which leads to that the velocity becomes unstable. It is therefore not possible to simulate the vehicle from stand still by using a traditional tire model. This is the reason to why the traditional tire model cannot be used directly.

7.4 Dynamic tire force

The low speed problems are solved by including the dynamics of the tire force, so that the tire not becomes too stiff at low speeds. This solution refers to a real physical behavior, since the tire is not so stiff as treated in traditional tire models.

The tire force is then described by a first order system with low pass characteristics in the frequency domain. This consideration leads to that the fast frequencies of the system are filtered. By choosing a small time step of the numerical solver, no oscillations in the tire force will occur.

However, the slip value that determines the tire force is still not defined at speed equal to zero. To solve this problem a new slip definition is used. This definition is given by:

$$\mathbf{s} = \frac{A - B}{B} \quad (7.15)$$

Where:

- A : distance an undeformed tire element has to roll to obtain the desired force.
- B : describes the distance a deformed tire element that follows the road, has to roll to generate the desired force.

In figure 34 the distances A and B are shown. This slip definition describes how much the tire has to be stretched to generate the desired force. The problem is to determine the distance B at any time for the desired tire force value. This distance is however not easy to determine, since it is hard to measure the length of a stretched tire element. To get away from determining the distance B , the tire tread has to be modeled as flexible, which is solved by a spring and damper model.

By introducing a spring in the contact area, the model is given a new state variable, x , that macroscopically describes how much the tire has to be stretched to obtain the desired tire force. Further, a damper is implemented to avoid oscillations in the system at low speeds. The damping effect is also physical, since the rubber cannot be stretched without transforming some of the energy into heat.

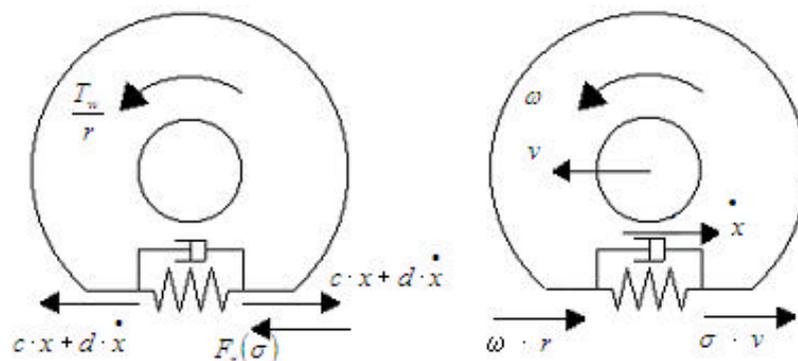


Figure 35. Forces and velocities acting in the spring and damper model. The figure is given from [5]

Figure 35 shows the forces and the velocities that appear in the contact area when the tire tread is modeled with a spring and a damper. The damper is modeled parallel to the spring, since heat energy is only generated when the rubber is stretched.

If the velocities in the contact area are summed up, the time derivative of x is given by:

$$\dot{x} = \mathbf{w} \cdot \mathbf{r} - v - \mathbf{s} \cdot \mathbf{v} \quad (7.16)$$

When the tire treads are totally stretched, i.e. in steady state, the equation above is exactly the same as the slip definition in equation (7.1). This means the traditional tire force function in (7.2) together with this tire model can be used to determine the tire force at each time instant. Further, equation (7.16) is also used to calculate the size of the stretching.

The slip value is determined, by using the fact that the force generated by the spring and the damper must equal the force given by the tire force function in equation (7.2), i.e.:

$$F_x(\mathbf{s}) = c \cdot x(t) + d \cdot \dot{x}(t)$$

Where:

- c : stiffness of the longitudinal spring.
- d : damping constant of the tire.

8 Simulations

Real-time simulations are done to see that the model works as expected and communicate with the control units in the test lab. The primary purpose of this master thesis project has not been to simulate different driving situations in the lab. The important task has been to be able to run the automatic gearbox system in the lab.

The simulated speed signals generated from the truck model are received by control units as then processes these signals and send out CAN messages on the bus network. By checking the CAN signals such as current gear, selected gear, actual gear ratio, shift in process etc, the status of the automatic gearbox system can be verified. The CAN messages sent from the automatic gearbox system give therefore information about how the system interprets the simulated speed signals. If suitable and synchronized speed signals are received by the ECU of the gearbox system, then the ECU is able to command a shift to be performed. To fulfill a shift the speed signals have to be correctly related to each other through the whole gear shifting, otherwise would error frames be generated and no shift be performed.

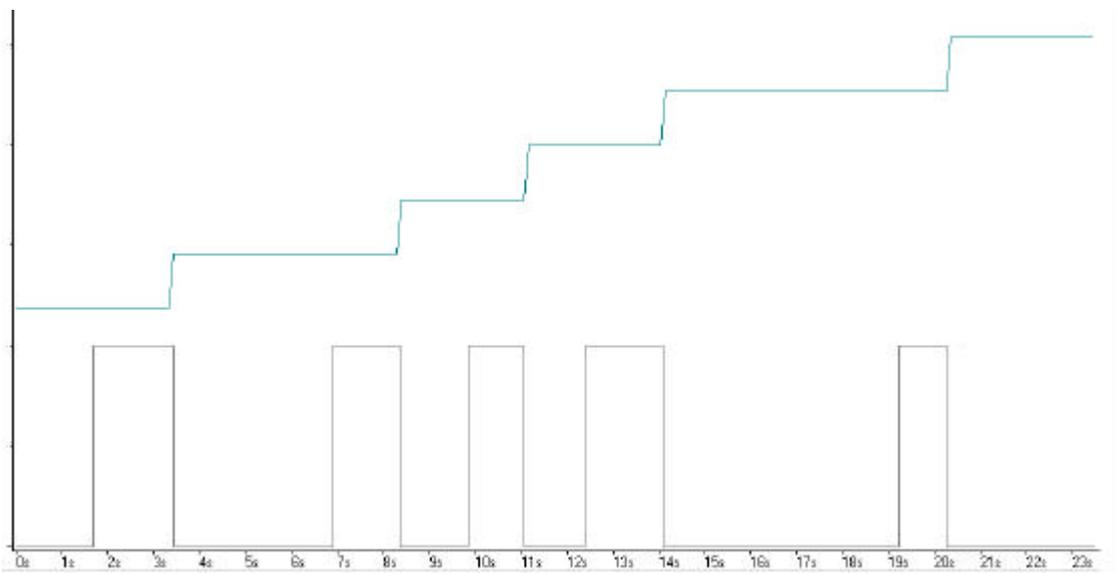


Figure 36. A simulation that shows an upshift to the fifth gear. The upper signal represents the current gear and the lower signal describes the shift in process signal

By running the developed simulation model automatic gear shifts can be performed. The simulation is controlled by an accelerator pedal actuator, a real brake pedal and a retarder hand level. When the simulation is started and the ignition key is turned on, the engine speed is given an idling speed and the speed signals of the transmission are set to zero. Then by putting the gear level selector in drive position and increase the accelerator pedal position, gear shifts will automatically be performed. The time, at which each shift will be initialised, depends on the values of the speed signals and the relationship between them. The simulation can be run from stand still up to the highest speed domains by controlling the level of the three actuators (mentioned above). By using the brake pedal the truck speed can be reduced all the way down to zero.

9 Conclusion

By this project it has shown possible to create a modular model of a driveline with an automatic gearbox. The modeling language Modelica has been used for the modeling work and Simulink was used as the interface to the hardware. By using a specific computer system, delivered by dSPACE, was it possible to run simulations in real-time and communicate with control units located in a test lab.

The main work has been put into modeling of the engine, the converter and the gearbox. Since the converter acts as a fluid coupling, there are no obvious relationships between the engine speed and the turbine speed of the converter. Further, there is nearly impossible to make physical calculations of a converter in operation, i.e. when the engine and the transmission not stand still. This means it is difficult to physically model the performance of the converter. Measurements of a real converter during running are utilized to state up empirical formulas of the converter's capability to absorb and transmit torque. By these measurements the problem with modeling the converter was solved.

By using the developed model has it been possible to simulate a normal run of a truck with an automatic gearbox system in a laboratory. Further, by running the simulation a majority of variables can be displayed in real time, for example torques and velocities from different parts of the driveline that cannot be measured in a truck. This can be used for a wide variety of purposes. By application of the model several future test cases that demands realistically simulated speed signals will be able to be checked in the laboratory, which will save a lot of time.

In this moment it is only possible to simulate the truck in real-time with one front and one rear wheel. During simulation of the truck model with four wheels, the turnaround time of the solver is about 15 % longer than the sample time. The reason to the problem is not further surveyed, which demands more knowledge about the translation process of the models.

10 Future work

It is possible to further develop modules of the truck model, for example enhance the model with a propeller shaft model and a model for the final shaft. These axles can be modeled with springs and dampers, which means they can be seen as elastic and not like in this project where they were considered as rigid. The final gear module can also be further developed, for example by adding a gear efficiency to the ideal gear, to model energy losses.

There can in the future be of interest to run simulations of a truck with automatic transmission that contains four wheels, where the right and the left wheels rotate with different speeds. Such cases can appear for example during braking in a curve. To simulate these cases the problem with the turnaround time of the solver has to be solved, so that real-time simulations of a model with more than two wheels can be run.

No particular model validation work has been done, since many parameter values especially the one according to the brake and the chassis model, have been inherited from previously developed models at Scania, where they have been validated. However, there can for example be of interest to determine the acceleration of the vehicle in the laboratory and compare it to a real truck test, to see the degree of agreement. Such test demands close preparations to achieve two similar driving situations.

A future extension of the model that can be useful, is to let the simulation be controlled by a predetermined road topography with a certain accelerator pedal position, i.e. recorded from for example a part of Scania's test track. Then it would be possible to exactly compare, among other things, when gear shifting occurs, the time needed for each shift and the speed from different parts of the driveline. Hopefully these results will agree fairly well and then the model will be able to deliver useful information about the amount of torque generated in the driveline.

Another future work that can be done is to modify the model so it fits other automatic gearbox systems. This can be done since the behavior of automatic gearboxes intended for trucks and buses are usually the same. Therefore by changing the absorption characteristics and the gear ratios, a new automatic gearbox system can be built up. Since the bus and truck development at Scania nowadays have a close cooperation, can it be possible to run the model for other automatic gearboxes that are used in Scania buses. An automatic gearbox system intended for buses will soon be installed in the test lab, and therefore to be able to run all test cases according to the system, there is demand for a dynamic model of the driveline. Further, the model of the converter can be adapted to fit other types of fluid couplings, that in the future can be asked to be simulated.

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TORQ_TQ_GROSSTORQUE_MC.

Appendix A Controller Area Network

CAN stands for Controller Area Network and is a serial bus system that was introduced in February 1986 at a SAE congress. It is a communication protocol that is optimized for real-time control. CAN supports communication between Electronic Control Units (ECUs) connected to the network, linking together the different systems. The protocol is designed to handle short messages and offers a high degree of reliability.

Each message contains of eight hexadecimal digits with different purpose. The first two digits decide the message's priority on the bus. The next four digits represent the Parameter Group Number, which is unique for each message. The last two digits of the ID represent the message's source address, i.e. from which node the message is sent.

CAN uses a protocol called Carrier Sense Multiple Access / Collision Avoidance. The communication is based on broadcasting. This means that each node on the network listen to all traffic on the bus. Further, a message is identified by the information it contains, and on the basis of the received identifier each node decides whether it should process the message or not. This means the protocol can be seen as message oriented instead of address oriented. Since each message is specified with its unique ID and not specified by addresses, this facilitates the possibilities to change the number of nodes in the network, i.e. add or remove a node. However, the message's source addresses from the transmitting ECUs are always a part of its ID.

The messages are transmitted with different time intervals depending on its content and need of responses. If there is a need for a fast response, a short transmission interval is used. The interval times are varying between 10 ms and 10s. The message interval times are specified in the software of each transmitting ECU.

When hardware of the control system is extended in vehicles, it is important to have efficient communication between different parts of the system, i.e. between ECUs, actuators and sensors. In modern trucks today data buses are used for serial communication. The advantages of using data bus instead of a pure point-to-point wire connection are several:

- Fewer wires in the vehicles save cost, weight and space.
- Increased dependability due to fewer connection points and automatic transmission error detect.
- Easier to extend new functions that use signals from other control units.
- The communication enables diagnostic tests.

The messages contain the following parts; a priority control, a data field and an error handling filed. Below in figure 29 a sketch of a message frame can be seen.



Figure A1. The contents of a message frame

The shortenings above have the following meanings:

SOF = Start Of Frame.

ID = Identifier, these bits decide the node of highest priority.

DLC = Data Length Code, decide the size of the data field.

DATA = Transmitted data.

CRC = Checksum-control. Controls if something has gone wrong during the transmission.

ACK = Are sent by all nodes when a message is received without error in the checksum.

EOF = End of frame.

The ID field contains information about the message's priority on the network. In the situation where more than one node sends at the same time, the node with the highest priority will continue to send. The less prioritized message has to wait for its turn to be transmitted.

The data field contains up to 64 bits and the contents are defined by the transmitting ECU. The number of bits in the data field is constant, but its values can be changed.

The last parts of a message frame are used for error detect and ACK is used as an acknowledgement of correctly received messages.

Appendix B Pulse Width Modulation

Pulse Width Modulation (PWM) is a powerful technique for controlling analog circuits with a processor's digital outputs. In other words the PWM is a way of digitally encoding analog signal levels. The analog signals are transformed to a digital signal, square wave, by converting the analog value to a corresponding period time for the signal. PWM signals can be considered as a series of repeating on and off pulses. Further, the PWM signals are non-negative square wave signals with a run-time adjustable frequency and a run-time adjustable duty cycle. This means the amplitude is constant and pre-determined. In this project PWM signals are utilized for speed signals of the driveline. A higher speed signal results in a higher frequency and therefore also in a shorter period time.

The *duty cycle* of a square wave is determined by the time the signal is high, i.e. the percent value of a period time the pulse is active, defines the duty cycle. Therefore the duty cycle can be assign values between zero and one.

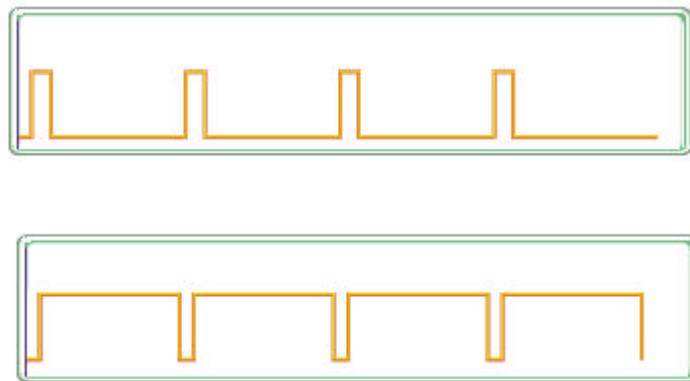


Figure B1. PWM-signal with different duty cycle

The upper figure above shows a PWM output signal with 10% duty cycle and the lower figure a PWM signal with 90% duty cycle. The first case means that the signal is on for 10% of the period time and off for the remaining 90%. These two PWM outputs encode different analog signal values at 10% and 90% of full strength. In the project PWM-signals with a duty cycle of 50 % are used.

Appendix C Friction Base

The friction base is a component from the Modelica Standard library. The component is used to handle shifts between different states, i.e. the Friction base works as a state machine. It controls what state that is active and refers them to different equations. The friction base is particular intended to be used when modeling a clutch and a brake. Through inheritance is the friction base in this project used in the brake model.

The state machine handles four different states: stuck, free, forward sliding and backward sliding. The states will be discussed individually.

C1 The stuck mode

This state becomes active when two surfaces are pressed together so hard that no sliding occurs between them. This is the case when the generated frictional torque is big enough to keep the relative angular velocity between the frictional surfaces to zero. In this mode a static frictional torque defines the size of the torque that has to be transferred to keep the surfaces stuck to each other. If the value becomes larger than the maximum static frictional torque then the surfaces will begin to slide against each other.

The stuck mode describes the physical behaviour of the brake system during deceleration of the truck with locked wheels.

C2 The sliding modes

This mode is active when the transferred torque exceeds the maximum static frictional torque and therefore, as mentioned above, in that case the surfaces will begin to slide against each other. Sliding can either occur forward or backward. The frictional torque is in this mode determined by a function of the relative angular velocity of the component and the normal force that presses the two surfaces together. In other words:

$$\mathbf{t} = f(\mathbf{w}_{out} - \mathbf{w}_{in}, F_{normal}) \quad (C.1)$$

In opposite to the stuck mode, there is in this mode a need for a maximum kinetic frictional torque to be defined. If the transferred torque becomes smaller than the torque limit, the surfaces will be rigidly connected to each other. This means a transition back to the stuck mode.

The sliding mode is active during braking with non-locked wheels.

C3 The free mode

When the two surfaces are released from each other, then the free mode becomes active. This means the friction elements will begin to rotate independent of each other. Furthermore, no torque is delivered through the component. Therefore this state is active when the brake is disengaged.

Appendix D Tire variables

D1 Magic formula

Scania has in previously performed simulations used a traditional tire model, called Magic formula. In these simulations, tire data for three different tires were used. These data has been used in the work to calculate the longitudinal tire force.

The empirical formula, Magic Formula, for calculating the tire force can be expressed by:

$$F_x = D \cdot \sin(C \cdot \arctan(B \cdot x - E \cdot (B \cdot x - \arctan(B \cdot x)))) \quad (D.1)$$

Where:

- B : stiffness factor.
- C : shape factor.
- D : peak value.
- E : curvature factor.

These variables are determined from different tire parameters and the normal force acting on each wheel.

$$BCD = (a_3 \cdot F_z^2 + a_4 \cdot F_z) \cdot \exp(-a_5 \cdot F_z) \quad (D.2)$$

$$C = a_0 \quad (D.3)$$

$$D = a_1 \cdot F_z^2 + a_2 \cdot F_z \quad (D.4)$$

$$E = a_6 \cdot F_z^2 + a_7 \cdot F_z + a_8 \quad (D.5)$$

$$B = \frac{(a_3 \cdot F_z^2 + a_4 \cdot F_z)}{a_0 \cdot (a_1 \cdot F_z^2 + a_2 \cdot F_z)} \cdot \exp(-a_5 \cdot F_z) \quad (D.6)$$

$$D = \mathbf{m}_{xm} \cdot F_z \quad (D.7)$$

$$\mathbf{m}_{xm} = a_1 \cdot F_z + a_2 \quad (D.8)$$

Where:

- \mathbf{m}_{xm} : longitudinal friction coefficient.
- a_1 : load dependency of longitudinal friction.
- a_2 : longitudinal friction level.
- a_0 to a_8 : tire parameters.

Appendix E Numerical properties

Many differential equations cannot be solved analytically. In such case numerical methods can be used to obtain a good approximate solution.

When a dynamic model is going to be simulated, it becomes necessary to use a numerical integrator. This integrator is used to calculate the states of the model. In real-time simulations, the computer has to update the states faster than the sample time. Otherwise it is impossible to run simulations in real-time.

To get a fast update of the states is there a good advice to use an integrator that needs few calculations to solve. Euler's two numerical methods demand few calculations and are therefore useful. Equation (E.1) and (E.2) describes the backward and the forward method. T represents the fixed step time of the solver.

$$x(t+T) = T \cdot \dot{x}(t+T) + x(t) \quad (E.1)$$

$$x(t+T) = T \cdot \dot{x}(t) + x(t) \quad (E.2)$$

Since the states of the model are calculated with differential equations. The time derivatives are described as a function of the states and the input signals. Which means the derivative can be expressed by:

$$\dot{x}(t) = f(x(t), u(t)) \quad (E.3)$$

Equation (E.3) together with equation (E.1) gives:

$$x(t+T) = T \cdot f(x(t+T), u(t+T)) + x(t)$$

By looking at the right side of equation (E.3) and observe that it contains values of the input signal at the next sample time, there is easy to establish that it is impossible to pre-compute the state. Therefore Euler's backward method cannot be used for real-time simulations. However by using Euler's forward method already known values of the states and input signals can be used to calculate the next state within the sample time.

The step time in the calculations was chosen to 1ms. This value has previously been used in real-time simulations, and worked well also in these simulations. The demand of a high sampling frequency is derived from the natural frequencies of the tire model.