CHALMERS

Driving resistance analysis of long haulage trucks at Volvo

-Test methods evaluation

Master's Thesis in the Master's programme Automotive engineering

HENRIK STENVALL

Department of Applied mechanics
Division of Vehicle engineering & autonomous systems
CHALMERS UNIVERSITY OF TECHNOLOGY
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Cover:
Volvo FH with T052 trailer equipped with 5th wheel for coast down test. This vehicle is the reference vehicle, driven at autumn 2009, for coast down results see Figure 24.

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ABSTRACT

To further improve fuel consumption of future trucks and to be able to measure the gains obtained from new inventions, accurate test methods have to be defined. The three main tests currently used at Volvo are: road testing, chassis dynamometer and computer simulations. In this project the three fuel consumption test methods are evaluated with the Volvo FH and its main rivals to get an indication of the results reliability. If the test gives different fuel consumptions but has similar percentage differences between the trucks a fixed relation between test methods is obtained.

The chassis dynamometer's road load is based on the measured force at the vehicle for every vehicle speed. To find this relation a coast down test for each truck is performed; the test is based on continuous logging of vehicle speed and time while driving the equipage at neutral gear from 85-16 km/h. The tests were done in February and consequently weather influenced the results, with wet tracks, winds and low temperatures.

The road tests were carried out in April with less restrictive environment, but also using bedded tires and different trailers. The resulting fuel consumption at the Lv-Bo-Lv (Landvetter-Borås-Landvetter) duty cycle for each truck were without exception lower for the road tests compared to the chassis dynamometer results. The percentage difference between the tests were not constant for the different trucks but rather close to the difference in-between trucks attained from the coast downs at full speed, which reflects the Lv-Bo-Lv duty cycle well, (except for the road inclinations). This proves that the coast downs' were influenced by non-truck specific matters.

Great care must be taken during preparation of vehicles prior to fuel consumption and driving resistance tests. Weather influences the results the most and has to be measurably stable between the different tests. The vehicles have to be accurately prepared with similar tires, correctly adjusted deflectors, same trailer and engines with the similar specifications and wear. At the conducted tests several of previously mentioned issues were omitted, the results were therefore heavily affected, finally suggestions for future testing has been established.

Key words: Coast down, chassis dynamometer, fuel consumption, driving resistances, aerodynamic resistance, rolling resistance, powertrain resistance and test methods

Körmotstånds analys av lång transports lastbilar på Volvo
-Testmetods utvärdering
Examensarbete inom Automotive engineering

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SAMMANFATTNING

För att sänka framtida lastbilars bränsleförbrukning och kunna kvantifiera de förbättringar nya innovationer ger så måste nogranna testmetoder vara definerade. De tre huvudsakliga testerna på Volvo nuförtiden är: vägkörning, chassis dynamometer och datorsimuleringar. I detta projekt är de tre bränsleförbrukningstesterna utvärderade med hjälp av Volvo's FH och dess huvudkonkurenter för att få en indikation av resultatens tillförlitlighet. Om testerna ger olika bränsleförbruknings värden men har liknande procentuella skilland mellan lastbilarna så erhålles en skillnad mellan de olika testmetoderna.

Chassi dynamometerns väglast bygger på den uppmätta kraften på fordonet för varje hastighet. För att finna detta förhållande utförs ett utrullningsprov för varje lastbil; testet bygger på en kontinuerlig sampling av hastigheten och tiden medan fordonet frambringas i neutralväxel från 85-16 km/h. Proven utfördes i Februari och följdaktligen så påverkades resultatet av vädret så som: våta vägbanor, blåstt och låga temperaturer.

Vägkörningen gjordes under April vilket betyder mindre påverkan av omgivningen men också med inkördadäck och andra trailers. Den uppmätta bränsleförbrukningen på Lv-Bo-Lv (Landvetter-Borås-Landvetter) körcykel för varje lastbil var uteslutande lägre under vägkörningen jämfört med chassi dynamometern. Den procentuella skillnaden var inte konstant för de olika lastbilarna men differansen var relativt lik den skillnad i motstånd som erhållits från utrullningen vid tophastigheten, som reflekterar Lv-Bo-Lv testcykel, (förutom vid väglutningar). Detta bevisar at utrullningarna var påverkade av icke lastbils relaterade faktorer.

Under fordonsförberedelserna för bränsleförbruknings och körmotstånds tester måste stor noggranhet tillämpas. Väder påverkar resultatet mest och måste vara mätbart stabilt emellan tester. Fordonen måste vara noggrant förberedda med liknande däck, korrekt inställda vindriktare, samma trailer och motorer med liknande specifiaktioner och slitage. På de utförda testerna var de tidigare nämnda problemen neglegerade, resultatet var därmed tydligt påverkat, förslag för framtidatestning har slutligen föreslagits.

Nyckelord: Utrullning, chassi dynamometer, bränsleförbrukning, luftmotstånd, rullmotstånd, drivlineförluster och testmetoder

Contents

Preface

Analysis and performance of fuel consumption test methods have been conducted from January 2010 to June 2010. The project is a necessary part for accurately distinguishing the driving resistances of future heavy vehicle enhancements. The work is executed at Department Complete Vehicle, Feature and Competitor Analysis, Volvo 3P, Sweden. The project is supervised by Professor Lennart Löfdahl, Applied Mechanics, Vehicle Engineering & Autonomous Systems, Chalmers University of Technology, Sweden.

The vehicles testing have been carried out with B.Sc. Jens Björnsson, Volvo 3P and M.Sc. Inge Qvarford, Volvo 3P. The testings were taking place at Hällered Proving Ground and at roads with the suitable duty cycles. Dr. Jan Melin, Volvo 3P has supervised and initiated the project and most importantly been a great support during the whole procedure.

I would especially like to thank Jens Björnsson for his supervision and mentors like support during the project, and also give my highest appreciation to Jan Melin and the rest of the Feature and Competitor Analysis team for their contribution and help. Finally, I would like to thank Lars Rudling and his colleagues in VFL for their support and professionalism with the testing, and Elie Garcia for his guidance in all fuel economy areas, and my Chalmers supervision given by Lennart Löfdahl that has been excellent.

Gothenburg, May 2010

Henrik Stenvall

Notations

Roman upper case letters

A	Frontal area
C_D	Drag coefficient
F	Total vehicle drive force
F_D	Aerodynamic force
F_g	Gravitational force
F_P	Powertrain force
F_R	Rolling resistance force
F_{trac}	Traction force
$I_{brake\ disc}$	Inertia of rear brake disc
$I_{differential}$	Inertia of differential
$I_{drive\ shaft}$	Inertia of drive shaft
$I_{dual\ tire}$	Inertia of a dual tire including the rim
$I_{gearbox}$	Inertia of gearbox at neutral gear
I_{hub}	Inertia of rear wheel hub
I_{pinion}	Inertia of final drive pinion
$I_{propeller\ \mathrm{shaft}}$	Inertia of propeller shaft
R_m	Rolling resistance deceleration
S	Gradient inclination in percentage
T_{eng}	Engine torque
V_1	Volume of part one rear wheel hub
V_2	Volume of part two rear wheel hub

Roman lower case letters

α	Road inclination
η_{trans}	Efficiency of transmission
ρ	Air density
ω_g	Engine rotational speed
f_r	Rolling resistance coefficient
g	Gravitational acceleration
i_f	Final drive ratio
i_g	Gear ratio
m	Vehicle mass
m_b	Mass of rear brake disc
m_{W1}	Mass of part one rear wheel hub
m_{W2}	Mass of part two rear wheel hub
$r_{1,inner}$	Inner radius of rear wheel hub part one
$r_{1,outer}$	Outer radius of rear wheel hub part one
$r_{2,inner}$	Inner radius of rear wheel hub part two
$r_{2,outer}$	Outer radius of rear wheel hub part two
r_e	Effective wheel radius
r_{inner}	Inner radius of rear brake disc
r_{outer}	Outer radius of rear brake disc
t_1	Thickness of rear wheel hub part one
t_2	Thickness of rear wheel hub part two
v	Vehicle speed
v_{wind}	Wind speed

1 Introduction

Fuel consumption has always been one of the key issues in the truck business, not least in the last couple of years when environmental concerns have been increasing globally. Decreasing operating costs for the customers have been driving the development for many years whereby a lot of experiences in the fuel consumption field have been obtained. For economical reasons most of the improvements suggested have not been implemented, but now that governmental and customer demands are increasing, more drastic enhancements are to be adapted onto future trucks.

Before new developments are applied, it is important to analyze which areas having the highest need of improvements. To be able to distinguish where upgrades are most effectively implemented, vehicle testing needs to be carried out. Presently there are three kinds of fuel consumption tests that are used for evaluating trucks at Volvo: driving on roads and measuring the fuel consumed during the cycle, driving the truck on a chassis dynamometer by adding the vehicle resistances at different speeds, to the moving ground. The final way to measure fuel consumption is by using a computer simulation program to calculate the resistance at every point of the drive cycle.

The testing will involve the Volvo FH, see Figure 1, and four competitors: Scania R480, Mercedes Actros 1848, Renault Premium and a DAF XF105. All vehicles except the Renault Premium have similar specifications such as sleeping cabins, deflectors and most importantly the Euro V specification powertrain.

Figure 1: Volvo FH 500 Euro V used at the fuel consumption testing

The work will be carried out for Volvo 3P which is part of the Volvo group. Volvo 3P deals with truck related development for all truck brands within the Volvo group. More specifically the work will be performed at complete vehicle in Lundby, Gothenburg and at the test track of Hällered.

2 Background

To construct more effective trucks than competitors, Volvo needs analysis of how their trucks are performing compared to the main rivals. To be able to compare the fuel consumption on an accurate scale, different tests of the Volvo FH as well as its main competitors are carried out. The project will look into the test methods used to get hold of the fuel consumption and finding drawbacks in using the particular method.

The results from the testing will be analyzed with the purpose of finding areas where Volvo needs to improve relatively to the competitors. The different driving resistances are the areas of interest, by dividing the complete drive resistance into the different losses caused by the rolling, aerodynamic and powertrain it is easier to recommend individual improvements suitable for the Volvo FH. The forces acting on a tractor with semi-trailer are summarized in Figure 2. The gravitational contribution F_g is not analyzed to any large extent in this study.

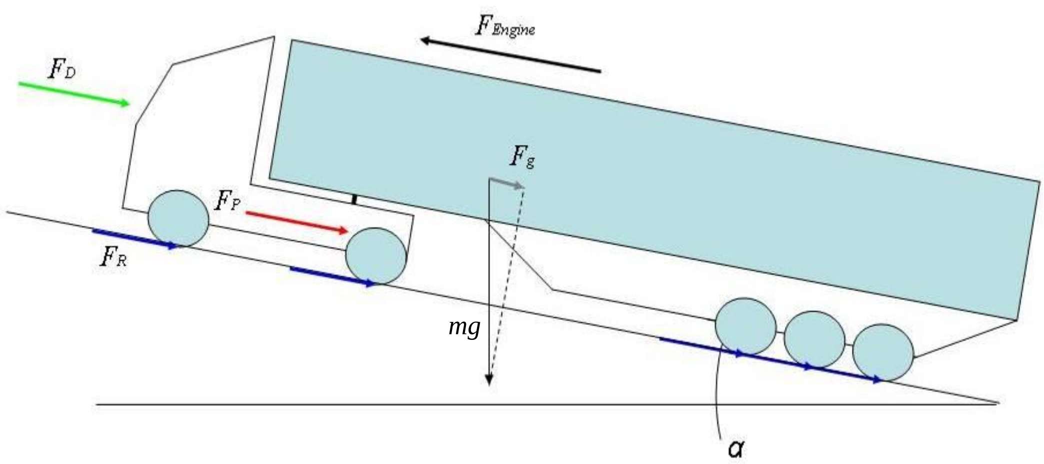

Figure 2: Complete set of forces acting on a truck

In order to do fuel consumption measurements in the chassis dynamometer the speed dependent resistance of the complete vehicle needs to be obtained. This is done by performing a coast down test. The coast down test is a good indicator of which kind of resistances that dominates. The test is however very sensitive to exterior influences and should be performed with care. To gain experience for the forthcoming Euro VI vehicles that arrive in 2011, the coast down of the Euro V's are to be used as an indicator of which parameters of the truck that needs to be fixed during the coast down, in order to get reliable results.

3 Boundaries

The complexity and size of the project make it difficult to analyze emissions which are one of the main purposes of this kind of vehicle testing. The emission analysis is carried out parallel at Volvo Powertrain and does not require further attention. The main reason why Euro V vehicles were used for the tests was for the emissions, but the main attention of this report is on the drive resistances, the emissions should be considered as a side step. Some of the duty cycles are however mainly interesting for emissions whereby less attention to the fuel consumption results of these duty cycles are seen.

Modifications of the tested vehicles were not possible to do due to the short time period of this project, even if such measures would have been important in order to achieve comparable results between the various trucks. Another reason was that the competitor vehicles were borrowed and therefore no changes could be done. Variations in fundamental parts such as tires were for example an area that undeniably affected the resistances at the coast down test.

The analysis as well as the scope of the project is applicable on long haulage trucks, consisting of a tractor with a semi-trailer. This is the most common configuration of cargo trucks and therefore the greatest interest of improvement is found there. The main results are nonetheless easy to apply onto other kinds of heavy vehicles.

4 Fuel Consumption Test Methods

Presently fuel consumption is measured in various ways in the vehicle industry. Some ways are preferred when it comes to accuracy and some for their cost-effectiveness. The most common methods are either real world road testing, chassis dynamometer full vehicle simulation, computer simulation and engine bench measurement. Tests of the first three methods have been performed and compared with each other in Chapter 5.

To be able to perform chassis dynamometer tests a load curve of the truck has to be inserted into the computer. Load curves are created by measuring the driving force on a complete truck for different speeds, one such test is called coast down test. The vehicle is left on neutral gear for the entire speed interval of importance, by continuously measuring the speed and time, from that accelerations and later on forces can be calculated. The coast down testing also allows a lot of losses analyze to be performed. Such as how much rolling and aerodynamic resistances contributes to respectively, but also in what extent environmental parameters affect the results.

In order to more accurately investigate the coast down results a powertrain coast down was performed. The test measures only the losses in the driveline and as a consequence the two remaining resistances can more easily be distinguished.

The first two tests do not result in any fuel consumption read-outs, but they give an insight of the driving resistances acting on the vehicle, which are affecting the fuel consumption for the various configured trucks. Most importantly these tests act as the base for the chassis dynamometer measurement.

The later part of this chapter is mainly focused on fuel consumption. The central part in the report is to distinguish obvious differences between the test methods while the comparison between competitors is a side track. The reason for this is the potentially low repeatability of the tests; little care was taken to ensure equal conditions during testing. The comparison between the test methods gives a clear indication of what advantages and disadvantages that can be found, and hopefully help as a guide for future testing of trucks. In the same time results comparisons between trucks can be misguiding due to the unknown accuracy of test methods and the results dependency on both truck and non-truck related factors.

4.1 Powertrain coast down

A powertrain coast down is a test conducted in order to find the speed dependent resistance of the complete driveline. The rotational speed of the wheels are logged with high frequency and used for calculating the wheel retardation.

4.1.1 Powertrain coast down description

To perform the test, the rear of the truck was lifted from the ground, for safety reasons the rear axle was both put on jack stands as well as lifted by an overhead crane, see Figure 3. To measure purely powertrain losses the retarder has to be switched off, moreover the wheel speed has to be measured on both wheels to prevent any frictional difference in the bearings affecting the rotational speed. The differential distributes the input speed to the wheel pair, but concentrates the speed to the side with less resistance, as a consequence a small difference between wheel speeds can occur. The average value of the two sides is finally used as the output speed in calculations.

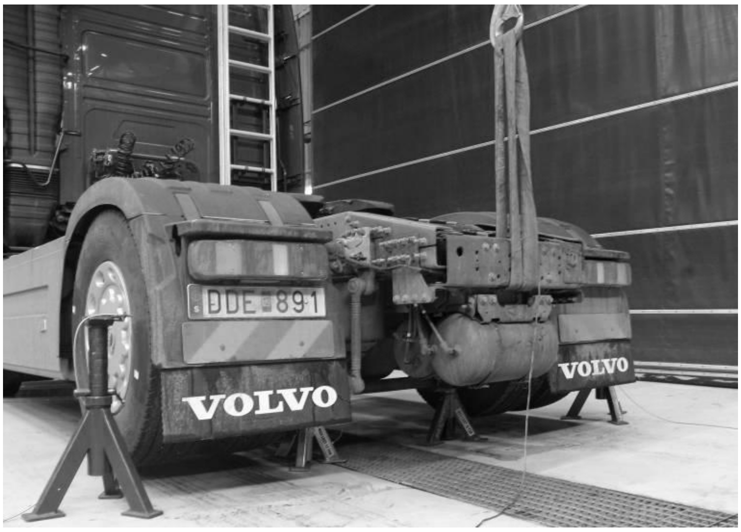

Figure 3: Lifted vehicle at powertrain coast down test

The measurement is carried out with “imc devices” where the signals from the measuring lasers are interpreted. The lasers are sending pulses at 100 Hz through some optical fibres, the signal is sent back if a reflecting surface is hit by the beam. Two pieces of reflecting tape is placed on the rubber with 180º spacing, to achieve more accurate readings. (From the test it was found that a couple of more pieces of tape could have been used to enhance the accuracy). The lasers are placed two centimetres from the wheels and at the level where the tapes are mounted by firmly attaching them onto jack stands, see Figure 4. The signals from the lasers are then transmitted to a data acquisition unit which converts the data into rotational speeds shown at the computer software.

Figure 4: Laser mounting for powertrain coast down test

In order to run the engine in the workshop the exhaust was attached to a flexible exhaust pipe. The ESP (Electronic Stability programme) and ABS (Anti-lock Braking System) fuses were also disconnected just for precaution not to have any influence from these systems.

To do the test the vehicle firstly needs to be propelled to 90 km/h (or close by), at this speed the speed limiter cuts the fuel feeding and prohibit any speeds above 90 km/h. It takes some time for the engine to regain the moment, and as a consequence the top speeds are difficult to capture. As close to 90 km/h as possible the gearbox is put in neutral. Before the gear change the logger program is triggered, all the extra data can easily be cut out afterwards, the measurement is stopped when the low range gear is connected at 16 km/h. When the low range gears connects the rotational speed of the main shaft in the gearbox increases and raises the driving resistance.

4.1.2 Issues with powertrain coast down

The results from the measurement can for various reasons be inaccurate and/or wrong. By listing the problems that might occur while testing, better knowledge on how to interpret the results and improve future testing is obtained.

- From the results a clear difference between cold and pre-heated components was seen, the reason is the decreased viscosity of the various lubricants in the powertrain. No temperatures in the gearbox or final drive were logged and therefore it is difficult to accurately define whether the operating temperature was reached or not. A decrease in transmission losses would most certainly be achieved if the truck was warmed up to a higher extent.

- Another factor was the inaccurate reading that only two pieces of tape contributed to, when the wheels rotated slowly. The program worked in such a way that if no signal was sent during the time interval the previous value was stored instead. This can be seen in Appendix F 1 at low speeds where the graphs become stair shaped.

- In order to get a continuous curve the different runs are averaged and a trendline based on a second order polynomial is used to describe the curve. This trendline is later on used for calculation of the rotational speed difference, by looking at the deceleration between two different speeds. In this method some information is lost, but the trend should still be fairly accurate.

- When calculating the moment at the wheels the inertia of all the components has to be known. This information was collected from several sources, and consequently different accuracies were used. The error should not be big because the major contributors are the tires and those values were given from the tire supplier Michelin for the specific tires used.

- The inertia of the tires was approximated for a brand new set of rubber and because material is consumed when the tires are worn the inertia could be changed significantly.

- Brake drag was not avoided in any extent, except that both wheels were rotated to distinguish whether there were any big differences in resisting force. It can be debated if the brake drag should be included in the calculation or not, because at real driving some resisting force are caused by the brakes, the magnitude is however difficult to recreate, because losses can be irregular between tests.

- The amount of oil in the gearbox as well as the final drive changes the splash losses and no measurement was carried out to distinguish the amount, which makes the test difficult to duplicate.

- The bearing friction is decreased by having the rear axle lifted, and not having any extra loading on the bearing surface.

4.1.3 The powertrain test

In order to perform the powertrain coast down test a Volvo FH13 was used with similar specifications as the one used later on at the coast down test, see Figure 5. Firstly a test where all the components were cold was conducted. Five runs were carried out with this condition, but only four of them were valid due to a late start of data logging in one of them. The same tendency could be seen in all the valid runs, see Appendix F 1. The weather prior to the drive into the workshop was -2 ºC. Due to some preparation time inside the workshop before the actual measurement took place a slight increase in component temperature can be expected.

Figure 5: Volvo FH used at the powertrain coast down test

Before the next test set the tractor (without trailer) was driven for 30 minutes around Hisingen in order to build up some heat in all the rotating components. At the end of the run the engine temperature had reached around one third of the maximum temperature on the gauge scale. No temperatures in the powertrain components were logged and therefore a difference between the test temperature and real world driving can be suspected.

For the warm conditions six runs were performed while one of them failed because of too much missing data at high speeds when the speed limiter cut the fuel feed. (The control system of the speed limiter regulates slowly, and therefore the speed had time to drop 15 km/h before the engine was feed with fuel again. That is because the control loop is suitable for a loaded vehicle and not only an unloaded powertrain). A clear difference of around four seconds of coasting time was seen at the two tested temperatures.

4.1.4 Post-processing

The data stored from the tests was exported into Excel for post-processing. Firstly the data outside the measuring region was erased. A constant starting speed of 74 km/h was chosen for all measurements which were a limit that all tests managed to keep, the stop speed used was 17 km/h. The runs were combined and averaged for each temperature. To get a continuous curve the average run was approximated by a second order polynomial equation. The equation was then used for calculating the rotational speed difference at every five km/h. Along with the inertia of the driveline components the moment acting on the wheels can be calculated, and then by finally dividing the moment by the wheel radius the resisting force at different speeds can be obtained.

The inertia of the various powertrain components were collected from different departments at Volvo group and probably have various reliabilities. The inertia of tires was given from Michelin for their specific model XDA2 which are the main contributors of the inertia. The driveshaft's, differential gear's, (with gear ratio 2.64:1) and pinion's inertia was given from department: axle engineering in France. The gearbox inertia of the I-shift transmission in neutral and with the high range gear connected was given from department: drivelines and hybrids. The propeller shaft C 2055's inertia was given from department: driveline subsystems. The brake disc and wheel hub were measured in size and approximated as cylinders of different sizes in order to calculate their inertia, see Appendix H.

The inertia times the change in rotational speed is basically proportional to the frictional losses and the splash losses. The frictional losses can be divided into rubbing between gear wheels, (in this case that is mostly the final drive, because it is heavily pre-tensioned), and bearing friction. Most bearings in the powertrain are of conical type in order to withstand the axial loads caused by helical and hypoid gears.

The losses in the powertrain are presented in Figure 6 for the two temperature cases. At Cold temperatures the losses are significantly higher, and if analyzing the shapes it can be seen that the Cold temperature curve has a higher influence of the second order term because it increases rapidly at higher speeds. The second order term is mainly affected by the splash losses because it is a fluid dynamic loss. Splash losses are in this case mostly focused to the final drive because almost no gears are rotating in the gearbox. The frictional losses are naturally also higher when the viscosity is higher as can be seen from the curve at low speeds. (At low speeds splash losses are negligible and therefore most losses are due to friction). The pre-tension should however be decreased at low temperatures and contradictory to what is seen in Figure 6, lowers the frictional losses. Consequently the increase in bearing friction due to viscosity changes is dominant.

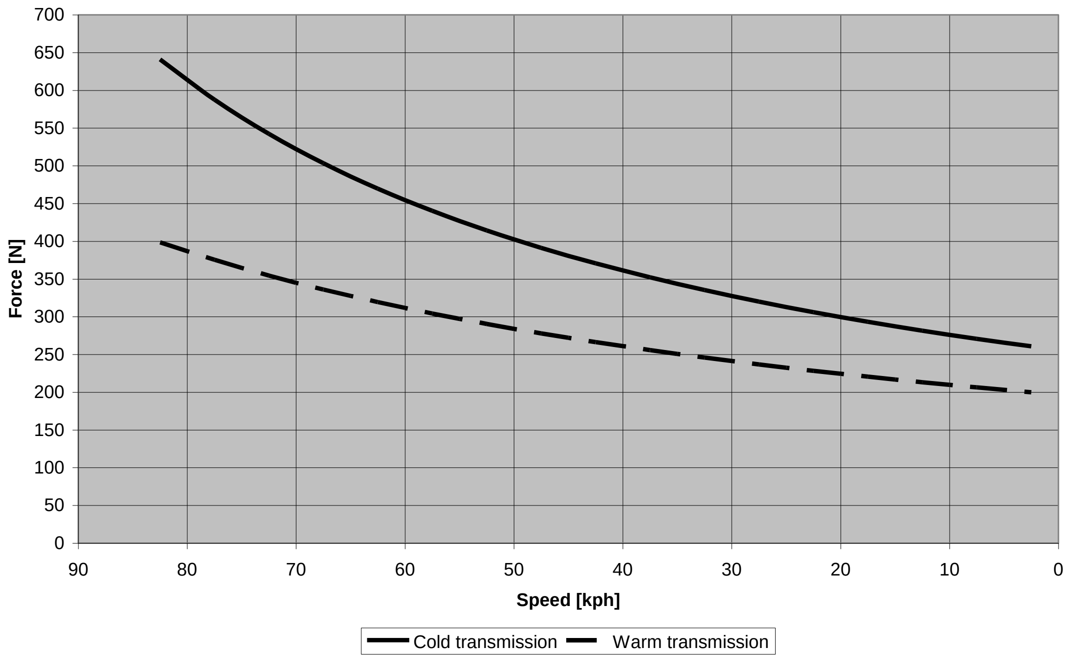

Figure 6: Powertrain coast down resistances for the two temperatures, (extrapolated results)

4.2 Coast down test

To improve fuel consumption of trucks driving resistances must be kept low. The main losses are divided into aerodynamic resistance, rolling resistance and resistance due to gradients. While the first three losses are much dependent on how efficient the vehicle is, the last one is mainly dependent on the terrain. The main interests here are therefore to decrease the first three losses. The most accurate way is by measuring the actual fuel consumption for a specific cycle on the road. Repeatability of road tests is however low and it is not feasible to perform for all different combinations, of cabins, trailers, engines, transmissions etc. A far more attractive approach is to run the vehicles inside, in-house in a chassis dynamometer. The complication is to get the actual resistance at different vehicle speeds for the complete vehicle. One way of measuring the combined resistances at varying vehicle speeds is performing a coast down test.

4.2.1 Coast down description

The coast down is a difficult test to perform if accurate and repeatable results are to be obtained, parallel to that it is both time and cost consuming, especially if competitor vehicles are to be rented or bought for comparison. The basic principle is to measure for how long the vehicle is able to coast from a certain speed. A starting speed a bit above 85 km/h[1] is sufficient for a truck because the maximum allowed speed is 80 km/h. When the vehicle has reached the starting speed and a sufficiently long straight, (800m is sufficient), the gearlever is put into neutral position. The vehicle is now coasting down the straight and the friction from moving components along with the approaching air decelerates the vehicle's speed. During the deceleration the time and speed are being measured.

For good accuracy the speed is obtained from a 5th wheel mounted onto the truck. The 5th wheel is mounted tight to the vehicle but is allowed to follow the roads unevenness with help of a spring and damper. It's important that the wheel has contact with the ground at all conditions and preferably not varying the contact load much, because it changes the effective rolling radius of the tire. The 5th wheel has to be calibrated prior to the measurement in order to find the calibration factor, which equals the amount of sensing pulses per meter road. To get the vehicle speed the amount of pulses per second is divided by the calibration factor.

The coast down course has to be straight and flat to exclude side forces and the influence of gradients. Moreover the surface needs to be dry in order not to exaggerate the rolling resistance contribution. A day with absence of wind is also preferred so that a consistent measurement is conducted and variations in aerodynamic force are avoided. If the flow is hitting the truck from the sides the drag coefficient can increase with up to 70 % with 15º yaw angle[2]. This is however based on a semitrailer without aerodynamic add-ons, and especially with side deflectors this increase in drag is significantly lower. The main reason why side winds affects drag is due to wind passing thru the trailer gap, but a secondary reason is the increased size of the wake region, see Figure 7. Lastly it should be noted that even the frontal area increases with a slightly increases with a slightly angled head wind.

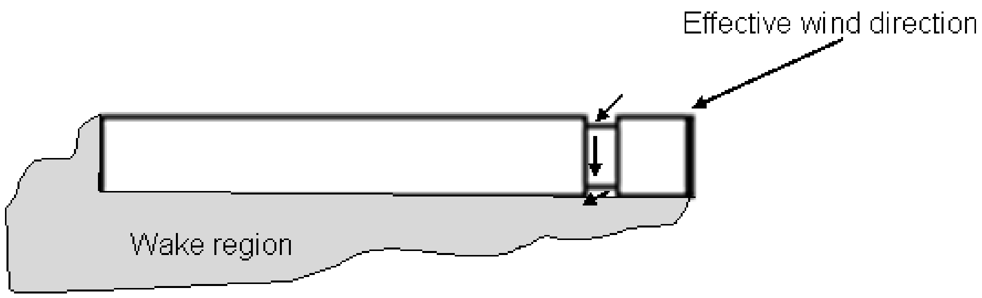

Figure 7: Side wind motion around semitrailer, plus wake region.

Normally a heavy truck is coasting for quite a while, resulting in a number of required runs down the straight in order to complete the sequence. To get a continuous data sequence the next run must be started with a bit of overlap in speed. Normally the procedure stops when the speed is reaching 10 km/h because then more factors comes into play. For Volvo trucks the limit is a bit higher because the low range neutral gear automatically engages below 16 km/h and changes the driveline force significantly, (this speed can vary between manufacturers and gearbox type).

4.2.2 Issues with coast down

To achieve reliable test results some different conditions must be met. In most cases the procedure to go thru all faults is very time consuming and therefore most of them can be neglected, the result can however be very misleading in the worst scenarios.

- Temperature of all components must be kept constant between tests, for example oil temperatures of transmission and rear axle have to be relatively constant to achieve comparable results. If significant temperature differences occur the viscosity difference of the oil will result in varying driveline resistances.
- Temperature differences in the tires can also affect the results, because friction increases when tires are warm. In order to quickly build up temperature in the tire the hysteresis must be high, which is achieved by driving fast and/or at bumpy roads.
- Road conditions are also affecting the results partly because water on the road sticks to the rubber and increases rolling resistance. But also because the water acts as coolant and thereby changes the bounding force between the rubber and the asphalt, (decreasing hysteresis).

- Winds are also affecting the results. Head- and tailwind obviously change the aerodynamic resistance, but side winds not only increases the relative speed it also results in higher drag coefficient. Especially sensitive to side winds are trucks with gaps between the tractor and trailer, (such as the semi-trailer). Headwinds above 2 m/s are usually a level where significant influence on the results is occuring.
- If different roads are used between tests the results might be affected by different adhesion factors, and consequently different rolling resistances. The different types of asphalt can also vary the up and down movements of the wheels and thereby increase the losses due to energy absorption in the suspension and tires.
- Road inclinations are also an issue affecting the force required to propel the vehicle. The height difference between start and finish point of measurement is important to keep constant, but also the shape of the course can affect, as if the track is shaped as a hammock. On Hällered the straights varies in height with around 1 m. To compensate for this the height above sea level can be continuously measured to be able to subtract that contribution.
- The right amount of oil in the gearbox is also important to avoid non-comparable splash losses, between different trucks.
- Brake pads and shoes must be sufficiently worn-in to present realistic surface friction. Worn brake pads also have a tendency to move the rubbing friction surface away from the disc due to more unevenness at the friction surface, and thereby decrease the rolling resistance.
- Tire pressure must be accurately checked and filled at a standard temperature. If the tires were checked at low temperature the pressure will show lower values than for high temperature measurements.
- In winters snow on the surface of the truck can drastically increase the viscous forces, and especially at the large trailer surface area. In summers and winters dirt can also increase the surface friction, whereby a clean and dry vehicle is preferred.
- The surrounding air pressure also affects the aerodynamic force factor by changing the air density and is around 7 % higher at -5ºC relative to +15ºC, and therefore the aerodynamic force has a larger contribution at cold weathers.
- The trucks utilize air suspensions and can consequently change ride heights. The comparison between different vehicles is therefore much dependent on correctly set ride heights.
- Correctly adjusted roof and side deflectors are necessary in order to have similar aerodynamic conditions for the different vehicles tested. That is a smooth transition between the tractor and trailer.
- Similar tires are needed to achieve comparable results between different vehicles. The tire brand, tire size, the tire pattern as well as the filler content of silica-silane as replacement for carbon black are factors needed to be kept constant to have comparable losses between trucks. The wear of the tire is another factor influencing the rolling resistance coefficient, up to 20 % lower for worn tires, than for new ones.

- If the truck is not sufficiently worn-in there might be rolling resistance related losses from, suspensions that are not sufficiently seated and consequently have the wrong wheel angles or just a lot of energy absorption. Transmission parts are also required to be worn-in to represent actual friction losses.
- Toe-in can significantly increase rolling resistance, by having high toe-in values the vehicle becomes more stable whereby some manufacturers tends to use high values. The wheel angles can also be distorted by severe contacts to heavy obstacles. In order to have reliable comparisons these angles should be kept at the same level for all vehicles or at the manufacturer specified values.
- Vertical movement of the truck due to road unevenness is also affecting the results, because forward motion is transferred into vertical motion and lost in the dampers as heat. But the results can also be spoiled if the road unevenness changes the effective rolling radius of the 5th wheel tire and consequently changing the amount of pulses per wheel revolution.
- The aerodynamics of the truck is changed by removing some of the radiator panels. The absences of the panels are due to mountings of the 5th wheel. It is hard to predict whether or not the panels give any clear change in drag coefficient, but they do not only change the shape they also allow more air to pass through the radiator, and consequently the engine might get colder, especially at winters. This is however not affecting the coast down results.

4.2.3 Test preparations

Before the coast down test a number of vehicle preparations are required. Most importantly the 5th wheel must be firmly mounted to the body, either at the front on the beams behind the radiator grille, or to mount the wheel behind the tractor, as long as it moves freely from the trailer. For the trailer used, there is not enough space between the trailer and the tractor, whereby the mounting has to be in front. For most trucks the towing hook is used to hold the 5th wheel armature. The important aspect is to insure that the wheel is not able to move relative to the ground, and a stiff mounting is therefore required.

The speed measuring wheel is held in place by two towing hooks for cars, seen on Figure 8. A spring and damper are mounted to the wheel in order to allow for some movement. In the bearing of the wheel a pulse sensor is located, which sends signals through the cable to the signal conditioning unit. The signal conditioning unit is sending the sample data to the break out box, which divides the signals to different outputs, in this case that is only to the AAC-2 unit, which uses the digital pulse signal, timing and triggering signal, and relates them in time. The whole package of electric units is shown in Figure 9.

The signals are then transmitted to the computer, where Easyview software is processing the signals. To find the signal the connection has to be changed to COM1. Easyview is firstly needed for calibration in order to find the relation between distance travelled and pulses, this procedure should preferably be done both before and after the coast down test. Calibration is done by driving a measured distance and accumulating the amount of pulses. In Easyview this is done by just clicking logger start/stop, and defining a name of the file. The different input signals are to be defined; number 3 is the trigger signal, while the wheel speed sensor is number 25. The wheel speed has to be set to accumulated measurement, all other parameters are using the preset values. The sampling rate might have to be changed to 10 Hz, (if this is not changed automatically). When pressing Finish the sampling will start, to observe the measurement the green voltage meter can be used. When the finishing line is reached the value is stored and used for calculating the calibration factor by dividing the value by the distance travelled.

Figure 8: 5th wheel and mounting plates on the Scania R-series

Next step is to start the measurement, this is done by pressing Logger Start, by using the previously defined parameters the only things to change is the wheel pulses sampling. Instead of accumulated, reset pulses are to be used, the rest is just to leave unchanged. When pressing Finish the logger will start calculating pulses, the real measurement will however not start until the trigger has been pushed. In order to stop the sampling the trigger is used to indicate when this will occur. The continuous sampling is stopped by pressing Logger Stop. The file can now be saved and exported as a txt file. The whole measurement range has to be exported and this is done by changing the view to see the complete sample and then export.

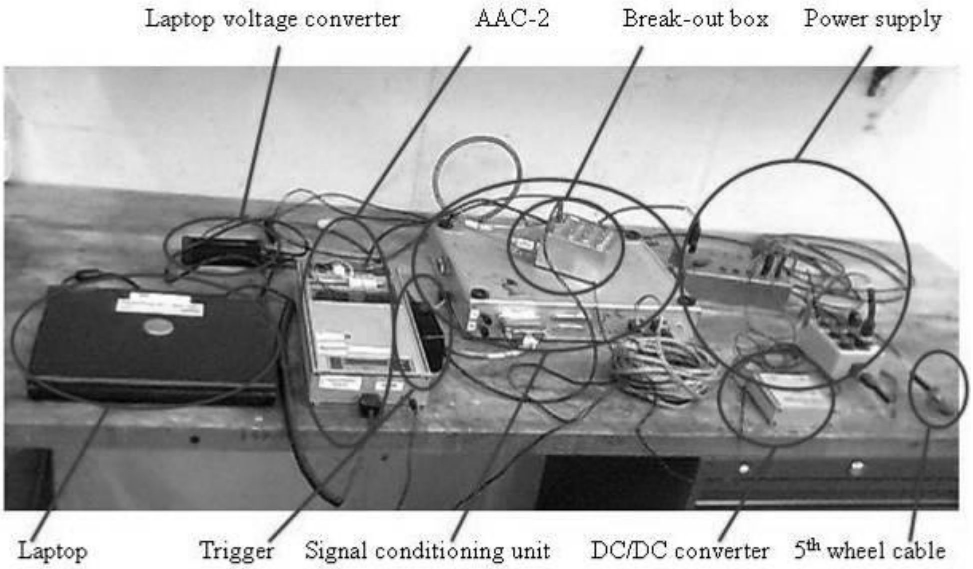

Figure 9: Measuring equipment for 5th wheel and trigger

Finally the components are all connected and driven by the 24V auxiliary system of the truck. As a result a DC/DC converter has to be applied to produce the 12V needed for the AAC-2. The other components use the power supply box as source. From that box there is one 12V output that is used to provide the laptop with the right voltage.

4.2.4 Coast down post-processing

The data accumulated in Easyview is exported as a txt file in order to open the data in Excel. The data file is cut in Excel to only show the values of where the measurement occurred, for simplification all runs were arranged after the same starting speed of measurement. The data string was ended when speeds dropped below 16 km/h, in order to avoid the low range neutral gear affecting the resistance. After the first data clean-up the runs are plotted in order to find individual runs diverging, and in worst case exclude them from the averaging.

The different runs that are significant are then averaged for both straights, the data are then converted from pulses into speeds by dividing it with the calibration factor. The results from the two straights are then combined in order to lessen the effect of wind and road inclination differences. The time is now plotted relative to speed in order to make a curve fit where a second order polynomial equation is describing the speed relative to time. The equation is now used to calculate the time elapsed between some vehicle speeds. The speed difference divided by the elapsed time is equal to the deceleration. The force acting on the truck at a certain speed is then obtained by multiplying the acceleration with the mass. Force versus speed can now be presented by a second order polynomial, the equation contains three terms that are describing the load curve used in the chassis dynamometer.

4.2.5 The Hällered course

The tests were performed at Hällered in February and March during some cold weather and somewhat wet track. The coast down was carried out on both straights of the main track, see Figure 10, mostly on lane one, even if lane two was used on some occasions when hindrances were present on the first lane. At Hällered the road at lane one is fairly uneven and can therefore cause a lot of vertical movement of the heavy truck and consequently result in losses in the suspension system and especially if the suspension is soft and is allowed to travel a significant distance. With 40 ton of mass pushing from behind the issue with bumpy roads is probably small, but still causes losses to some extent.

Performing coast down in the winter might result in issues concerning snow. At the test days no snow was present on the track, but the big barriers of snow at the side of the track were melting and water were pouring down the track. The water's impact on the rolling resistance is however difficult to measure. Naturally more water increases the rolling resistance but to what extent is difficult to distinguish, without some kind of test. The risk of getting heavily varying results from day to day is big and because no quantifiable measurement was done for the water, the error could possibly affect the results significantly.

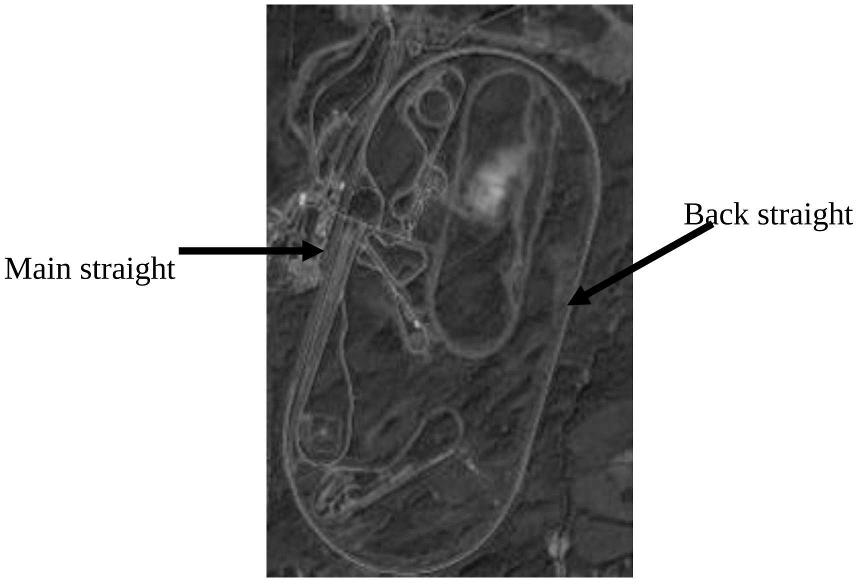

Figure 10: The Hällered track with the two long straights, used for coast down, marked

The weather conditions at the test days were probably not favourable because of cold weather at an average temperature of -4 ºC. All the data from the different test days can be found in Appendix A 1. The temperature affects the hysteresis of the tires, preventing them from deforming as much as they would at summer times, leading to lower rolling resistance. The temperatures also contribute to differences in air density. Density of air is affecting the tire pressure and the aerodynamic resistance. By inflating the tires at -4 ºC the equivalent pressure, (due to different densities), at +15 ºC, is around 4 % higher. The same percentage applies for the aerodynamic drag. Exact values of the air density change is calculated in a Matlab-code, Appendix C (Matlab-code for air density calculation), and presented in Table 1.

Table 1: Air densities at the different tests

Vehicle:	Air density at test day [kg/m^3]	Percentage difference to reference vehicle [%]
Volvo FH at 15 ºC, reference[3]	1.228	-
Scania R-series	1.283	4.3
Mercedes Actros	1.280	4.1
Volvo FH	1.294	5.1
Renault Premium	1.268	3.1
DAF XF	1.268	3.2

The wind is shown in Appendix A 1 and the direction is referred to angles clockwise counted from north winds, which are 0º. The Hällered main track straights are directed at 18º from north direction[4]. The wind is naturally affecting the aerodynamic resistance, but by averaging the results from both straights the influence of wind can to some extent be cancelled. Worst case is if the wind is hitting the truck from the side resulting in an increase of drag at both straights.

4.2.6 The coast down tests

Before the testing of each vehicle some vehicle specifications had to be stored. Not all information was possible to log because the vehicle was borrowed, but the major contributor to force differences are stored, such as weight, aerodynamic spoilers and vehicle tires. The information is saved in order to easily compare and later on conclude what parameters caused the differences in the results. The vehicle specifications are also important for future testing and follow-ups. Most of the data is presented in appendix, while more abstract devices, such as side skirts are described in the text for each vehicle.

4.2.6.1 Scania coast down

The Scania R480 used in the coast down test is equipped with a full size sleeper cabin and therefore does not need any large roof deflector in order to create a smooth transition between the cabin and trailer. The steep front can however worsen the aerodynamics, because of a larger region where stagnation pressure is reached. The vehicle also utilizes a lot of add-ons such as side and front mirrors and sunvisor, which does not improve the flow around the front of the vehicle, the shape can be seen in Figure 11.

The transmission used was Scania's 12 speed opti-cruise gearbox and a final drive ratio of 2.71:1. The torque is finally transmitted to a dual pair of tires on each axle with a rough rubber pattern.

Figure 11: Scania R480 Euro V

The vehicle was driven to Hällered the week before the test and had been parked outside which meant a roof covered with snow, and unfortunately a drained battery. The vehicle along with the trailer was driven to a wash point where the roof was cleaned by a brush. Some snow was still present after the cleaning of the vehicle. The snow did not fall off during the test either so it both contributed to a slightly larger mass as well as less attached flow over the trailer surface. The vehicle were then driven to the weight measuring location where tire pressure and weight distribution were measured, the weight distribution is shown in Appendix B 1. The tires were filled to 8.3 bars (120 psi) overpressure at -4 ºC, the equivalent pressure at +15 ºC is above 8.6 bars.

The Scania R-series had a rigidly mounted roof deflector and could not be optimized for the trailer used, but it matched the trailer height approximately. The pneumatic suspension was set in normal drive mode in order to simulate correct ride height, (that is however not equal to optimal ride height for fuel consumption. Rather a compromise between ride quality and fuel consumption).

To achieve reliable results from the coast down the vehicle should be sufficiently worn-in, in the Scania case the vehicle only had run for about 1000 km which is not enough to fully allow all bearing surfaces or gearwheel contacts to embed. But brake pads and wheel angles are assumed to be close to ideal because of the short time of vehicle use, and thereby less time for uneven wear of pads or distortion of the suspension. The tires were also of a rough model with very little wear and probably affected the rolling resistance badly, the model was Bridgestone M 729, a tire suitable for winters.

When performing coast downs a significant amount of warm-up driving is required in order to build-up normal operating temperature in the driveline components and tires. Normally the temperatures in the driveline would have been monitored but due to the fact that the vehicle was borrowed none such sensors were fitted. After one and a half hour of warm-up the 5^{th} wheel was calibrated to a fixed distance in order to relate number of pulses to distance rolled.

The coast down test starts at the back straight, seen in Figure 10, from 90 km/h and continues with the same procedure at the main straight with a start speed of 90 km/h. The next time around the start speed of the two straights was down to 75 km/h which is an overlapping speed of around 5 km/h from the finishing speed of the first run. The procedure followed until the low range neutral gear was engaged at around 16 km/h. When the gear engages the resisting force changes quickly and is therefore not representative. On some of the runs there were obstacles on the lane and consequently a lane change had to be performed, the effect of such manoeuvre was however very small and consequently the results from these runs were still valid. When the vehicle turns the rolling resistance increase, because of the side force and slip-angle caused by the turning tire, but with very small steer angles the side force effect becomes negligible.

Four different runs were performed on each side of the track in order to get statistically credible result. The coast down time variation of the different runs were however small and all runs were thereby used for calculation of an average run, the individual runs is presented in Appendix D 1. The wind was almost directed perpendicular to the truck on both straights, but still a considerable difference in aerodynamic drag was noticeable between the two straights, see Figure 12. The probable reason was that the back straight had less windbreaks than the main straight, and that there were slightly more headwind on the back straight, due to the wind direction. Water was present on both straights and most likely caused an equal increase in rolling resistance, which also can be seen in Figure 12 at the low speeds.

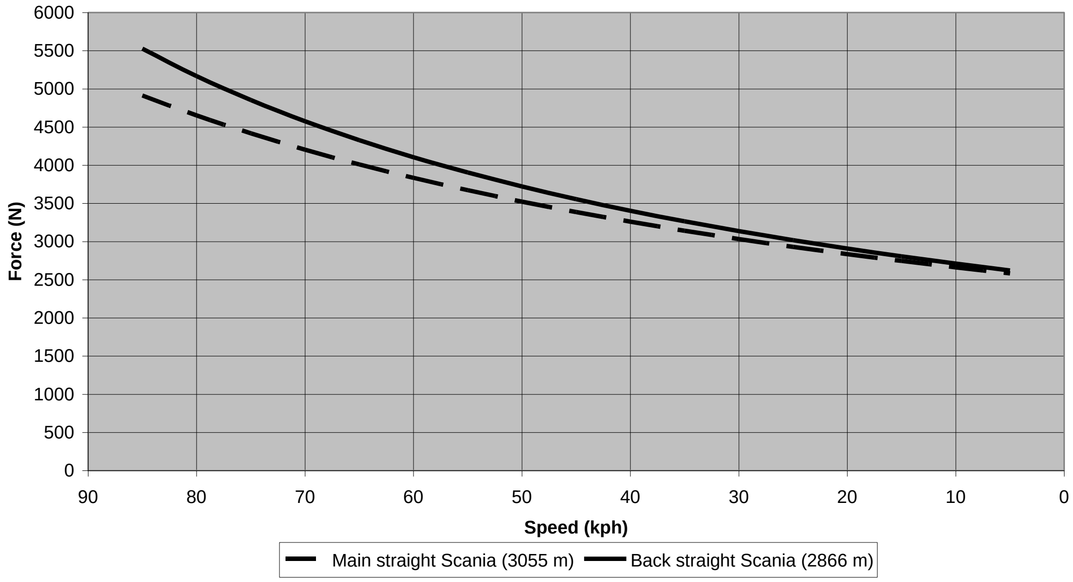

Figure 12: Scania R-series, force vs. speed graph for both straights at Hällered coast down, values in brackets are the total distance coasted

4.2.6.2 Mercedes Actros coast down

The Actros used for comparison is a Euro V vehicle equipped with the megaspace cabin as well as roof and side deflectors. Similar as for the Scania a lot of add-ons were used, such as the extra mirrors, sunvisors and roof mounted compressor horns. But also drag improving spoilers such as, front spoiler and side skirts, presented in Figure 13.

The powertrain of the Actros starts with their powershift gearbox, which is an automated sequential transmission. The final drive is made for highway cruising with a low gear ratio of 2.533:1, but the highway engine speed is still not lower than for the Scania, because the Mercedes run on tires with 11 % smaller diameters. The low gear ratio together with smaller wheel radius allows the mass of the powertrain to decrease. Then the Mercedes also rest on rims made of aluminium which potentially can decrease the rotating masses even further.

Figure 13: Mercedes Actros 1848 Euro V

The Mercedes was driven to Hällered prior to the test and was covered with snow, a quick cleaning of the vehicle was done, but not all snow was removed. The weighting process as well as the tire inflation was the same as for the Scania, once again 8.3 bars was inflated into the rubbers at -4 ºC.

A more embedded transmission was found on the Mercedes compared to the Scania, where odometer read-outs showed 10´000 km. The distance is however not up to the 30´000 km limit that is assumed to be the point where no further gains in bedding of bearings are seen. The distance can nevertheless be enough to severely distort some of the original suspension geometries. Low-speed resistance showed a clear tendency of being much lower than the Scania's so presumably that was partly due to the bedded components. Another reason why the rolling resistance was lower for the Actros were the more effective rubbers used, but when no data of tire characteristic is given, it is hard to predict the real benefit from the Michelin Energy XDA2+.

A big issue with this Mercedes was the driving mode of the air suspensions which was set far too low. With the standard height of the suspension the wheels were scratching the inner wheel arch, therefore an arbitrary height of the suspensions were chosen in order to avoid contact. Another issue relating to the air suspensions were the trailer which made some sound from the bellows in the rear. Apparently the system was leaking and presumably made the rear of the trailer lower than usual and therefore changing the aerodynamic of the vehicle, see Figure 14. This is however not the Mercedes and thus not having the same height of the fifth wheel, (not the 5th wheel measuring speed). But it gives an indication of the slanting trailer, and with such an angle the mud-flaps even touched the ground leading to increased rolling losses.

Figure 14: Leaking rear air suspension of trailer

Before the test the 5th wheel had to be calibrated again, this time it rotated a bit faster, but in such a small range that the effect probably was due to air leakage of the bicycle tire. The procedure of coast down was carried out in the same way as for the Scania, except for the first two runs that were conducted with just half an hour warm-up. Results were not varying a lot even with the short amount of preheating therefore the runs were included in the averaging. Round seven was ruined by a measurement error, but with help of interpolation the complete run was still salvaged. For further information of the runs see Appendix D 2.

In Figure 15 the average coast down result on each straight is shown. As one can see at the low speeds the Mercedes had similar rolling resistance on both straights, but the aerodynamic drag varied between the straights. From the weather data, Appendix A 2 it can be seen that the wind is almost a pure head wind at the back straight, (and the opposite at the main straight). During the test it was also observed that less water was present on most parts of the track, compared to the Scania.

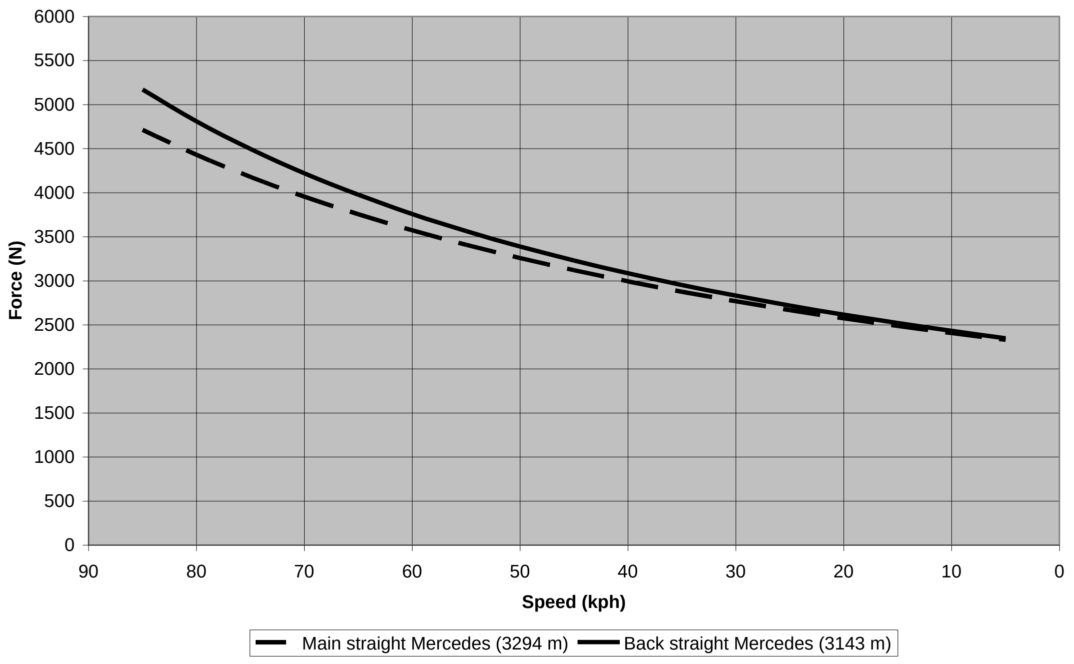

Figure 15: Mercedes Actros, force vs. speed graph for both straights at Hällered coast down, values in brackets are the total distance coasted

4.2.6.3 Volvo FH coast down

In order to evaluate the relevance of the competitors tests a Volvo FH was also coasted at Hällered. The vehicle was borrowed from colleagues at Hällered and was only used during the coast down, another similar truck was used for the fuel and emissions tests. The coast down vehicle was equipped with the small globetrotter roof deflector, side fairings and side skirts. In order to check for failures on the trailer (T052) that was found leaking in earlier tests, another similar trailer (AA) was used as a reference, see Figure 16. Other failures that could have been present on the T052 trailer were high brake drags.

The vehicle was well maintained because it was used frequently, the tire pressure and weight of the vehicle was however not measured, but the same trailer for a similar truck was weighted at RPG (Råda Proving Ground), earlier at the autumn. The same weight was approximated for this tractor and trailer combination. The tire pressure had recently been checked according to staff at Hällered. The tires used were the Michelin XDA2 which were slightly worn in comparison to the Mercedes and Scania tests.

Figure 16: Volvo FH500 with AA-trailer used for coast down

During the coast down the course was undeniably wetter on the main straight, most likely because the snow banks at the sides where melting and water flowed across the track. While the back straight was almost completely dry. The rolling resistance should therefore be significantly different between sides. Similar conditions were present for both trailers. The wind was not changing during the test day except for marginally higher peak gusts during the AA-trailer test, see Appendix A 3.

Only two coast downs on each straight and trailer were performed, which results in a large uncertainty. The results in Figure 17 indicate that the AA-trailer have lower rolling resistance than the T052-trailer. Worn tires were used on the T052-trailer while the AA-trailer had rather new sets of rubber, which normally contributes to higher rolling resistance. The T052-trailer had a fault in the air suspension, a leakage in one of the air cushions, this lead to lower ride height of the trailer at the rear. But this would hardly influence the rolling resistance at all, except for the mud-flaps that slid on the ground. The difference in resistance can not be said to be due to measurement errors either, because 4 runs with each trailer were performed, and all of them showed the same tendency, either on dry or wet surface.

The AA-trailer was later on used for fuel consumption tests at road and continuously gave lower values than the tests carried out in the chassis dynamometer, based on the T052 trailer. Therefore it can be assumed that the AA-trailer's axles were better maintained and adjusted and therefore rolled more easily, alternatively the tires on the AA-trailer were more efficient.

The aerodynamics is another factor that varied between the trailers. The main observation is thou that a head-wind was present on the main straight, and thereby increasing the drag for those test, it affected the AA-trailer more, but probably this is just cycle to cycle variations. With more trials the aerodynamic increase would probably be less severe for the AA-trailer. Still a considerable difference was present, for the recently mentioned trailer the side skirts were of a different sort, so was the trailer height which was five centimetres higher. Five centimetres is however not enough to affect the aerodynamic loss in the magnitude seen in Figure 17, but as the roof deflector was not adjusted to fit the trailer perhaps some of the losses can be explained.

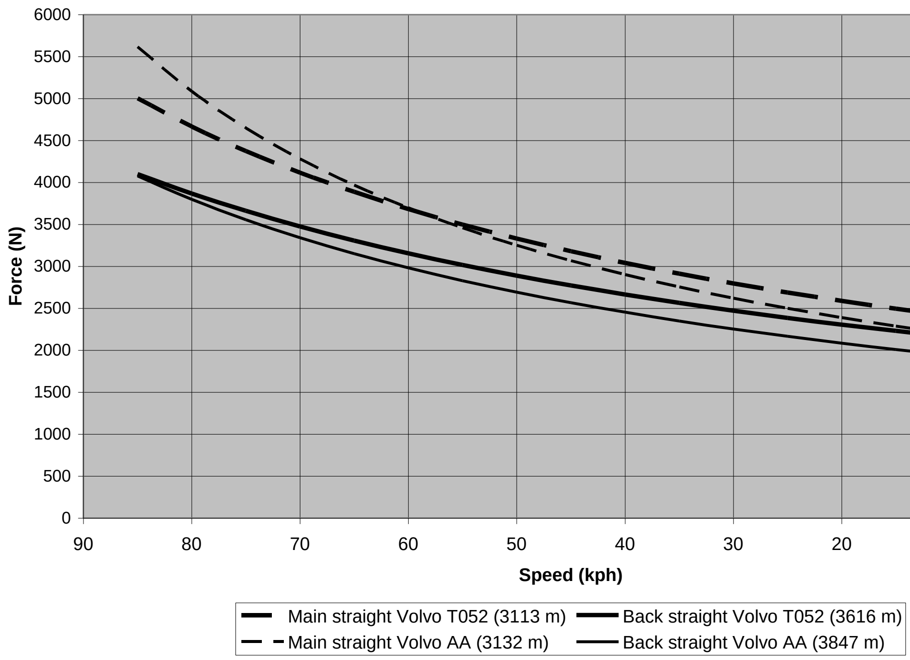

Figure 17: Volvo FH, force vs. speed graph for both straights and trailers at Hällered coast down, values in brackets are the total distance coasted

4.2.6.4 Renault Premium coast down

A Renault Premium was sent to Lundby from Lyon to also participate in the test. The vehicle is hardly the same type as the other full size long-haulage trucks, and thereby hard to compare. The main issue with the Premium was however the absence of roof and side deflector, and sleeper cabin, seen in Figure 18. The pressure drag of the vehicle will drastically increase as well as losses caused by a large separation region around leading edges of the trailer. The increased trailer gap will also cause turbulence as well as increased sensitivity to side winds[5]. On the day when the Renault was driven the wind was perpendicular to the straights, causing worst case scenario for the vehicle. Wind speeds were nevertheless small so hopefully it did not affect the test significantly.

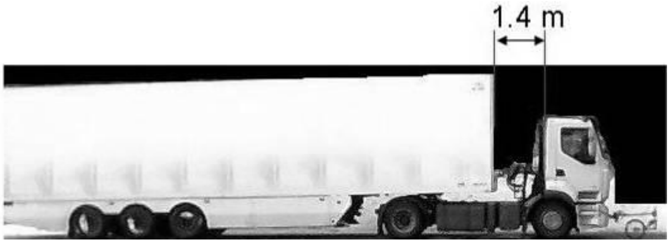

Figure 18: Renault Premium used in coast down

Weighting occurred at Hällered where also the tires pressure were checked and adjusted to 8.3 bars. The only issue found at the preparation was once more the leaking air suspension. As can be seen in Figure 14 the rear end was clearly dipping and causing the mud-flaps to touch the road. The aerodynamics could perhaps be slightly improved by the slanting trailer, as long as the flow stays attached to the trailer roof it will direct the wake downwards and decrease the turbulence.

The vehicle had run for 17600 km and was not completely worn-in according to the rule of thumb, it utilized Michelin’s Energy tires all around, but with steel rims. As can be seen from Figure 19 the rolling resistance was higher at the back straight, on this occasion the roads were visibly drier on the main side, while the back straight still got some wet areas. The average rolling resistance for both straights is at least undoubtedly lower than all other trucks tested so far, meaning that wetness of the road is a major matter affecting for rolling resistance.

From Figure 19 it can be seen that the resisting force increases rapidly after approximately 50 km/h. When the vehicle was sent to the chassis dynamometer, they found out that the vehicle does not allow neutral gear to be engaged at speeds above 50 km/h. The high forces are a consequence of the engine braking, which especially has a large impact on high engine speeds.

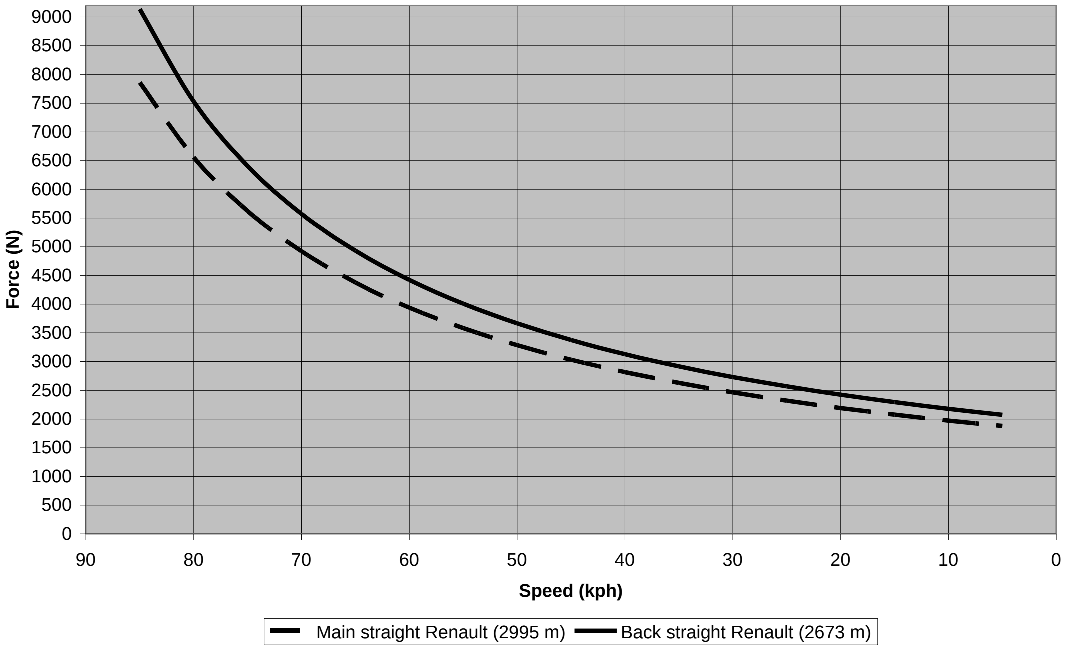

Figure 19: Renault Premium, force vs. speed graph for both straights at Hällered coast down, measured result, values in brackets are the total distance coasted

A new road load profile was proposed by using only the vehicle speeds below 50 km/h and then extrapolating the forces up to 85 km/h. Relevant loads should be found at those speeds, and as long as there is a good relation for the 2nd order polynomial the extrapolation should be fairly accurate. The extrapolated load at 85 km/h was correlated to the theoretical force acting on the vehicle with the particular specifications, see Equation 1.

$$F = \frac{1}{2}\rho A C_D v^2 + mgf_r = 0.5*1.268*9.7*0.82*\left(\frac{85}{3.6}\right)^2 + 39630*9.81*0.0052 = 4831\,N$$

Equation 1

The constants are obtained from the vehicle, the mass is the measured weight, and the coefficient of rolling resistance is calculated from the force at 5 km/h (assuming no aerodynamic resistance). The frontal area is approximated from the trailer width 2.55 m times the trailer height of 4 m and finally by subtracting 0.5 m^2. The drag coefficient is obtained from tabulated values of trucks with the specific gap and height distance between the tractor and trailer. The theoretical force value is a bit lower than the extrapolated force at 85 km/h in Figure 20, but the formula neglects side winds and speed depending rolling resistance and therefore a more realistic force would probably be around 500 N higher and then in between the two straights' force shown in Figure 20.

The large differences between the two curves are due to the extrapolation and side winds that were directed perpendicular to the truck during that day of testing. The calculated value is however suggesting a similar result as the extrapolation proving that the forces are at the right magnitude.

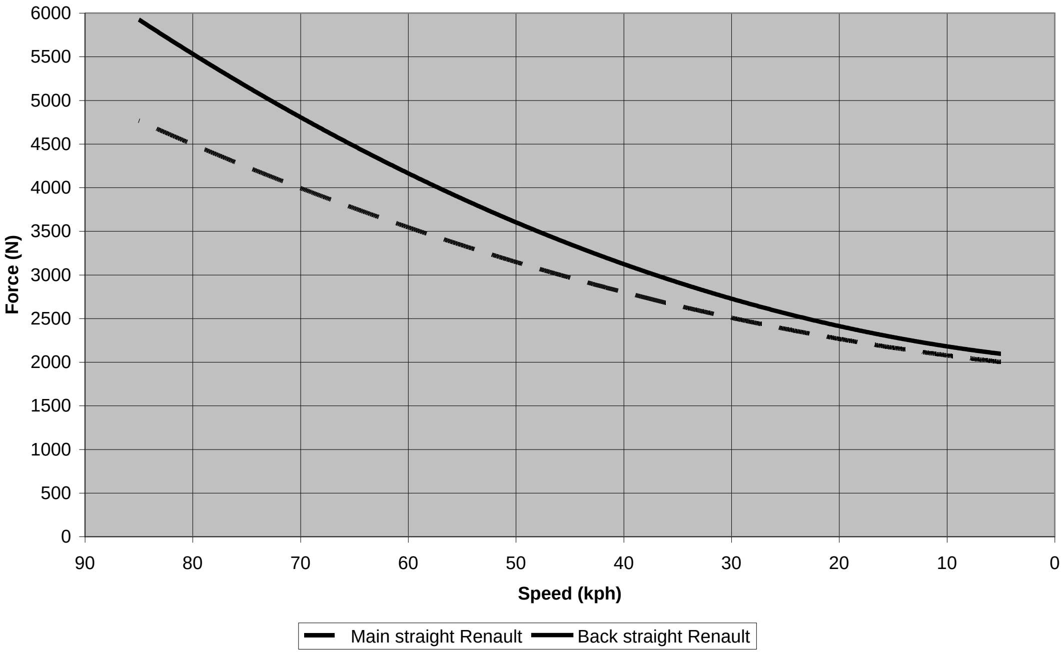

Figure 20: Renault Premium, force vs. speed graph for both straights at Hällered coast down, extrapolated results

4.2.6.5 DAF XF coast down

DAF XF was the last vehicle to be analyzed at the Hällered coast down test. The shape is comparable with the Scania, because it posses the flat front with small smooth curves at the corners of the cabin, see Figure 21. The form is however very effective when it comes to space, but causing a lot of separation around the leading edges and a larger stagnation pressure region. The super space cab was used, which means that a small fixed shaped roof deflector is enough to create a smooth transition between cabin and trailer.

At Hällered the vehicle was weighted and tires were inflated to 8.3 bars as all the competitors. The indoor scale was not available and therefore the outer one was used, which presumably had the same accuracy, for weights see Appendix B 1. The vehicle itself was well used with an odometer reading of 120´130 km, no further inspection of brakes or suspension was done prior to the test, but they were probably in good shape. The tires were supplied by Goodyear and were resting on aluminium rims.

The weather situation at Hällered was improving and temperatures were approaching -2 ºC, and due to the sun both straights was dry. The wind was also weak but occasionally some gusts were present affecting the results momentarily. The wind was mainly headed straight towards the front of the vehcile at the main straight, also see Appendix A 5.

Figure 21: DAF XF105 EEV Euro V

The test was conducted in the same manner as usual, starting with calibration of the 5th wheel, the calibration factor were continuously increasing from test to test, suggesting that the tire were leaking and thereby decreasing the rolling radius. Only 3 runs on each straight were conducted due to consistency between the runs, the only obvious issue was the head wind at the main straight that clearly increased the losses at that side, see Appendix D 5.

In Figure 22 the coast down result for the DAF is presented, the main straight where the wind was blowing towards the front of the vehicle has clearly larger losses. Strangely enough even the rolling resistance were higher, this is explained to be either a small inclination of the straight or increased rolling resistance due to more bumpy road surface. Aerodynamically related losses differences of the DAF are affected by the wind, but still fairly small differences were observed.

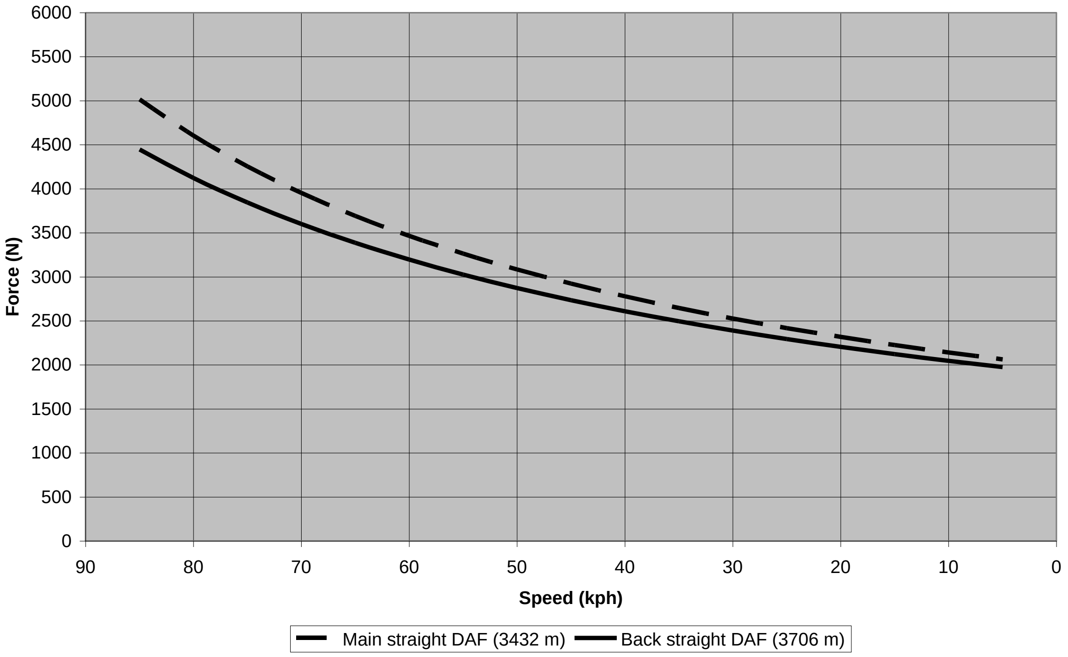

Figure 22: DAF XF, force vs. speed graph for both straights at Hällered coast down, values in brackets are the total distance coasted

4.2.7 Analysis and comparison of coast down

The coast down test is performed in order to obtain the load curve for a specific truck trailer combination. The load curve can then be implemented into VFL the chassis dynamometer, to enable vehicle simulations close to ideal conditions. The load curve can also provide a lot of information concerning both aerodynamic and rolling resistance. The rolling resistance and the driveline losses are nearly the only forces acting on the vehicle at low speeds, (below 10 km/h). Parameters affecting the rolling resistance can therefore easily be distinguished, by comparing vehicles with different tires or weather conditions. The aerodynamic resistance is on the other hand harder to interpret from the load curve, even if some hints of how well the aerodynamics works can be seen by looking at the difference between resisting force at full speed and low speed.

For some of the vehicles the difference in resisting force at high and low speed is significantly higher than for others. This phenomenon occurs partly because of different aerodynamics, but also because different tire manufacturers have tires with varying characteristics at different speeds. Driveline losses are another parameter that increases at higher speeds. Factors such as wind and higher air density also affects the vehicle and especially at higher speeds where aerodynamics becomes an increasing issue.

4.2.7.1 Aerodynamic comparison

Figure 23 presents the force increase relative to the resistance at 5 km/h, from the graphs it is fairly obvious what kinds of forces that dominates on the different trucks. The Volvo FH driven at summer has the best performance at high speeds, which are due to the combination of low wind speeds, low air density and an efficient cabin. If comparing with the similar Volvo driven at winter the wind speed and the air density were higher at the winter test.

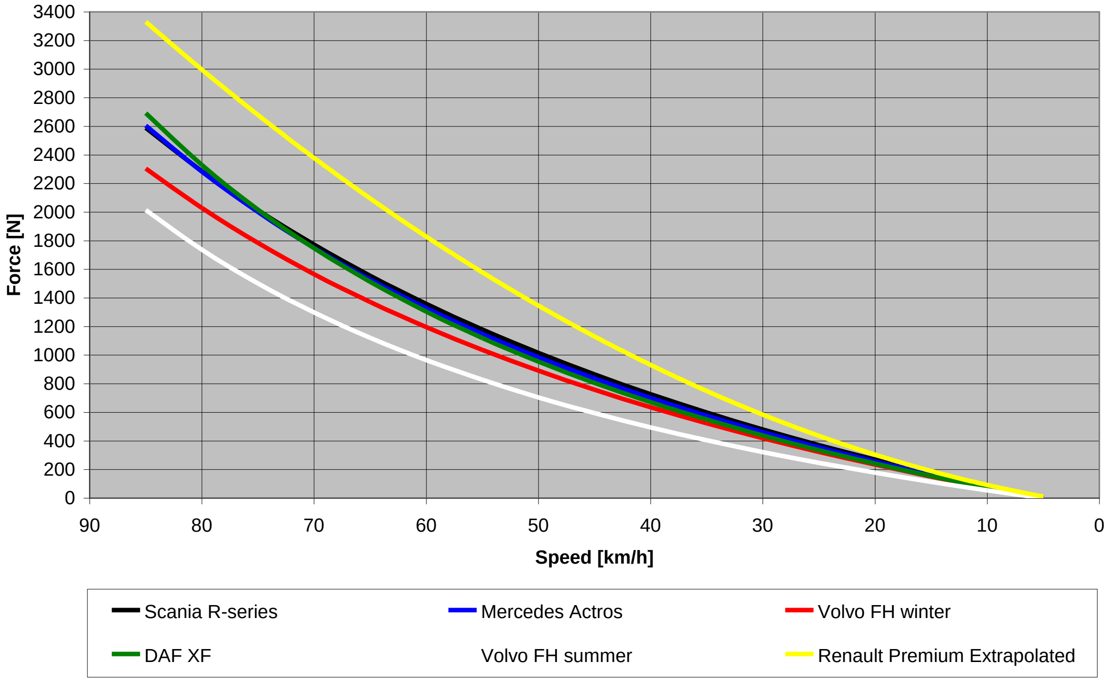

Figure 23: Force increase relative to the resistance at 5 km/h

By assuming a constant rolling resistance of the tires, an approximate value of the drag coefficient can be obtained. A second order polynomial relation is however a better way of approximating the rolling resistance[6]. In the following calculation example some approximations concerning wind speed and direction are done. First of all an assumed head wind is used for each case proportional to half of the averaged measured wind speed due to the advantage of having a back wind on one straight and head wind on the other. Even if the wind is helping reducing the relative speed on one straight the gain in having back wind is not as big as the loose in having head wind, therefore an approximation in using the half of the head wind speed as average for both straights is done. When the wind becomes angled a corrected drag coefficient with different yaw angles are calculated in order to get a feeling on how much the wind affects the drag coefficient[7]. The results in Table 2 are calculated from Equation 2 which is the general equation for aerodynamic drag force, including the wind approximation.

$$\frac{F_D}{\frac{1}{2}\rho A\left(v+\frac{v_{wind}}{2}\right)^2}=C_D$$

Equation 2

Table 2: Calculation example of aerodynamic resistance at coast down assuming constant rolling resistances, values in brackets are only used for yaw angle calculation

Vehicle:	**Air density (ρ) [kg/m³]**	**Frontal area (A) [m²]**	**Force at 85 km/h (F_D) [N]**	**Vehicle speed (v) [m/s]**	**Wind speed (v_{wind}) [m/s]**	**Drag coefficient (C_D)**	**Yaw angle [º]**	**Corrected drag coefficient (C_D)**
Scania R-series	1.283	9.7	2600	23.6	(3.5)	0.75	8	0.62
Mercedes Actros	1.280		2600		(1)	0.75	2.5	0.71
Volvo FH winter	1.294		2300		3	0.58	-	0.58
Renault Premium	1.268		3300		(1.5)	0.96	3.5	0.91
DAF XF	1.268		2700		2.5	0.71	-	0.71
Volvo FH summer	1.228		2000		0.5	0.59	-	0.59

The calculated drag coefficients from Table 2 are higher than expected, (0.55 approximately for a FH with semi-trailer), but if a speed dependent rolling resistance was to be used a much lower amount of the force would be aerodynamically related, and then a lower C_D would be obtained. The air density was also varying between the tests and C_D values could also vary as much as 5 % between the vehicles.

One general conclusion is still that the two Volvo FHs had similar drag coefficients and also the lowest in magnitude. Some reasons why the Volvos managed better than the main rivals, could be the high water level on the road for the Scania and Mercedes tests. More water generally results in a stronger speed relation of rolling resistance, which was not included in the above calculation. A stronger speed relation of the rolling resistance would decrease the aerodynamic resistance and consequently lower the drag coefficient. Wind is of course an issue and when not logged continuously the wind blowing towards the vehicles could be far different from the average speeds used in Table 2.

4.2.7.2 Rolling resistance comparison

The obvious way to compare rolling resistance, (including driveline losses), is by looking at the forces acting on the vehicle at low speed, the force at 5 km/h is presented in Table 3. One can easily distinguish the large difference between performing the tests in winter and summer. At 5 km/h most driving losses are in the form of rolling resistance but some hundred Newtons are caused by the frictional losses in the powertrain, the aerodynamic contribution are negligible in comparison to the other two. In order to approximate rolling resistance coefficients the driveline losses have to be separated from the total force at 5 km/h, which can be seen in Table 3. The Powertrain losses used are for the Volvo FH tested and described in the powertrain coast down part, Chapter 4, Section 4.1. The same resistance are used for all vehicles, because the differences between the various transmissions are assumed to be small. The main contributor of the inertia of the powertrain is the wheels, which are fairly equal between trucks. The powertrain resistance used is the one for the warm transmission and at 5 km/h, see Figure 6.

Table 3: Forces acting on vehicles at 5 km/h

Vehicle:	Force at 5 km/h (F) [N]	Powertrain losses part (F_P) [N]	Rolling resistance part (F_R) [N]
Scania R-series	2600	204	2396
Mercedes Actros	2350	204	2146
Volvo FH winter	2200	204	1996
Renault Premium	2000	204	1796
DAF XF	2000	204	1796
Volvo FH summer	1450	204	1246

With the help of the rolling resistance forces measured from tests, the coefficients of rolling resistance can be obtained for each vehicle by using Equation 3. The mass of the vehicle was measured at Hällered on the day of coast down testing, the gravitation used is 9.81 m/s^2 and the results are found in Table 4.

$$f_r = \frac{F_R}{mg}$$

Equation 3

Table 4: Coefficient of rolling resistance

Vehicle:	Rolling resistance force (F_R) [N]	Mass of vehicle (m) [kg]	Coefficient of rolling resistance (f_r)
Scania R-series	2396	40790	0.0060
Mercedes Actros	2146	40875	0.0054
Volvo FH winter	1996	39700	0.0051
Renault Premium	1796	39630	0.0046
DAF XF	1796	40940	0.0045
Volvo FH summer	1246	39620	0.0032

From Table 4 it can be seen that a large variation in rolling resistance coefficients was seen. Naturally weather and differences in tire wear affected most of the tests. Normally a scrubbed set of energy tires is assumed to have a rolling resistance of 0.0051 but dependent on the wear it can drop down to 0.0041 as well. The largest contributor to the rolling resistance is the trailer, in this case the trailer used well worn energy tires so assuming a rolling resistance coefficient of 0.0041 for the trailer seems realistic. The tires for all the tractors were considerably less worn, so an increase in rolling resistance coefficient is probable. Factors such as tire pressures, road surface condition and outdoor temperature also affected the rolling resistance from test to test. For the Scania the tire pattern was coarser and the road was wet which probably were the responsible factors for the high coefficient. The Mercedes had similar road conditions but much more effective rubbers when it comes to rolling losses. The Volvo at winter had a lot drier road and the Renault and DAF even had completely dry roads. As can be seen their coefficients were still a bit higher than that approximated for the trailer.

The largest surprise is of course the reference truck which has significantly lower rolling resistance than that assumed for the trailer. The explanation is that the weather conditions were optimal, the tire pressures were set to the maximum value and the vehicle’s suspension was serviced to decrease the rolling losses. The vehicle was also fully pre-heated and realistically a lower contribution of the powertrain loss than assumed here was present. If the powertrain loss was halved the rolling resistance coefficient would lie around 0.0035 for the Volvo FH at summer.

4.2.7.3 Complete vehicle resistance at coast down

The total driving resistances of the vehicle is presented in Figure 24. Especially noticeable is the large difference between the truck driven at summer versus the ones driven at winter. The same trailer loaded with the same weights was used in both occasions. The tires were not changed during that period which concludes that rolling resistance only could have been affected by the tire pressure or road surface resistance as well as suspension geometries. On most of the winter tests the road was wet, but for the Renault and DAF the track was perfectly dry. This concludes that 500 N of rolling resistance difference have to be explained by lower tire pressure, the scratching mud-flaps, the less efficient suspensions or the colder weather, resulting in more powertrain losses as well as another hysteresis characteristic of the tires. The cold weather should however normally decrease the hysteresis of the tire and thereby decrease the rolling resistance.

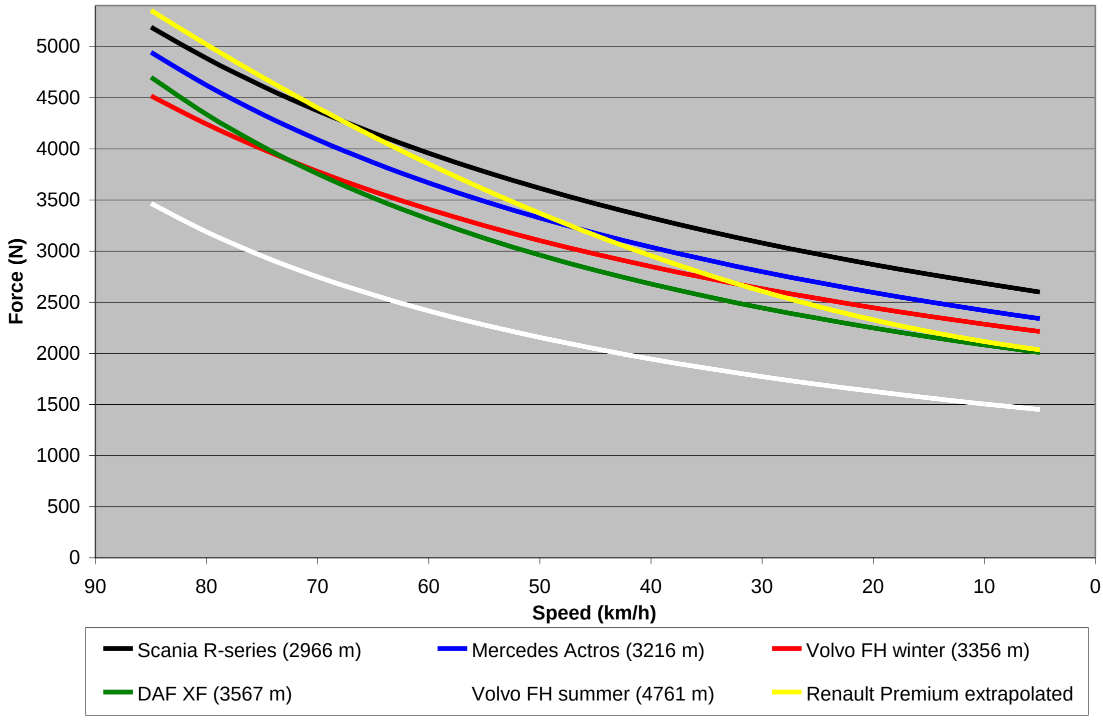

Figure 24: Coast down comparison of competitors of the average run at both straights

4.3 PERF (Performance calculations) simulations

PERF is one of the computer simulation tools used at Volvo presently. The simulation tool is used for comparing results that are achieved either from real world testing or chassis dynamometer simulations. The program is using the governing equation of motion for vehicles, see Equation 4[8].

$$F = \frac{1}{2}\rho A C_D v^2 + mgf_r + mg\sin(\alpha)$$

Equation 4

Other known parameters are the load curve of the engine, together with the ratios, losses and efficiencies of the rear axle and gearbox. By also defining the limiting friction of the tire/asphalt contact the performance criteria can be calculated. These criteria can be anything from startability to acceleration parameters.

By letting the vehicle drive a predefined route the fuel consumption can be obtained. There are however a lot of assumption such as the way the vehicle follows the target speed, the influence of weather e.g. wind, rain and temperature differences. A pre set wind can be added to the calculation in order to create more realistic drives, but other climate influences are missing.

4.3.1 Issues with PERF simulations

Even if good approximations of the different driving conditions are used, it is still difficult to fully simulate the variability of real world driving. For an automated manual transmission it is however possible to obtain realistic driving scenarios because the gear shifting degree of freedom is following the same control laws, both in PERF and in real life driving. The degree of acceleration is however different between the two types of test, as well as the foreseeing ability a driver possess'. The main issues are however more related to the difficulty approximating the physical parameters. The governing equations in PERF are all listed below as Equation 5 - Equation 9.

$$F_{trac} = \frac{T_{eng} * i_g * i_f * \eta_{trans}}{r_e}$$

Equation 5

$$F_D = \frac{1}{2} * \rho * A * C_D (v + v_{wind})^2$$

Equation 6

Where:

$$v = \frac{\omega_{eng} * \pi * 2 * 60 * r_e}{i_g * i_f * 1000}$$

Equation 7

$$F_R = R_m * m$$

Equation 8

$$F_g = 0.01 * 9.8 * S * m * 1000$$

Equation 9

S in Equation 9, is the gradient inclination specified in percentage, mathematically expressed as height travelled divided by length travelled. The mass constant (*m*) is here expressed in ton. And as a consequence the rolling resistance coefficient R_m is based on the mass in ton, the gravitational term is also included in the rolling coefficient, thereby it has the unit Newton per ton.

The tractive effort specified in Equation 5 is also limited to the maximum force the tires are able to transmit due to coefficient of friction. The friction coefficient is however simplified as just one value, not dependent on slip, but for most driving cases the tractive effort never reaches close to the maximum limit. The disadvantages of describing the drive-cycle with help of above governing equations are the following:

- Drag coefficient is the value describing the shapes effectiveness of moving through air, the value preset in PERF is specific for 80 km/h. For a truck the shape factor varies with vehicle speed, this effect is however small and is mostly due to the size of the laminar boundary layer. For a truck where gaps between the different vehicle parts are big, the boundary layer growth can be observed at different regions of the truck and consequently affect with a larger magnitude then for a car for instance. In PERF there is however a sentence in the results indicating that a certain speed dependence is used, but no relation of this kind is shown or predefined.
- A wind speed can be set in order to simulate more realistic behaviour, the issue is still that winds are never constant in the real world and can also blow from behind the truck. The probably most critical wind is also side winds where a yaw angle towards the truck is created. Such effect changes the drag coefficient significantly, partly because of more interference around the trailer gap but also because the heading direction relative to the wind is not parallel to the length of the truck anymore. The shape of the truck is as a result changed and so is the effective frontal area.
- Air density can easily change the drag force with around 5 % between summer and winter due to the increased viscosity in cold air. The main factors affecting the air density is temperature, pressure and humidity. In PERF a fixed value resembling normal density is used for all simulations.
- The effective rolling radius is not constant for all speeds as defined in PERF, because the centripetal force increases the active wheel radius at high speeds.
- Tire rolling resistance coefficient is speed dependent in a second order polynomial manner while a linear relation is approximating the dependency in PERF. Rolling resistance coefficient is also much dependent on the type of asphalt, tire properties, the amount of wear and the smoothness of the road. A typical constant initial value is used, because all these dependencies are very difficult to quantify. A lot of information is consequently lost, one example is up to 20 % difference in rolling resistance coefficient between new and well worn tires. Other parameters also affecting the rolling resistance coefficient are tire pressure, temperature of the rubber as well as the temperature of the inflating air. Finally parameters of the suspension are affecting the easiness of wheel rotation. Toe-in is one of them and is adjusted positively in order to create a stable vehicle, but the less toe-in that is used the lower the rolling resistance coefficient. Brake drag can also be a problem on vehicles even if trucks usually are not affected due to the pneumatic control of the brake pads. All these parameters are usually similar between trucks and well recreated in PERF, but especially when it comes to competitors it is difficult to approximate the effect of every single component. Consequently a large uncertainty of the actual rolling resistance is present.

- Vehicle mass can also vary during tests especially at long cycles where the amount of fuel can affect the weight with up to 2 % between full and empty fuel tanks of a 40 ton vehicle. An average mass of the cycle is suitable to use. When simulating distribution cycles and correlating them to customers' values, in practice the cargo weight is decreasing and as a result the fuel consumption drops at the end of the delivery. This is still not a big issue now that PERF simulations normally are compared to test cycles, and no loading or unloading is taking place there either.
- A significant amount of energy can also be lost by uneven roads resulting in suspension travel and energy consumed as heat in shock absorbers and springs.
- The transmission efficiency is affected by the amount of oil in the gearbox and final drive, the gears are usually allowed to splash into the oil in order to lubricate the contact surface, but some gearboxes are filled with more oil than others. The efficiencies are preset values representing the friction losses caused by the contact between gears, this loss is not constant but rather speed dependent, even if a good approximation can be obtained for the constant. The efficiency is also strongly dependent on oil viscosity and consequently temperature of the surrounding air. The temperature relation is unfortunately difficult to represent because it affects a lot of parameters and a lot of data would have been required to be stored in the program. Using fully warmed-up vehicles before the road test is also required in order to be able to correlate the road test results with PERF, as can be seen from the results achieved in the powertrain coast down part, Chapter 4, section 4.1.
- The engine maps of the particular engines are accessible in PERF but in order to decrease the amount of data there is only a limited number of torque (T_{eng}) and engine speed (ω_{eng}) combinations with the definite brake specific fuel consumption. All other loads and speeds are interpolated between the known data and as a result is not representing exactly the true value.
- When producing an engine map a predefined environment is used, by comparing this lab environment with the varying climate of a real world test the engine load can vary a lot, especially if air pressure is changed, for example at elevated locations.
- When simulating in PERF a fixed auxiliary loss is used, based on the normal requirements. The importance of having similar conditions at the real world test becomes large, auxiliaries such as the AC compressor can be turned off, but most others are driven when needed at road tests. Once again outdoor temperature is affecting, for instance the amount in which the cooling fan is required. A large increase in power is a result of having the cooling fan switched on instead of off.
- PERF simulations are based on ideal driving conditions, which include no steering effort. The absence of any resistance caused by steering, such as increase drag coefficient and increased rolling resistance, results in lower simulated fuel consumption.

- A final problem with simulation tools in general is of course the difficulty in recreating the driver's contribution, which is not only a problem for simulation programs, rather a major problem for all tests especially because humans are unpredictable and some variations can be seen from driver to driver or just test to test with the same driver.

4.3.2 PERF simulation results

To get some results to analyze, different types of vehicles are simulated with different technical configurations on two duty cycles. One duty cycle is Lv-Bo-Lv (Landvetter-Borås-Landvetter), which represents a typical drive for a long-haulage truck, see Appendix I 1. A target speed for the cycle is also used, for Lv-Bo-Lv that is constantly 85 km/h, the actual speed is however varying with road inclination and therefore requires extra accelerations and decelerations to keep the speed, also see Appendix I 2.

The other duty cycle used is a regional distribution (RDH) cycle which runs on hilly roads, the reason is to illustrate a contrast to Lv-Bo-Lv and find advantages or disadvantages of components on different routes. The vehicles simulated can be found in Appendix J, the different specifications of the vehicle at the duty cycles are also presented. The technical differences between the simulated vehicles try to illustrate the areas where improvements can be achieved now and in a near future. The trucks and their fuel consumption are shown in Table 5.

The largest resistance on a truck is the rolling resistance, by utilizing very low resistance tires, for instance well worn tires, the fuel consumption can be altered very much for both drive cycles. The other factor influencing the rolling resistance is the vehicle mass. For some heavily loaded trucks the weight can not really be decreased, because a decrease in vehicle weight will only result in more freight. For most trucks the weight is not limiting but rather the trailer volume and a decrease of mass of the truck will be a decrease of the total weight. Nevertheless a clear advantage of either lower fuel consumption or more cargo loading can be obtained by the driver. The mass of the vehicle is also a factor influencing more at hilly and distribution like routes where acceleration becomes more important.

The next large influencer is the aerodynamic resistance, which for a truck is less dominant than the rolling resistance. At Lv-Bo-Lv cycle it becomes more important than at RDH but still not in the same magnitude as the rolling resistance. The aerodynamic shape as well as the size, (in this case the frontal area of the trailer), matters in the same extent to lower the fuel consumption. Frontal area is however difficult to decrease because loading volume is affected which usually is: the limiting factor for the transportation. It can also be said here, that a larger difference in fuel consumption would be achieved if the roof deflector were adjusted poorly and the trailer height was modified. But in the simulation the same drag coefficient is assumed. But if relating to the difference between the AA and T052 trailer in the coast down a much larger would be achieved.

Table 5: Vehicles' fuel consumption (l/100 km) from simulations in PERF

Vehicle:	Fuel consumption Lv-Bo-Lv	Fuel consumption RDH
Reference vehicle	30.96	55.24
Decreased rolling resistance, -19.7 %	29.42	54.03
Increased rolling resistance, +17.6 %	32.49	56.44
Decreased weight, -1 ton	30.51	54.10
Increased weight, + 1 ton	31.39	56.36
Larger tires, +6.2 % in Ø	30.83	55.54
Smaller tires, -6.2 % in Ø	31.16	55.47
Decreased drag, -5.7 %	30.48	55.00
Increased drag, +5.7 %	31.42	55.28
Lower trailer, -2.6 % in frontal area	30.75	55.08
500 hp engine	31.07	55.63
540 hp engine	31.20	56.45
420 hp engine	31.08	55.22
380 hp engine	30.98	54.79
Higher ratio rear axle, +8.0 %	31.28	55.64
Higher ratio rear axle, +20.5 %	31.67	55.51
Higher target speed, +5 km/h	31.85	
Lower target speed, -5 km/h	30.15	
2.5 m/s wind speed	32.82	56.25
5.0 m/s wind speed	34.93	57.32

Engines in combination with gearing are also affecting the fuel consumption but in a more complicated relation than the two above. Generally heavier gears are more fuel efficient because lower engine speeds can be utilized at high vehicle speeds. But there is a limit when low engine speeds tends to increase fuel consumption, if the engine speed drops to far away from the maximum engine efficiency the fuel consumption will increase. By decreasing the gear ratios the maximum acceleration and pulling capability decreases which can be other factors influencing the choice of ratio. It should also be mentioned that dependent on the duty cycle and the type of driving different gearing can affect the average operating point of the engine load map, and thereby change the fuel consumption. By downsizing the engines, in general lower fuel consumption is obtained, but this is much dependent on the losses of the vehicle. Lets say that a lot of cargo or an inefficiently equipped tractor is used the engine might have to be strong in order to run at the optimal operating point at normal driving, while a smaller engine which consumes less fuel for a certain engine speed and load needs to run at higher load, in order to produce the same power, and as a consequence differ from the best efficiency region. Therefore it is difficult to generalize between which gearing and/or engine that is the most efficient for the usage and duty cycle.

Lastly some driving conditions are simulated as well, this is done by varying the target speed from 80 km/h up to 90 km/h for the Lv-Bo-Lv duty cycle. Another parameter affecting a specific drive is the wind speed. Some experimentation of up to 5 m/s wind speed is conducted in order to describe how much unchangeable wind conditions can affect the results. The constant head wind of 5 m/s proved to be the largest single fuel consumption increaser, as can be seen from Table 5. But it should be kept in mind that it hardly ever blows with a constant headwind of 5 m/s.

4.3.3 Efficient combination in PERF

Some areas of improvement give even higher gains when combining them with others therefore some simulations of an optimized truck were carried out, Table 6. The most evident conclusion is that the downsized engine will not contribute to any fuel consumption gains on the Lv-Bo-Lv cycle and with a gross weight of 40 ton. On the more energy consuming distribution cycle the less powerful engine performed better, probably because it operated at higher efficiency regions for longer time than the more powerful engine. In practice such low fuel consumption obtained from the optimized vehicles are hard to realize. That is of course as long as the mass is unchanged. Lower mass can decrease the fuel consumption to a very large extent, as seen in Table 5, especially if the duty cycle is more of a distribution kind, with a lot of acceleration. The discussion is however a bit irrelevant because if less cargo is carried the fuel consumption of the specific truck is decreased, but in such case more vehicles might be required to move the freight. Two half full trucks are significantly less effective than one full, but usually it is the volume that limits the amount of freight. A half full truck on the Lv-Bo-Lv cycle would manage 26.61 l/100 km, with otherwise the same specifications as the reference vehicle according to simulations.

Downsized engines are discussed frequently in order to decrease future trucks fuel consumption, from the test the downsizing did not pay off. Partly because the engine did not decrease in size, (the most obvious difference was the increased turbo pressure), but it is also dependent on where at the engine load map the operating point is moved to. From Appendix J it was seen that the average engine torque and power decreased for the smaller engine during the Lv-Bo-Lv cycle, which suggests that the operating point is moving down the load axle, from the high efficiency region and causing higher fuel consumption. The average engine speed would also increase slightly, because the torque decreased more than the power, and higher engine speeds means higher friction losses and increased fuel consumption. It can however be expected that a decrease in fuel consumption would be obtained from the less powerful engine if the vehicle mass was lowered with some tonnage relative to the same modification on a more powerful engine.

A test vehicle that represents the one used at the fuel consumption road test was also simulated, the main difference with this and the reference is the frontal area which has increased to 10.08 m^2. The vehicle weight was also measured to 120 kg more than the reference, the engine was also more powerful, 500 hp instead of 460 hp.

When comparing the fuel consumption with those obtained from road tests one should bare in mind that a slight difference occurs because PERF assumes a stand still start, which is not performed at the road test, (mostly for practical reasons).

Table 6: Vehicles optimized for fuel consumption (l/100km) according to individual component simulations

Vehicle:	**Fuel consumption Lv-Bo-Lv**	**Fuel consumption RDH**
Reference vehicle	30.96	55.24
Volvo FH from test	31.44	55.86
Optimized vehicle 460 hp	28.23	53.00
Optimized vehicle 380 hp	28.25	52.68
Low rolling resistance 460 hp	29.42	54.03
Low rolling resistance 380 hp	29.46	53.58

4.4 VFL fuel consumption simulation

VFL is one of Volvo’s chassis dynamometers that can simulate driving conditions of any duty cycle. The driving load needs to be measured and then transferred into the computer which controls the large rollers that causes the resisting force to the vehicle’s propulsion. The room is also a wind tunnel that creates a head wind representing the vehicle speed. Rather than simulating the aerodynamic resistance the wind only works as a realistic cooling fluid for components and especially the radiators. All driving resistances are summed up as a force in the rollers.

The vehicles tested in VFL are the same as the ones used for the coast down, except for the Volvo, which was with similar specifications but with another wear characteristic.

4.4.1 Issues with VFL fuel consumption measurement

VFL simulations can only be carried out if an input curve of how the forces on the vehicle are varying with speed is inserted. That is the main purpose of doing a coast down test. By using the coast down results all faults that might have occurred during that test is transferred along with its uncertainties to the VFL test. If it is assumed that the coast down test is differing by 10 % to the true coast down curve, (that is approximately a variation of 200 N at low speeds and 300-400 N at top speed), then a fuel consumption difference of approximately 2-3 % is the outcome. Beside this there are other factors affecting the results in VFL:

- The first thing is the difficulty in approximating a flat road with a roller, the tire deformation and contact patch is completely changed. There have been measures taken to compensate for this so probably this issue is small. Then it should also be remembered that a rubber tire does not have the same friction limits on a steel roller, and higher slip values can occur, which has to be compensated for, either by increasing the pull down force, but then less realistic tire deformation is created, so normally a typical loss caused by this phenomena is added to the rolling force.

- The fuel consumption is measured with two means, either by measuring the carbon dioxide content in the exhausts and then do chemical carbon balances in order find the amount of fuel being injected, (the intake air is also needed to be measured). It is based upon the assumption that all Diesel are being burnt. In practice some unburned hydrocarbons are passing the engine to the exhaust. These hydrocarbons are usually fuel that conceal in crevice volumes such as between the piston and cylinder surface, (that is usually a very small amount on Diesel engines), some Diesel fuel are present in the injector sac volume and do not burn due to the rich mixture, the same can occur in some regions of the combustion chamber where the mixture is to rich to burn. The carbon balance is also much dependent on an accurate measurement of the air flow into the engine. Leaks that can occur at every seal are a very problematic matter, which arises at duty cycles with a lot of stop and goes. The reason was the sudden increase in intake pressure caused by the turbo lag when the accelerator pedal is released.

  The other way of measuring fuel consumption is by constantly weigh the amount of fuel supplied to the engine, (any leaks can ruin the results), and the accuracy of the scale is of course of great importance. The fuel that is being returned to the fuel tank is also weighted and subtracted from the fuel feed.

- When performing a coast down test the combined resistances of rolling, aerodynamics and driveline are measured. That correlates to the force the engine is required to overcome in order to propel the vehicle. In VFL that force can not directly be converted to the rollers, partly because the rollers own resistance already mentioned. Due to this the force at the rollers has to be decreased to simulate the reality, but the dragging force on the rollers should not include the driveline resistance either. The driveline is already compensated for between the tire and engine and therefore has to be subtracted from the force curve added to the rollers. That is partly done by performing a coast down inside VFL, and erasing the difference between the two coast downs. That difference is the combination of the losses in the rollers, the driveline and naturally all other factors differing from the real world coast down test and the one in VFL, such as temperature, weather and the absence of a trailer. The difference of temperatures in the driveline between the two tests is also a factor affecting the resistance as was seen in Chapter 4, Section 4.1.4.

- When the total vehicle resistance in VFL is established it needs to match the coast down profile. The parameters describing the profile are inserted to the computers and a coast down in VFL is performed with these forces on the rollers. The final curve is however not usually matching the coast down load curve and therefore the input parameters needs to be changed by iteration. By just changing the parameters in order to find a curve matching the coast down, potential errors can be included and perhaps more importantly: the individual areas of losses can not be noticeable.

- No trailer is used in VFL which means that less work is required from the air pump for the suspensions. This pump consumes around 6.5 kW when used and 1 kW when no pumping effort is done, but without the trailer the pump is used less frequently. So some fuel can be saved by not having the trailer suspension to feed. On the Lv-Bo-Lv drive cycle around 0.4 % of the fuel consumption is consumed by the air compressor.[9]

- Lower fuel consumption is achieved in VFL because no increased rolling resistance due to cornering is assumed, at the same time the power steering pump is not used and therefore a decrease in fuel consumption of 0.15 % is obtained for the Lv-Bo-Lv duty cycle.[10]

- Steering not only influences the rolling resistance but an increase in the drag coefficient, due to a less smooth flow over the trailer gap. The absence of any steering in the road load and the simulations makes the fuel consumption somewhat lower than in practice.

4.4.2 VFL drive cycles

Three different drive cycles are simulated in VFL, partly to match the fuel consumption test cycle but also to get clear correlation for the emissions tests. Lv-Bo-Lv is the fuel consumption test cycle while Via-Bol-Via, (Viared-Bollebygd-Viared) and City Hällered are mainly used for emissions. Lv-Bo-Lv was already described in the PERF chapter and is the cycle of highest interest for the fuel consumption part. Via-Bol-Via test cycle is a new cycle that is mainly intended for emissions, the cycle is made for two laps of running, in order to get higher confidence of the actual emission reading.

In order to construct a new test cycle the road needs to be measured with an accurate GPS where both distance and elevation is continuously logged. Vehicle target speed is also required to be set. This was done by using the speed limit signs and inserting them at the right distance measured. During the cycle three stops were made, all lasted for five minutes in order to simulate how idling emissions affects the results. The cycle is 73 kilometres and the specific road elevation and target speed can be seen in Appendix L.

The third cycle is a buss cycle with 12 stops at Hällered’s country side course, which roughly represents the stopping sequence of a city bus. The cycle is definitely not representing any common truck drive, but it gives high values of emissions and thereby more distinguishable results.

4.4.3 VFL fuel consumptions

In VFL the vehicles needs to be carefully prepared with logging equipment of the vehicle’s ECU parameters. The intake air as well as the exhaust has to be carefully measured and therefore no leakage in any of the sealing’s are allowed. The air and exhaust are feed to and from the engine in large pipes that has to be well sealed, the same applies for the fuel, that is distributed from an external fuel tank in order to measure the amounts. The vehicle is held in place on the rollers by a steel construction that forces the rear wheels down with the certain drive axle load, previously measured.

The front wheels are also connected to a set of rollers. These rollers does not create any resistance, but are mainly intended to rotate the front wheels with the same speed as the rear in order to have a correctly measured vehicle speed that is needed for some of the electrical control systems in the vehicle.

The vehicles were also equipped with PEMS (Portable Emissions Measurement System) equipment, to measure emissions in the proposed legislative way. The PEMS is basically an exhaust analyzer that correlates the exhaust with a reference tube filled with the pure emission composition. More extensive information about the PEMS equipment is not fitting inside the boundaries of this text.

The fuel consumption is measured in two different ways as described earlier, the results of the two methods are shown in Table 7, Table 9 and Table 10 for the different duty cycles respectively. Generally the weighting is the more accurate one, as long as no leaks are present, the fuel parameters of the Diesel fuel is shown in Appendix M 1. The tests were conducted at two different operating temperatures, 10 and 25 ºC in order to get a feeling in how the emissions were affected by lower temperatures. But even the fuel consumption is temperature dependent.

The approximated fuel consumption calculated from the carbon balance were based on the "raw after" data which is the exhaust gases after the after treatment devices.

Table 7: VFL fuel consumption results for the Lv-Bo-Lv duty cycle

Lv-Bo-Lv			
Vehicle:	**Fuel consumption, fuel input weighting, [l/100 km]**[1]	**Fuel consumption, carbon balance, [l/100 km]**[1]	**Average duty cycle speed**
Scania R480, 10 ºC	40.85	39.05	83.5 km/h
Scania R480, 25 ºC	40.84	39.67	83.6 km/h
Mercedes Actros, 10 ºC	38.03	36.65	84.1 km/h
Mercedes Actros, 25 ºC	37.08	36.11	82.6 km/h
Volvo FH500, 10 ºC	35.21	(29.66)	83.6 km/h
Volvo FH500, 25 ºC	35.36	35.15	83.7 km/h
Renault Premium (DAF load curve), 10 ºC	36.02	35.15	83.8 km/h
Renault Premium (DAF load curve), 25 ºC	36.22	35.38	83.7 km/h
Renault Premium (extrapolated load curve), 10 ºC	36.34	35.84	83.9 km/h
Renault Premium (extrapolated load curve), 25 ºC	-	-	-
DAF XF105, 10 ºC	37.85	36.78	83.9 km/h
DAF XF105, 25 ºC	37.79	37.02	83.4 km/h

[1] Based on fuel at 25 ºC

Generally only small variations in fuel consumption were found between the two surrounding temperatures, strangely enough the Mercedes is showing large reductions in fuel consumption at 25 ºC. Most others tended to have an increase, part of the decrease for the Mercedes can be explained by the fairly large average speed difference between the two tests. The variation in fuel consumption is however as much as 2.5 %, which suggests in which region VFL's reproducibility lies. The main influencer of the fuel consumption according to the Mercedes example is the human driver or the control system of the cruise control, (if used), which probably are the weakest parts of the complete simulation.

The exhaust gas balance based fuel consumption followed the weighted one with approximately up to 3 % lower readings, the only exception was the Volvo FH at 10 ºC where some malfunctioning must have occurred.

As can be seen in Appendix I 2 and from the average speed data in Table 7, the driving speed for the Lv-Bo-Lv drive cycle is almost constantly 85 km/h. To correlate if the VFL simulation is having a good accuracy the fuel consumption difference can be compared with the drive load difference at 85 km/h at the coast down test. The only varying components are the engine and transmission which can give an indication of which manufacturer produces the most efficient engine, (at least at highway driving).

The difference between the fuel consumption results and the coast down is found in Table 8. If VFL itself has a high accuracy the results are indicating that the Scania has a less efficient engine than the Volvo while the opposite implies for the Mercedes. For the Renault using the DAF load curve the results are not really relevant because vehicle resistance and the powertrain are mixed. The extrapolated Renault load curve is not realistic and large variations of the force at 85 km/h can probably be found, despite that very competitive fuel consumption was surprisingly achieved. The DAF tended to have a less competitive engine compared to the Volvo but also compared to the other rivals.

Table 8: Differences between VFL fuel consumption measurement and Coast down force

Vehicle:	**Coast down force at 85 km/h [N]**	**VFL fuel consumption Lv-Bo-Lv 10 ºC [l/100 km][1]**	**Percentage difference to reference from coast down**	**Percentage difference to reference from VFL fuel consumption**
Scania R480	5189	40.85	+14.9 %	+16.0 %
Mercedes Actros	4943	38.03	+9.4 %	+8.0 %
Volvo FH500 (Reference)	4517	35.21	-	-
Renault Premium (DAF force)	4699	36.02	+4.0 %	+2.3 %
Renault Premium (Extrapolated)	5350	36.34	+18.4 %	+3.2 %
DAF XF105	4699	37.85	+4.0 %	+7.5 %

[1] Based on fuel at 25 ºC

At the Via-Bol-Via duty cycle, Table 9, the average driving speed is much decreased which means that rolling resistance becomes more significant. This would generally favour the Renault and the DAF, but as can be seen even the Volvo proved to have good fuel consumption. Once again this is strengthening the theory that the PACCAR MX engine is less efficient than the rivals.

Table 9: VFL fuel consumption results for the Via-Bol-Via duty cycle

Via-Bol-Via			
Vehicle:	**Fuel consumption, fuel input weighting, [l/100 km][1]**	**Fuel consumption, carbon balance, [l/100 km][1]**	**Average duty cycle speed**
Scania R480, 10 ºC	52.81	49.94	40.7 km/h
Scania R480, 25 ºC	52.25	50.24	41.2 km/h
Mercedes Actros, 10 ºC	49.42	47.44	41.2 km/h
Mercedes Actros, 25 ºC	50.03	48.16	41.4 km/h
Volvo FH500, 10 ºC	47.08	45.77	40.9 km/h
Volvo FH500, 25 ºC	48.75	48.03	41.0 km/h
Renault Premium (DAF load curve), 10 ºC	-	-	-
Renault Premium (DAF load curve), 25 ºC	45.82	43.73	41.1 km/h
Renault Premium (extrapolated load curve), 10 ºC	45.27	43.34	41.0 km/h
Renault Premium (extrapolated load curve), 25 ºC	-	-	-
DAF XF105, 10 ºC	48.38	44.79	41.0 km/h
DAF XF105, 25 ºC	48.60	45.37	40.8 km/h

[1] Based on fuel at 25 ºC

The City Hällered bus cycle is driven at even lower average speeds and consequently an even larger part of the resistances are rolling resistances. At such low speeds and frequent stop and go’s, large losses due to acceleration and idling occurs, therefore a fuel efficient engine is vital. According to the test results the Scania is more fuel efficient than the Mercedes which probably can be described by the long periods of idling, but also because a large uncertainty can be found in the results, the carbon balance method for instance describes the opposite.

Large differences between the two temperatures can also be found, few conclusions can however be drawn because the fuel consumption are higher at 25 ºC for some trucks and lower for others.

The perhaps most evident conclusion from the bus cycle is that the fuel consumption is doubled in comparison to the distribution cycle Via-Bol-Via. Hybridization can therefore be an efficient improvement for the propulsion system at vehicles operating in city traffic, in general buses.

The two measurement methods did also show a large difference in values, the reason for this is assumed to be the large variation in intake pressure due to many hard accelerations and engine braking which can cause high pressure gradients and leakage at the seals. The extra air that either is drawn in or pushed out destroys the carbon balance, due to the inaccurate air mass reading. The effect was most evident for the competitor vehicles, suggesting that the connection between the intake system and the feed pipe was less suitable, in comparison to the Volvo.

Table 10: VFL fuel consumption results for the City Hällered duty cycle

City Hällered			
Vehicle:	**Fuel consumption, fuel input weighting, [l/100 km][1]**	**Fuel consumption, carbon balance, [l/100 km][1]**	**Average duty cycle speed**
Scania R480, 10 ºC	99.29	89.58	24.1 km/h
Scania R480, 25 ºC	-	-	-
Mercedes Actros, 10 ºC	99.91	89.29	23.9 km/h
Mercedes Actros, 25 ºC	102.10	91.76	23.6 km/h
Volvo FH500, 10 ºC	89.95	86.55	23.1 km/h
Volvo FH500, 25 ºC	89.53	88.43	22.7 km/h
Renault Premium (DAF load curve), 10 ºC	92.98	85.39	23.3 km/h
Renault Premium (DAF load curve), 25 ºC	88.32	81.36	23.8 km/h
Renault Premium (extrapolated load curve), 10 ºC	91.82	84.42	23.2 km/h
Renault Premium (extrapolated load curve), 25 ºC	-	-	-
DAF XF105, 10 ºC	91.01	75.81	23.5 km/h
DAF XF105, 25 ºC	91.71	76.58	23.9 km/h

[1] Based on fuel at 25 ºC

4.5 Road fuel consumption measurement

The classic way of measuring fuel consumption is by driving the vehicle at the actual duty cycle on road. The assumptions are minimized and as a result realistic values are gained, the repeatability is however low due to the variation between drivers and weather.

The measurement is conducted with help of a portable flow meter that accurately measures the fuel by measuring the amount of pulses of a constant volume that is feed by a cog gear. The amount of fuel is manually inserted to a file when a specific interval of the duty cycle is finished. The truck is also driven along with a reference truck that is equipped with the same measurement device, the same reference is used for all the tested trucks.

The road test is performed at the Borås-Landvetter duty cycle, note that for the road test there is a higher convenience of driving in the opposite direction Bo-Lv-Bo. The cycle is still the same and no difference in fuel consumption should be witnessed.

After the test the two trucks are normalized in order get comparable fuel consumptions. Factors such as weight frontal area and average speed are normalizing the fuel consumption with approximate correction factors, see Section 4.5.2.

4.5.1 Issues with road testing of fuel consumption

Even if road tests gives accurate results there is plenty of issues that can affect the absolute value between tests and days. The reference truck is however a good insurance and can make errors or vehicle differences distinguishable.

- At the road test two different fuel reference trailers were used and shifted between the tested vehicle and the reference vehicle, by taking the average of the different runs a good correlation is achieved. The trailers were however not the same as the one used at the coast down and which later on was used as base in VFL. The trailer height was almost the same, (the AA trailer was five centimetres higher), but other factors such as tire type and size, as well as aerodynamic components such as trailer side skirts were different.

- Weather can affect the results significantly, especially rain (such weathers are however avoided) but also temperatures and winds.

- One of the most sensitive areas of road testing is the drivers. At this test two different drivers were used which naturally worsen things because it is difficult for them to do the same kind of drive when it comes to accelerations and decelerations.

- All vehicle specific components that are found on some of the trucks but not on others are of course an error that is difficult to omit. This problem is on the other hand not specific for these kinds of tests. It also depends on what kind of comparison that is wanted: between competitors or between test methods. As long as the vehicles are comparable on equal grounds, their specific components are of less importance.

- The measurement equipment is an area of concern because a very accurate device is needed not get small errors of every sample that is measured, in the end such inaccuracies can become notable. The accuracy of the fuel flow measurement device should be within 0.5 %, according to its specification.

- The tires used on the tractors for the fuel consumption tests were the same for all runs except for the Mercedes Actros which required smaller diameter wheels. The tire type and pattern were different and potentially affecting the Mercedes negatively

- If a comparison between the VFL results and the road test results is to be carried out on equal ground the trailers and tires should be similar at both occasions. At the road test the tractor tires were shifted to similar ones for all trucks while the coast down utilized the tires fitted to the tractor initially. The trailers were also different between the two tests where the coast down trailer T052 was equipped with side skirts and well worn tires of other dimensions.

4.5.2 Road test corrections

When conducting the road fuel consumption testing a reference truck is used to correlate if external influencing factors are affecting the test. The reference is serviced frequently and thereby repeatable results should be achieved unless weather or driving behaviour has changed a lot.

The fuel consumption result is also manually transformed for the reference truck in case there are vehicular variations between the two trucks. The main disadvantage of correcting the reference truck's fuel consumption is that it becomes non comparable to the other tests where the reference vehicle is corrected for another vehicle configuration. But in case the difference between the reference and the tested vehicle is wanted the method is the most accurate one. It can still be argued whether it is fair or not to compensate for weight differences, if a truck is managing to carry the same load and still be lighter then a competitor, should the fuel consumption really needed to be increased for compensation?

Five parameters are presently used for normalization: difference in vehicle average speed, difference in aerodynamic and rolling resistance coefficients (in these tests the coefficients were not measured and therefore no corrections were done), the vehicle frontal area and the vehicle weight. The first parameter is the only one that is not vehicle specific and the most important one to normalize, the correction of the speed is based on the measured average speed between the start mark and the Ellos sign on the Bo-Lv-Bo cycle. This part of the duty cycle is flat and can by that distinguish differences in the control systems of the cruise control. The cruise control is basically used for the complete cycle, and a difference at the flat part will contribute to a similar error throughout the cycle.[11]

The normalization factors are established from PERF simulations of the specific duty cycle segments, they should be seen more like guidelines rather than absolute values, the factors can be seen in Table 11. The base values are the ones obtained from the tested truck, while the correction is done on the reference truck. The absolute value gained from the reference truck is of less interest while the difference between the two trucks is important.

Table 11: Correction factors of fuel consumption from PERF of reference truck at road testing

	Correction factors
Average speed (*v*) [km/h]	$0.1533 \left[\dfrac{l/100km}{km/h} \right]$
Aerodynamic coefficient (C_D) [-]	$0.1633 \left[\dfrac{l/100km}{0.01\Delta C_D} \right]$
Frontal area (*A*) [m^2]	$0.9 \left[\dfrac{l/100km}{m^2} \right]$
Rolling resistance coefficient (f_r) [-]	$0.78 \left[\dfrac{l/100km}{+10\% f_r} \right]$
Vehicle weight (*m*) [ton]	$0.42667 \left[\dfrac{l/100km}{1000kg} \right]$

4.5.3 Road test results

Prior to the runs the vehicles are well documented to be able to distinguish differences between future test conducted with similar trucks or equipment, the tested vehicles and the reference are all specified in Appendix N. The Renault Premium was not tested because the cabin shape differed from the rest of the trucks to a large extent.

The fuel consumptions from the road tests are found in Table 12, the most relevant data is the difference between the tested vehicle and the reference which give a clear indication on how well the vehicle performed.

The fuel consumption is based on four runs on the cycle with two different trailers. There were no clear indication of which trailer that performed better than the others but from the individual runs it was seen that a variation of 1 l/100 km was not uncommon. For the reference truck the fuel consumption varied as much as 2.5 l/100 km between different days, which gives an indication of how much the weather can affect. It should still be kept in mind that rain was avoided at all occasions and the only parameters affecting the exterior influences were the road conditions, temperatures and winds.

Table 12: Road test results

Borås-Landvetter-Borås		Tested vehicle	Reference Volvo	Corrected reference Volvo	Diff. test/ref vehicle
Scania test	Fuel [l/100 km]	33.62	31.46	31.47	**6.82 %**
	Speed [km/h]	85.63	85.65	85.22	0.48 %
Mercedes test	Fuel [l/100 km]	32.33	29.86	29.42	**9.87 %**
	Speed [km/h]	83.99	84.54	82.95	1.26 %
Volvo test	Fuel [l/100 km]	31.41	31.81	32.23	**-2.54 %**
	Speed [km/h]	86.11	85.68	87.91	-2.05 %
DAF test	Fuel [l/100 km]	34.79	32.40	32.58	**6.77 %**
	Speed [km/h]	85.43	85.39	86.49	-1.23 %

4.6 Other fuel consumption test methods

Another method not used for fuel consumption evaluation at this occasion is an engine bench, where a complete engine with gearbox is driven and loaded with an electrical generator that requires the amount of power and speed for every drive combination at a test cycle.[12] The vehicle load due to rolling resistance, aerodynamic resistance and driveline losses obtained from coast down tests are the input to the electrical machine therefore an accurate load with few external influencing factors are attained. The results are still much dependent on a coast down measurements with high repeatability, but other than that few other uncertainties.

5 Results

Fuel consumption measurements have a lot of uncertainties and consequently variations in absolute values are achieved. In this project the absolute fuel consumption values are not so important but rather the analysis of why certain methods performed better than others. An example could be that chassis dynamometer (VFL) results gave larger fuel consumption than the equivalent road tests for all the trucks.

5.1 Comparison of test results

The Bo-Lv-Bo test cycle was used as an evaluation cycle in the computer simulations (PERF), chassis dynamometer (VFL) and for the road tests. The lack of engine information for the competitor trucks made it impossible to accurately simulate the vehicles in PERF. The fuel consumptions based on the different test methods are presented in Table 13. The PERF simulation seemed to match the road test, but what probably is missing is the weather influence that should have affected the computer simualtions negatively, such as side winds, wet road surface, uneven roads and low temperatures. The conclusion is that the assumed rolling resistance for the PERF simulation was set to high. If that is true the rolling resistance achieved from the coast down that later on is used in VFL is way to high for the Scania and the Mercedes. The Volvo FH was approximated to have the same rolling resistance coefficient in PERF as for the coast down and VFL. This implies that the difference in fuel consumption between computer simulations and chassis dynamometer is only due to wind, errors in VFL measurements and/or calibration or higher than expected aerodynamic resistance in VFL, or less efficient way of following the duty cycles target speed.

Table 13: Fuel consumption results in [l/100 km] for the different vehicles and test methods at Lv-Bo-Lv duty cycle

Vehicle:	PERF simulation	VFL simulation	Road test
Scania R480	-	40.85	33.62
Mercedes Actros 1848	-	38.03	32.33
Volvo FH500	31.44	35.21	31.41
DAF XF105	-	37.85	34.79

Road testing and VFL testing also presented a large difference for all trucks. The largest deviations were found at the trucks where the coast down results indicated a higher than expected rolling resistance. Only the DAF was driven at dry conditions at Hällered and it obtained a realistic rolling resistance from that test, still a large difference between VFL and road tests were found. The coast down of the DAF which the VFL results are based upon were conducted at close to ideal conditions with low wind speeds and head wind in one direction and tail wind in the other. The weather at the road test can not be assumed to be much better, which concludes that VFL results generally give higher fuel consumption than road tests.

In the PERF chapter, Table 5 differences between 0.006 and 0.0041 were simulated to be 3 l/100 km at the Lv-Bo-Lv drive cycle. A rolling resistance coefficient of 0.0041 is far lower than expected for a truck but even with this kind of decrease the Scania's VFL fuel consumption would lie around 38 l/100 km, according to PERF simulations Table 5. Even with this correction the difference between the methods is above 4 l/100 km. From the weather information in Appendix A it is seen that a side wind of around 3-4 m/s was present at the coast down of the Scania, if it is assumed that this can be represented by a constant head wind of 2.5 m/s, which probably is an exaggeration, the fuel consumption would decrease by 2 l/100 km further on the Lv-Bo-Lv drive cycle. After these corrections a difference between the test methods similar to that of the DAF is obtained.

According to the examples evaluated previously the VFL results tends to overestimate the fuel consumption, alternatively the road test is underestimating them. The later alternative is however less probable because the only parts which can underestimate the fuel consumption in the measurement are the fuel or the speed measuring devices. Vehicle failures can of course occur but such issues are usually affecting the results more evidently. Then it should also be mentioned that the road test correlated well with the PERF simulation. The PERF simulation should however be lower because it is ideal, but it was probably affected by a too high estimation of the rolling resistance. (Estimation based on the coast down result, where slightly wet road was used which equalled Energy tires in rolling resistance in PERF). For the road tests the rolling resistance were minimized by using worn Michelin energy tires, which of course decreased the fuel consumption, but hardly below 0.0041 which was simulated in the PERF section as a test. But for VFL fuel consumption to decrease to the same magnitude as for the road tests even lower rolling resistance is needed. Another factor could be the higher than allowed target speed. At the road tests similar speeds were used, but the measured value could be measured inaccurately.

The weighting of fuel in VFL is at least as accurate as the measurement at the road test and therefore no big difference should be found. The difference in fuel consumption should then lie with the road load adaption or calibration. Road load curves are not inserted directly to the rollers, because the loss caused by the friction of the rollers' bearings, the inertia of the rollers as well as the powertrain must be excluded from the road load at the wheels. Therefore a similar coast down is performed in VFL, the resulting graph is adjusted by changing load parameters in VFL in order to get the shape of the actual coast down curve. The procedure is a trial and error way of doing things and large uncertainties occurs, especially because the road load in VFL is to represent a road load performed in another environment.

One major difference between the tests were the trailers, both trailers were used for the coast down with the Volvo FH and a significant reduction in rolling resistance was seen for the AA trailer but also a higher force at high speeds. It can be assumed that the increased resistance at high speeds was due to aerodynamic inefficiencies, in general higher trailer and perhaps less efficient side skirts. On a drive cycle like Bo-Lv-Bo such results would lead to an increase in fuel consumption for the road tests. But presumably the roof deflectors were adjusted to a more efficient position than was done at the coast down with the AA-trailer.

Table 14: Fuel consumption difference in percentage between road test and VFL simulations at Lv-Bo-Lv duty cycle

Vehicle:	Percentage difference between road test and VFL fuel consumption results, (road test used as reference)
Scania R480	+21.5 %
Mercedes Actros 1848	+17.6 %
Volvo FH500	+12.1 %
DAF XF105	+8.8 %

From Table 14 the VFL simulations percentage difference in fuel consumption relative to the road test can be seen. As was discussed before, the fuel consumption was generally higher in VFL, but no constant relation was found. The percentage differences reminded more of the coast down force difference at 85 km/h, see Table 8. The DAF is the exception, but it can be assumed that the DAF actually had better weather at the coast down test compared to the road test, while the opposite implied for the others. The temperatures at the DAF's tests were about the same, and no humidity was found on the track at the coast down and at low wind speeds. For the road test no such weather information was logged and thereby it is difficult to compare.

It is therefore assumed that the largest difference between the two test methods was due to weather, the remaining difference is probably because of the factors previously discussed in this chapter.

6 Conclusion

Testing of fuel consumption is a very sensitive area where all parts of the vehicle have to collaborate in order to achieve accurate and reliable results. In order to improve the relevance of future fuel consumption testing results, the different trucks should be driven at similar conditions and preferably at the same day in order to neutralize the weather dependence.

6.1 Major fuel consumption influencers

Prior to any kind of fuel consumption testing, several areas related to the truck but also to the conditions at the tests need to be examined. The most important ones are discussed here in order to get a feeling of how large variations that can be expected.

- Winter climates are one of the most significant factors. According to a survey of a customer truck fleet the fuel consumption varied as Figure 25 is showing through the different seasons. A fuel consumption variation of 5 l/100 km for their average drives between the summer and winter. It should be mentioned that this fleet changed to winter tires at some point at the autumn, these winter tires were probably equivalent to the ones mounted at the Scania for the coast down test. The main influencing factor of fuel consumption is however not the tires, but most certainly the road conditions and the air density that can change the aerodynamic resistance with up to 7 % between summer and winter. Rainy weathers probably affect the aerodynamic resistance even further.

Figure 25: Fuel consumption variation dependent on season of a customer fleet[13]

- Wind is probably the largest influencer on the aerodynamic resistance, (as long as the trucks are equipped with comparable aerodynamic devices). Normally windy conditions are avoided, but even small winds such as 2.5 m/s can increase the fuel consumption between 3-6 %, if it is a head wind.

- The most significant aerodynamic features should be the same between trucks to even hope for comparable results. Such devices are mainly the roof and side deflectors which are critical in order to decrease the aerodynamic drag coefficient. From VFL it was seen that the Renault Premium still performed well in fuel consumption, but it should be kept in mind that non-realistic force curves were used, and that the rolling resistance of the Renault was even better than the rest of the rivals, (due to dry roads at the coast down test).

- Tires can also vary the rolling resistance largely. Partly because of different compounds, patterns, dimensions but also because of the wear. Therefore future testing will require standard sets of tires to be used for all vehicles.

- It is important to run-in vehicles before the testing, much increased losses in bearings and joints can be observed with new vehicles especially in the engine. 30´000 km should be sufficient to bed in all the friction surfaces.

- Even if the bearings are well in-bedded they can still contribute to higher frictional losses by having inadequate pre-heating. The vehicle needs to be driven around for at least an hour and with a loaded trailer in order to reach normal operating temperatures.

- Trailers used for the different tests should be similar and especially specified with the same components; otherwise some vehicles might show favourable results for one of the trailers and worse for the other due to the adjustments of aerodynamic devices. This uncertainty makes it difficult to draw any conclusions from the tests. Between VFL and road tests this included various tires, trailer height and trailer side skirts.

- The probably largest single influencer of the fuel consumption is the driver. Large variations due to different drivers, varying driving conditions and trucks can be obtained. From Figure 25 it can also be seen how a trucking company has been able to decrease the fuel consumption by 2 l/100km during 4 years of time by better education of the drivers, naturally the fleet was updated continuously during that time. Therefore it is difficult to distinguish what matters that really decreased the fuel consumption.

6.2 Fuel consumption improvements conclusion

Potential areas of improvement are illuminated from the results of the different tests. To follow up the testing some advices on how to improve fuel consumption are established here.

In the powertrain coast down part it was found that splash losses in the final drive were the major contributor to why the resistance of the powertrain increased significantly at high speeds. One way of avoiding this loss is to add a simple oil pump in the final drive to lubricate the parts without the need of hitting the oil with the crownwheel. In general making the final drive a dry sump, the lubrication of the gear surface and the bearings are created by nozzles and drilled holes in the shafts feed by the oil pump.

The constant term of the resistances in the powertrain is caused by high loads on bearings and gearwheels. To decrease the bearing loads the pre-tensioned conical roller bearings in the gearbox and partly in the rear axle can be changed to axial and radial bearing couples. The normal forces would decrease drastically. Another way of avoiding the conical bearings is to use spur gears in the gearbox, because then all axial loads can be avoided. The noise level will however increase slightly but for a truck that should probably not be an issue because the noise will probably be neutralized by something else, louder. Frictional losses can also be decreased by changing the hypoid final drive to normal bevel gear. The contact force would decrease and by that lower viscosity lubrication oil can be used.

Setting up the wheels correctly, especially toe-in seems to be an important area, even if it is hard to distinguish if any losses are initiated from the suspensions specifically. Large losses can also occur if the vehicle needs to force not aligned axles in the driving direction. An increase in tire wear can also be seen from poorly aligned axles. One way of solving this problem is by introducing self-aligning axles that are continuously changing the toe angle with speeds. Normally the wheels are put into a position where some toe-in is used in order to have a more stable steering, which also creates self centralization of the wheels. To lower the dragging losses of the front wheels the toe angle should be close to neutral, (wheels point straight ahead). The concept could be using electrical motors rotating the steering rods to lengthen or shorten the rods and by that always insure that the wheels are rotated as easily as possible. The only risk is that the vehicle becomes more sensitive to road unevenness.

Aerodynamics is perhaps the easiest area to suggest improvements for, and obviously there are a lot of devices decreasing the drag coefficient. All parts allowing a smooth transition between the tractor and trailer are the most important. The shielding of the gap must however avoid any contact by the trailer, which can occur at both cornering and dips where the upper parts of the trailer and cabin closes in on each other. The gap is usually sufficient enough to avoid any interactions with the side flow nowadays, but the main issue occurs when side winds are present which can cause flows through the gap.

Another very important preparation is the position of the roof deflector. The AA trailer used at roads was around five centimetres higher than the T052 and that seemed to influence the aerodynamic resistance with 400 N at 85 km/h, see Figure 17. In order to avoid such errors in the future, both for testing but most importantly for truckers, the deflector should be self adjustable by using pressure transducer on the trailer front that can distinguish if the flow is stagnating at the front of the trailer surface, or if a smooth flow over the roof deflector is occurring.

6.3 The Volvo FH

From the testing it was observed that the Volvo FH performed better than its main rivals, it should be kept in mind that a lot of exterior factors influenced the results. Factors that should be favourable for the Volvo are a better serviced truck and more experiences in the driving behaviour to decrease fuel consumption. It can also be suspected that the Volvo is more suitable for the cold and unreliable weather present at most of the tests here in Sweden and more adapted to the Lv-Bo-Lv duty cycle.

6.4 Future testing

The large variation due to both truck and non-truck related issues has lead to the conclusion that an absolute figure will be nearly impossible to obtain before the new component is being tested in a large scale. Improvements can however be tested with good indications of the results but then focusing on the relative difference to the reference. This is most accurately achieved in the chassis dynamometer where good repeatability is utilized, but as long as a new load curve is required the actual repeatability becomes worse.

If the modifications of the truck is made on the engine the previous coast down can easily be used, and the gains would immediately be noticed as a relative improvement in the chassis dynamometer. But if any other truck related modifications are made, a new coast down is required to be performed. In order to get an accurate coast down the weather situations should be similar between the tests, but so should the truck as well, (except for the new improvement). Preferably there should be a specific truck available that is only used for coast down, and all modifications should be fitted to that truck. Parameters that should be kept constant at coast down tests in order to get good repeatability are found in Chapter 4, section 4.2.2.

Road tests are heavily affected by weather but also by the driver and the tested truck. By performing the test together with a reference vehicle the accuracy is however largely improved. The relative relation between the reference truck and the tested truck is more important than the absolute value, but even if some issues can be neglected with a reference truck there are some that still remain present. For instance if one of the vehicles are more sensitive to temperatures due to higher amount of oil in the final drive that vehicle will perform relatively worse at colder temperatures than at warm.

The absolutely most accurate way of measuring fuel consumption is by looking at large fleets during a longer time period. Such analysis is however difficult to perform for new parts, because they should not be implemented before the testing. The fuel consumption obtained is however difficult to relate to other fuel consumption measurements because the vehicle weight is heavily varying.

7 References

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[2] Hucho, Wolf-Heinrich (1998). *Aerodynamics of road vehicles – fourth edition*. Page 430.

[3] Rudling, Lars (2009). *Coast down and steady state fuel consumption tests at Råda Proving Ground (RPG) with complete vehicle FH-485, trailer T052. (40 tons).*

[4] Gary Gustafsson Volvo PV, e-mail 100215.

[5] Hucho, Wolf-Heinrich (1998). *Aerodynamics of road vehicles – fourth edition*. Page 435.

[6] Longueville, Marine (2009). *Stage de fin d'études* - *Ingénieur essais* - *Décélérations fractionnées*. Page 30.

[7] Hucho, Wolf-Heinrich (1998). *Aerodynamics of road vehicles – fourth edition*. Page 422.

[8] No author given. PERF (Performance calcualtions) Volvo document.

[9] Johansson, Therese Volvo powertrain & Amar, Pascal Volvo technology (2007). *Power consumption measurements on auxiliaries on a FH13 long haul and impact on fuel consumption*. Page 13.

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[11] Inge Qvarford Complete vehicle Volvo 3P, interview 100503.

[12] Noda, Akira, Tsukamoto, Yujiro and Sato, Tatsuji (2003). *Evaluation method for HDV fuel economy performance with PC simulation and mapping procedure*. Page 3.

[13] Börjesson, Michael (2009). *Follow up of fuel consumption in customers fleets. Comparisons are made between different models of Volvo trucks and some of their competitors. The conclusion is that Volvo FH is competitive in the compared fleets*. Page 16.

Appendix A (Weather information at coast down test)

Appendix A 1: Scania coast down

Date and time	Humi-dity (%)	Air temp (ºC)	Road temp (ºC)	Air pressure (mbar)	Wind direction (º)	Wind speed (m/s)	Wind speed max (m/s)
Feb 5 2010 11:00AM	94	-5.8	-6.2	994	124	3.1	8.9
Feb 5 2010 11:30AM	95	-4.9	-5.6	994	145	2.9	8.3
Feb 5 2010 12:00PM	94.7	-4.1	-5.1	994	133	3.3	9.1
Feb 5 2010 12:30PM	94.6	-3.7	-4.8	994	137	3.3	7.4
Feb 5 2010 1:00PM	94.5	-3.1	-4.3	993	119	4	6.5
Feb 5 2010 1:30PM	94	-2.6	-4.1	993	120	2.3	8.5
Feb 5 2010 2:00PM	93.8	-2.1	-3.8	994	118	2	6.1

Appendix A 2: Mercedes coast down

Feb 8 2010 10:00AM	96.3	-5.3	-5.1	990	229	0.8	2
Feb 8 2010 10:30AM	96.5	-4.6	-4.6	990	260	1.5	2.2
Feb 8 2010 11:00AM	96.6	-4.6	-4.3	990	291	2.1	3.5
Feb 8 2010 11:30AM	96.7	-4.3	-4	990	273	1.6	3.5
Feb 8 2010 12:00PM	97.1	-3.8	-3.6	989	231	0.2	4.1
Feb 8 2010 12:30PM	96.8	-3.7	-3.2	989	295	2.4	3.9
Feb 8 2010 1:00PM	96.9	-3.4	-3	989	293	1.9	4

Appendix A 3: Volvo coast down

Feb 12 2010 9:30AM	96.1	-6	-6.1	996	5	3.5	6
Feb 12 2010 10:00AM	96.2	-5.7	-5.9	997	13	2.3	5.6
Feb 12 2010 10:30AM	96.2	-5.6	-5.6	997	350	2	6.2
Feb 12 2010 11:00AM	96.3	-5.3	-5.3	997	10	3.2	7.1
Feb 12 2010 11:30AM	96.4	-5.1	-5	997	13	2.3	8.9
Feb 12 2010 12:00PM	96.6	-4.9	-4.8	997	18	3.1	8.2
Feb 12 2010 12:30PM	96.6	-4.5	-4.5	997	354	2.9	8.7
Feb 12 2010 1:00PM	96.5	-4.3	-4.3	997	351	3.1	6.2

Appendix A 4: Renault coast down

Feb 24 2010 11:30AM	96	-4.6	-4.3	981	144	0.9	2.2
Feb 24 2010 12:00PM	95.7	-4.7	-3.8	981	199	2.1	3
Feb 24 2010 12:30PM	96.2	-3.4	-2.9	981	131	1.4	2.5
Feb 24 2010 1:00PM	95.6	-3.2	-2	981	134	1.4	3

Appendix A 5: DAF coast down

Mar 3 2010 10:30AM	67.4	-2.9	-6.3	987	292	1.8	4.4
Mar 3 2010 11:00AM	57.3	-2.1	-3.1	987	11	2	5
Mar 3 2010 11:30AM	55.9	-2.1	-2.1	988	333	3	7.6
Mar 3 2010 12:00PM	54.2	-1.6	-2.1	988	342	3.3	6.8

Appendix B (Weight distribution at coast down test)

Appendix B 1: Weight distribution of vehicles

Vehicle:	Front axle weight (kg)		Rear axle weight (kg)		Trailer axle weight (kg)		**Total weight (kg)***
	L	**R**	**L**	**R**	**L**	**R**	
Scania R-series	4070	3715	5735	5300	11035	10735	40590
Mercedes Actros	3715	3630	5335	4645	11600	11750	40675
Volvo FH	-	-	-	-	-	-	39500
Renault Premium	3225	3115	5125	4840	11775	11350	39430
DAF XF	7500		10480		22760		40740

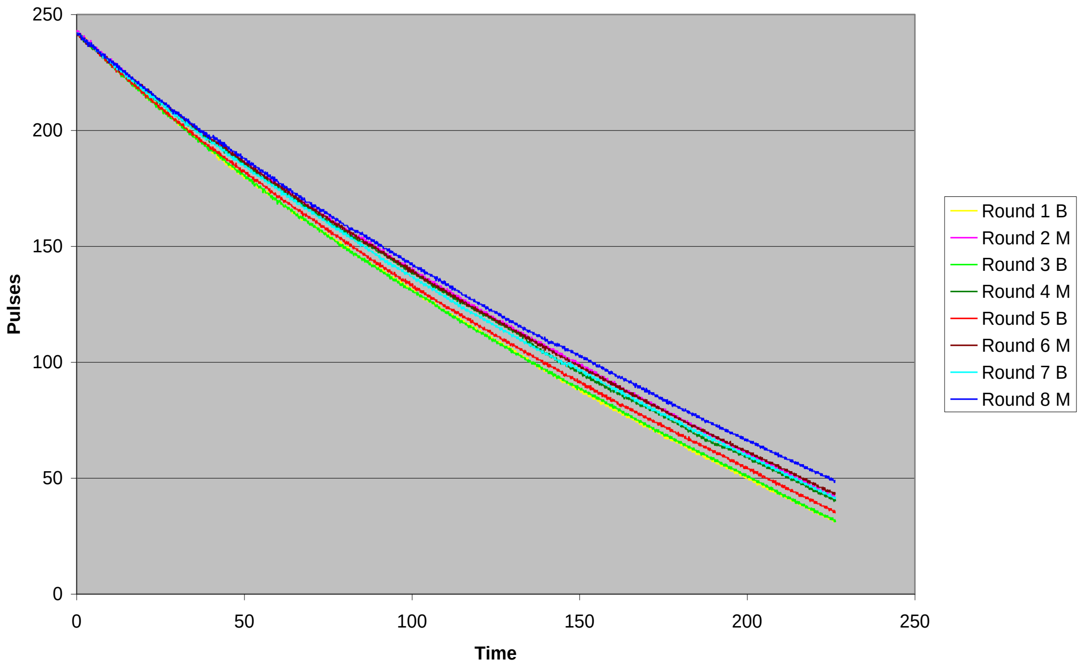
* 200 kg of drivers are to be added to total weight

Appendix C (Matlab-code for air density calculation)

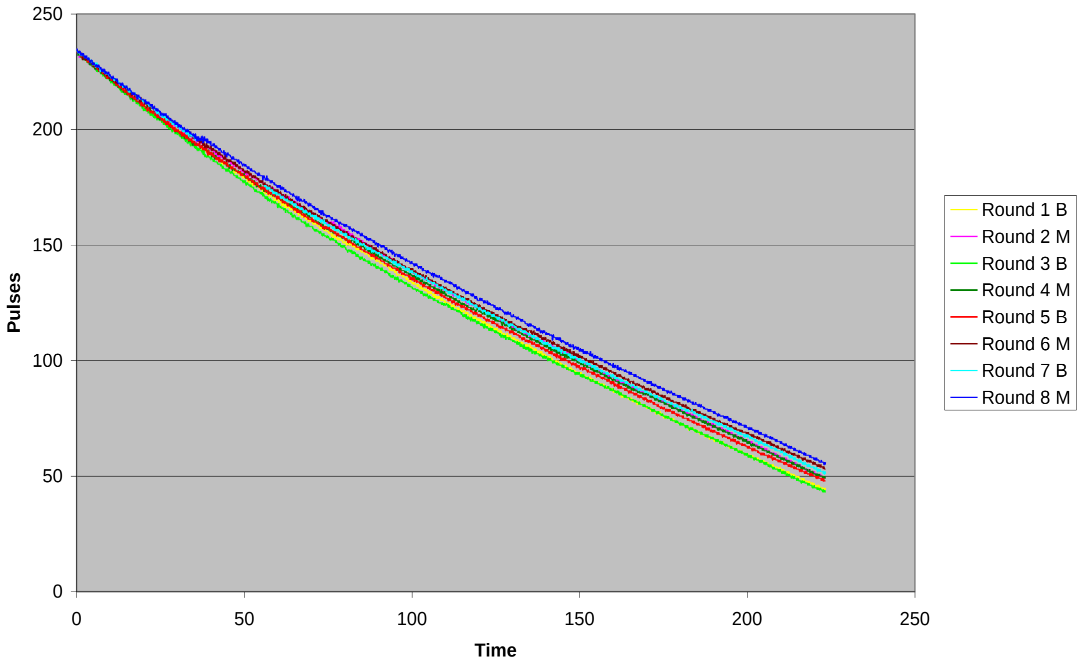
```
clc
clear all
close all

R_d=287.05; %[J/kg*K]
R_v=461.495; %[J/kg*K]

% Summer reference Volvo FH
phi_ref=0.80;
T_ref=15+273.12; %[K]
p_ref=102050; %[Pa]

p_ref_sat=6.1078*10^((7.5*T_ref-2048.625)/(T_ref-35.85))*100; %[Pa]
p_ref_v=phi_ref*p_ref_sat;
p_ref_d=p_ref-p_ref_v;
rho_air_ref=p_ref_d/(R_d*T_ref)+p_ref_v/(R_v*T_ref)

% Competitor air density
% [Scania, Mercedes, Volvo, Renault, DAF]
T_period_S=[-5.8, -4.9, -4.1, -3.7, -3.1, -2.6, -2.1]+273.13; %[K]
T_period_M=[-5.3, -4.6, -4.6, -4.3, -3.8, -3.7, -3.4]+273.13; %[K]
T_period_V=[-6,   -5.7, -5.6, -5.3, -5.1, -4.9, -4.5, -4.3]+273.13;
%[K]
T_period_R=[-4.6, -4.7, -3.4, -3.2]+273.13; %[K]
T_period_D=[-2.9, -2.1, -2.1, -1.6]+273.13; %[K]

T=[mean(T_period_S);
   mean(T_period_M);
   mean(T_period_V);
   mean(T_period_R);
   mean(T_period_D)];



phi_period_S=[94,   95,   94.7, 94.6, 94.5, 94,   93.8]/100;
phi_period_M=[96.3, 96.5, 96.6, 96.7, 97.1, 96.8, 96.9]/100;
phi_period_V=[96.1, 96.2, 96.2, 96.3, 96.4, 96.6, 96.6, 96.5]/100;
phi_period_R=[96,   95.7, 96.2, 95.6]/100;
phi_period_D=[67.4, 57.3, 55.9, 54.2]/100;

phi=[mean(phi_period_S);
     mean(phi_period_M);
     mean(phi_period_V);
     mean(phi_period_R);
     mean(phi_period_D)];
```

```
p_period_S=[994, 994, 994, 994, 993, 993, 994]*100; %[Pa]
p_period_M=[990, 990, 990, 990, 989, 989, 989]*100; %[Pa]
p_period_V=[996, 997, 997, 997, 997, 997, 997, 997]*100; %[Pa]
p_period_R=[981, 981, 981, 981]*100; %[Pa]
p_period_D=[987, 987, 988, 988]*100; %[Pa]

p=[mean(p_period_S);
   mean(p_period_M);
   mean(p_period_V);
   mean(p_period_R);
   mean(p_period_D)];
p_sat=6.1078.*10.^((7.5.*T-2048.625)./(T-35.85)).*100; %[Pa]
p_v=phi.*p_sat;
p_d=p-p_v;

rho_air=p_d./(R_d.*T)+p_v./(R_v.*T)

%Comparison
Percent=(1-rho_air_ref./rho_air).*100
```

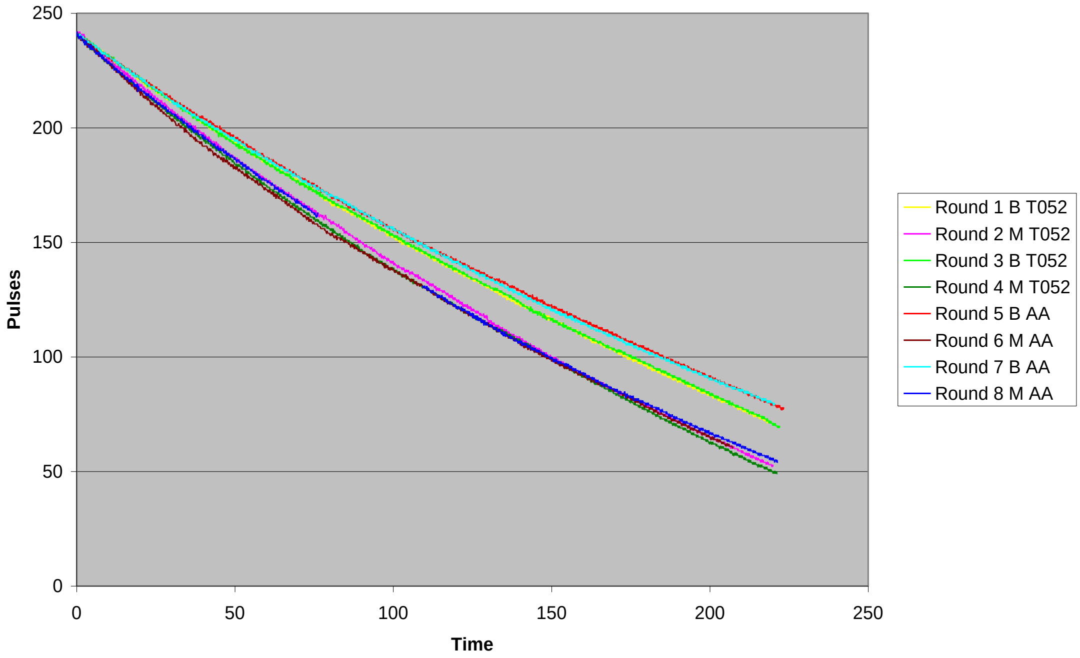
Appendix D (Coast down runs variations)



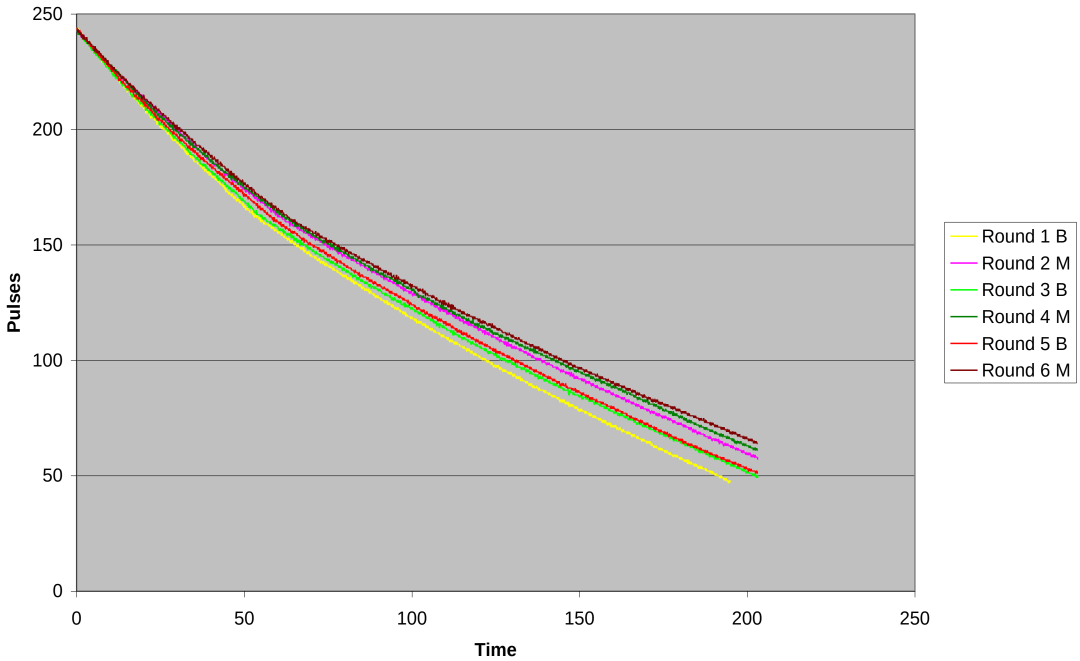
Appendix D 1: Scania coast down runs



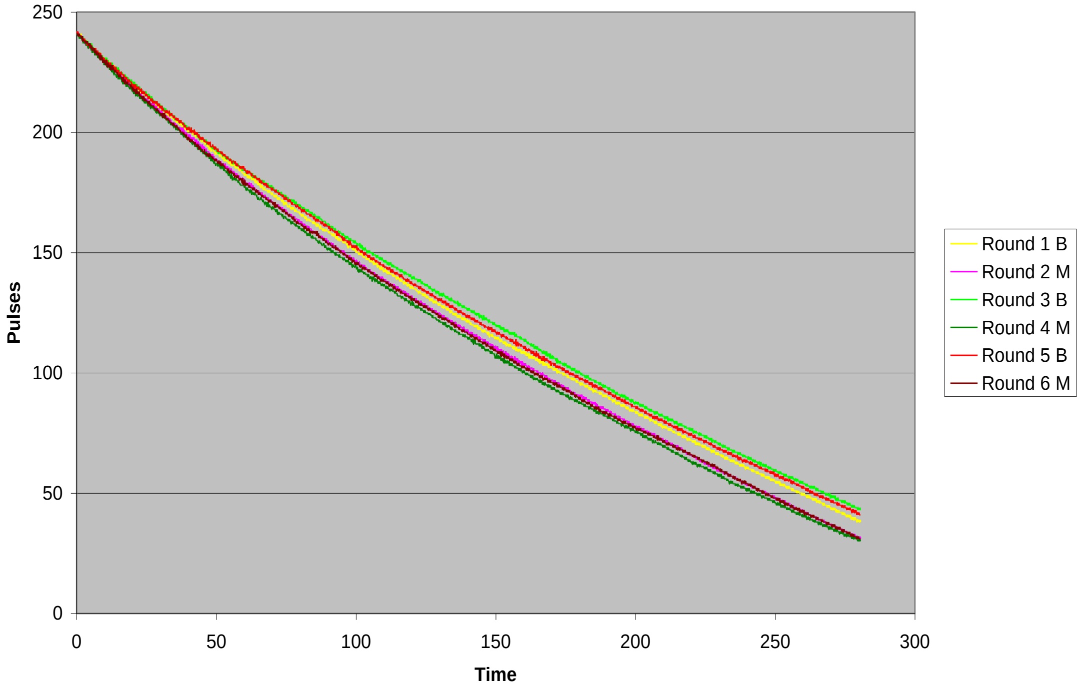
Appendix D 2: Mercedes coast down runs



Appendix D 3: Volvo coast down runs



Appendix D 4: Renault coast down runs



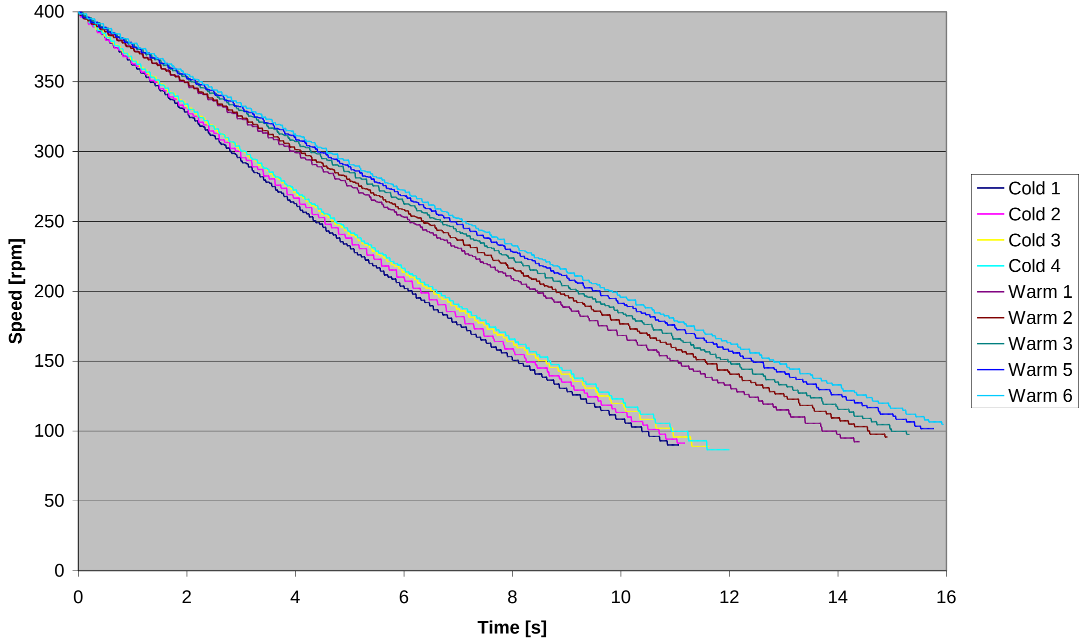
Appendix D 5: DAF coast down runs

Appendix E (Vehicle tires at coast down)

Appendix E 1: Tires used on vehicles at coast down test

Vehicle:	**Front**			**Rear**		
	Dimensions:	**Brand:**	**Model:**	**Dimensions:**	**Brand:**	**Model:**
Scania R480	385/65 R22.5	Bridgestone	M788	315/70 R22.5	Bridgestone	M729
Mercedes Actros	315/60 R22.5	Michelin	Energy XZA2	315/70 R22.5	Michelin	Energy XDA2+
Volvo FH	315/70 R22.5	Michelin	Energy XZA2	315/70 R22.5	Michelin	Energy XDA2
Renault Premium	315/70 R22.5	Michelin	Energy XZA2	315/70 R22.5	Michelin	Energy XDA2+
DAF XF	385/65 R22.5	Goodyear	Marathon LHS II	315/70 R22.5	Goodyear	Marathon LHD II
T052 Trailer	425/55 R19.5	Dunlop	SP241			
AA Trailer	385/55 R22.5	Michelin	Energy XTA2			

Appendix F (Powertrain coast down runs)



Appendix F 1: The collected data from the individual powertrain coast down runs of the Volvo FH

Appendix G (Inertia of powertrain components)

Appendix G 1: Inertia of powertrain components of Volvo FH13

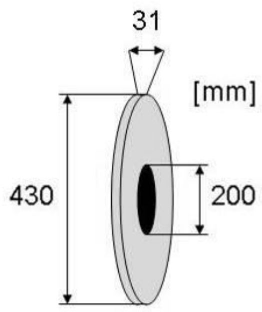
	Inertia [kgm^2]
$I_{dual\ tire}$	29.218 [1]
$I_{drive\ shaft}$	0.023 [2]
$I_{differential}$	1.347 [3]
$I_{brake\ disc}$	0.646 [4]
I_{pinion}	0.075 [5]
$I_{propeller\ shaft}$	0.201 [6]
$I_{gearbox}$	0.4 [7]
I_{hub}	0.523 [8]

[1] From: Elie Garcia, Volvo 3P, department 857
[2] From: Philippe Bronn, Volvo Powertrain, department 264
[3] From: Philippe Bronn, Volvo Powertrain, department 264
[4] From: Calculation and measurement, see Appendix H
[5] From: Philippe Bronn, Volvo Powertrain, department 264
[6] From: Jean-Paul Febvre, Volvo Powertrain, department 92551
[7] From: Anders Hedman, Volvo Powertrain, department 91554
[8] From: Calculation and measurement, see Appendix H

Appendix H (Calculating inertia of brake disc and rear wheel hub)

Brake disc

The brake disc at the rear was ventilated and 45 mm wide, but because 17 mm of the thickness was ventilated, an approximation that the effective massive thickness is around 31 mm was used, see Appendix H 1. To calculate the inertia Equation H 1 was used. The mass used in the equation was measured on a scale.



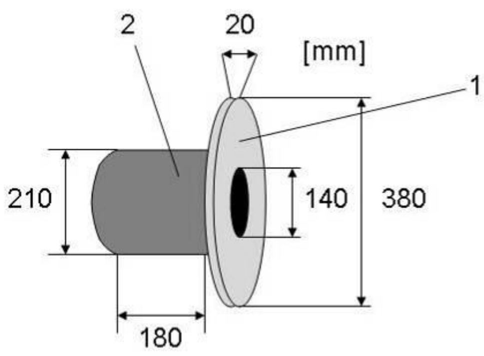
Appendix H 1: Measured dimensions of rear ventilated brake disc

$$I_{brakedisc} = \frac{1}{2} m_b \left(r_{inner}^2 + r_{outer}^2 \right) = \frac{1}{2} * 22.99 * \left(0.1^2 + 0.215^2 \right) = 0.646\, kgm^2$$

Equation H 1

Wheel hub

The wheel hub weight could not be measured so an approximation of the mass was conducted by calculating the volume of the shape that was assumed to be two cylinders, see Appendix H 2. The weights of the two components are calculated by assuming a density of 7800 kg/m^2 for iron. The volume of component 1 and 2 respectively are calculated in Equation H 2 and Equation H 3.



Appendix H 2: Measured dimensions of rear wheel hub

$$V_1 = r_{1,outer}{}^2 t_1 \pi - r_{1,inner}{}^2 t_1 \pi = 0.19^2 * 0.02 * \pi - 0.14^2 * 0.02 * \pi = 0.00196\, m^3$$

Equation H 2

$$V_2 = r_{2,outer}{}^2 t_2 \pi - r_{2,inner}{}^2 t_2 \pi = 0.105^2 * 0.18 * \pi - 0.07^2 * 0.18 * \pi = 0.00346\, m^3$$

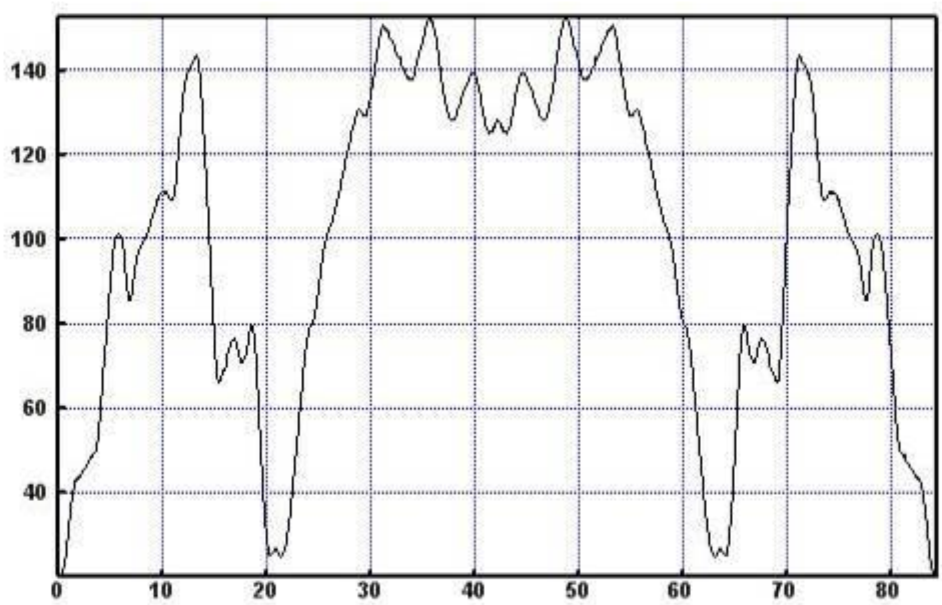
Equation H 3

By multiplying the volumes, (Equation H 2 and Equation H 3), by the density, the mass of each part are achieved as 15 kg and 27 kg for part 1 and 2 respectively. The inertia of the wheel hub is now calculated in Equation H 4.

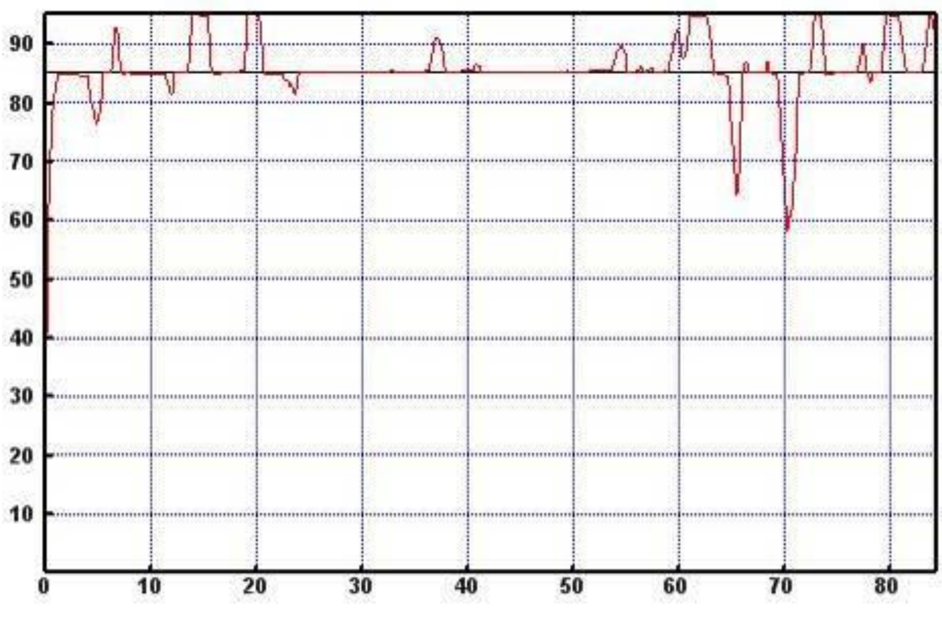
$$I_{wheelhub} = \frac{1}{2} m_{W1} \left(r_{1,inner}{}^2 + r_{1,outer}{}^2 \right) + \frac{1}{2} m_{W2} \left(r_{2,inner}{}^2 + r_{2,outer}{}^2 \right) = 0.523\, kgm^2$$

Equation H 4

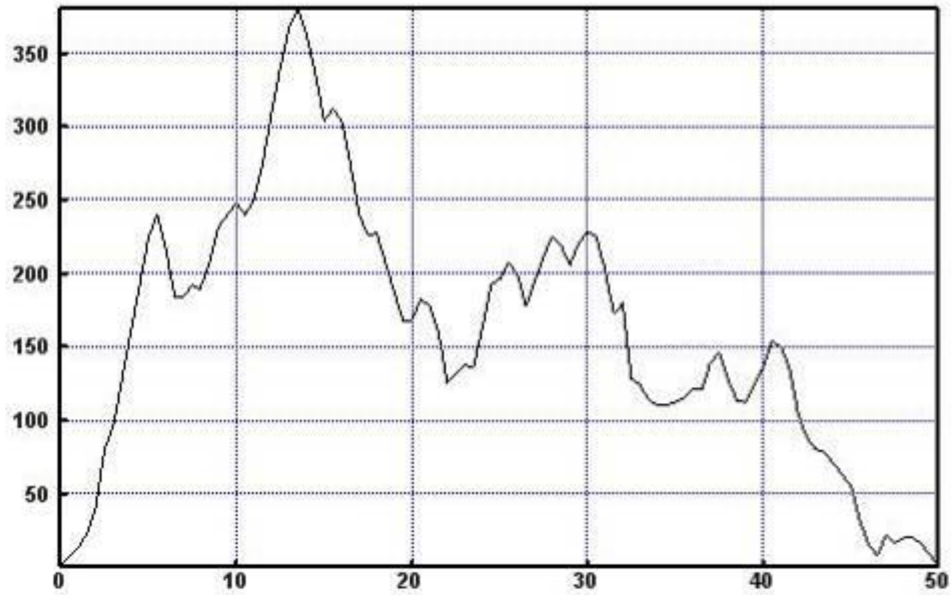
Appendix I (Duty cycles PERF)



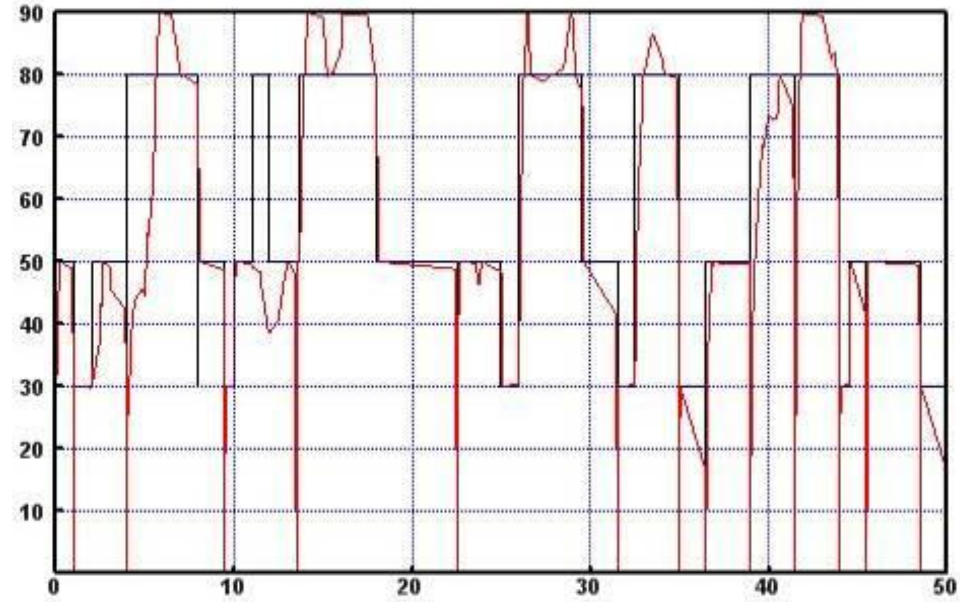
Appendix I 1: Lv-Bo-Lv duty cycle, distance [km] on x-axis and road elevation [m] on y-axis



Appendix I 2: Lv-Bo-Lv target speed, distance [km] on x-axis and speed [km/h] on y-axis. The red graph is the actual speed while the black (straight line at 85 km/h) is the target speed



Appendix I 3: Regional distribution hilly duty cycle, distance [km] on x-axis and road elevation [m] on y-axis



Appendix I 4: Regional distribution hilly target speed, distance [km] on x-axis and speed [km/h] on y-axis. The red graph is the actual speed while the black is the target speed

Appendix J (Simulation results PERF)

	Reference vehicle	Decreased rolling resistance	Increased rolling resistance
Engine:	D13C460 EU5	D13C460 EU5	D13C460
Gearbox:	AT2512C	AT2512C	AT2512C
Rear axle:	RSS1344C	RSS1344C	RSS1344C
Rear axle ratio:	2.64:1	2.64:1	2.64:1
Tires:	315/70 R22.5	315/70 R22.5	315/70 R22.5
Number of tires:	12	12	12
Weight [tonnage]:	40	40	40
Drive axle pressure [tonnage]:	11	11	11
Frontal area [m^2]:	9.7	9.7	9.7
Air resistance:	0.53	0.53	0.53
Rolling resistance:	0.0051	**0.0041**	**0.006**
Coefficient of friction:	0.8	0.8	0.8
Application:	Combination	Combination	Combination
Speed limiter [km/h]:	90	90	90
Wind speed [m/s]:	0	0	0
Target speed [km/h]:	85 (80 RDH)	85 (80)	85 (80)
Lv-Bo-Lv			
Fuel consumption [l/100 km]:	**30.96**	**29.42**	**32.49**
Average speed [km/h]:	84.57	84.78	84.36
Gear shifts:	28	26	28
Average torque [Nm]:	1267.1	1250.9	1290.3
Average power [kW]:	110.5	105.2	115.9
Regional distribution hilly			
Fuel consumption [l/100 km]:	**55.24**	**54.03**	**56.44**
Average speed [km/h]:	48.57	48.63	48.49
Gear shifts:	102	102	104
Average torque [Nm]:	1403.0	1401.8	1433.5
Average power [kW]:	111.5	109.2	113.8

	Decreased weight	**Increased weight**	**Larger tires**
Engine:	D13C460 EU5	D13C460 EU5	D13C460 EU5
Gearbox:	AT2512C	AT2512C	AT2512C
Rear axle:	RSS1344C	RSS1344C	RSS1344C
Rear axle ratio:	2.64:1	2.64:1	2.64:1
Tires:	315/70 R22.5	315/70 R22.5	**315/80 R22.5**
Number of tires:	12	12	12
Weight [tonnage]:	**39**	**41**	40
Drive axle pressure [tonnage]:	11	11	11
Frontal area [m^2]:	9.7	9.7	9.7
Air resistance:	0.53	0.53	0.53
Rolling resistance:	0.0051	0.0051	0.0051
Coefficient of friction:	0.8	0.8	0.8
Application:	Combination	Combination	Combination
Speed limiter [km/h]:	90	90	90
Wind speed [m/s]:	0	0	0
Target speed [km/h]:	85 (80)	85 (80)	85 (80)
Lv-Bo-Lv			
Fuel consumption [l/100 km]:	**30.51**	**31.39**	**30.83**
Average speed [km/h]:	84.64	84.48	84.59
Gear shifts:	26	28	33
Average torque [Nm]:	1244.0	1292.3	1333.4
Average power [kW]:	108.8	112.1	110.4
Regional distribution hilly			
Fuel consumption [l/100 km]:	**54.10**	**56.36**	**55.54**
Average speed [km/h]:	48.68	48.44	48.77
Gear shifts:	102	105	106
Average torque [Nm]:	1393.8	1440.8	1414.1
Average power [kW]:	109.3	113.6	112.7

	Smaller tires	Decreased drag	Increased drag
Engine:	D13C460 EU5	D13C460 EU5	D13C460 EU5
Gearbox:	AT2512C	AT2512C	AT2512C
Rear axle:	RSS1344C	RSS1344C	RSS1344C
Rear axle ratio:	2.64:1	2.64:1	2.64:1
Tires:	**315/60 R22.5**	315/70 R22.5	315/70 R22.5
Number of tires:	12	12	12
Weight [tonnage]:	40	40	40
Drive axle pressure [tonnage]:	11	11	11
Frontal area [m^2]:	9.7	9.7	9.7
Air resistance:	0.53	**0.5**	**0.56**
Rolling resistance:	0.0051	0.0051	0.0051
Coefficient of friction:	0.8	0.8	0.8
Application:	Combination	Combination	Combination
Speed limiter [km/h]:	90	90	90
Wind speed [m/s]:	0	0	0
Target speed [km/h]:	85 (80)	85 (80)	85 (80)
Lv-Bo-Lv			
Fuel consumption [l/100 km]:	**31.16**	**30.48**	**31.42**
Average speed [km/h]:	84.59	84.61	84.50
Gear shifts:	24	28	28
Average torque [Nm]:	1211.7	1260.2	1276.2
Average power [kW]:	110.8	108.8	112.1
Regional distribution hilly			
Fuel consumption [l/100 km]:	**55.47**	**55.00**	**55.28**
Average speed [km/h]:	48.37	48.57	48.53
Gear shifts:	93	104	104
Average torque [Nm]:	1355.4	1419.2	1423.8
Average power [kW]:	111.1	111.2	111.7

	Lower trailer -10 cm	**500 hp engine**	**540 hp engine**
Engine:	D13C460 EU5	**D13C500 EU5**	**D13C540 EU5**
Gearbox:	AT2512C	AT2512C	AT2512C
Rear axle:	RSS1344C	RSS1344C	RSS1344C
Rear axle ratio:	2.64:1	2.64:1	2.64:1
Tires:	315/70 R22.5	315/70 R22.5	315/70 R22.5
Number of tires:	12	12	12
Weight [tonnage]:	40	40	40
Drive axle pressure [tonnage]:	11	11	11
Frontal area [m^2]:	**9.45**	9.7	9.7
Air resistance:	0.53	0.53	0.53
Rolling resistance:	0.0051	0.0051	0.0051
Coefficient of friction:	0.8	0.8	0.8
Application:	Combination	Combination	Combination
Speed limiter [km/h]:	90	90	90
Wind speed [m/s]:	0	0	0
Target speed [km/h]:	85 (80)	85 (80)	85 (80)
Lv-Bo-Lv			
Fuel consumption [l/100 km]:	**30.75**	**31.07**	**31.20**
Average speed [km/h]:	84.60	84.98	85.12
Gear shifts:	28	24	24
Average torque [Nm]:	1264.2	1284.0	1290.8
Average power [kW]:	109.8	111.2	111.4
Regional distribution hilly			
Fuel consumption [l/100 km]:	**55.08**	**55.63**	**56.45**
Average speed [km/h]:	48.56	49.00	49.45
Gear shifts:	104	95	93
Average torque [Nm]:	1420.4	1473.7	1493.5
Average power [kW]:	111.3	113.3	114.5

	420 hp engine	**380 hp engine**	**Higher ratio rear axle +**
Engine:	**D13C420 EU5**	**D13C380 EU5**	D13C460 EU5
Gearbox:	AT2512C	AT2512C	AT2512C
Rear axle:	RSS1344C	RSS1344C	RSS1344C
Rear axle ratio:	2.64:1	2.64:1	**2.85:1**
Tires:	315/70 R22.5	315/70 R22.5	315/70 R22.5
Number of tires:	12	12	12
Weight [tonnage]:	40	40	40
Drive axle pressure [tonnage]:	11	11	11
Frontal area [m^2]:	9.7	9.7	9.7
Air resistance:	0.53	0.53	0.53
Rolling resistance:	0.0051	0.0051	0.0051
Coefficient of friction:	0.8	0.8	0.8
Application:	Combination	Combination	Combination
Speed limiter [km/h]:	90	90	90
Wind speed [m/s]:	0	0	0
Target speed [km/h]:	85 (80)	85 (80)	85 (80)
Lv-Bo-Lv			
Fuel consumption [l/100 km]:	**31.08**	**30.98**	**31.28**
Average speed [km/h]:	83.97	83.30	84.62
Gear shifts:	32	34	22
Average torque [Nm]:	1246.2	1220.7	1183.2
Average power [kW]:	109.5	108.3	111.0
Regional distribution hilly			
Fuel consumption [l/100 km]:	**55.22**	**54.79**	**55.64**
Average speed [km/h]:	47.83	46.82	48.37
Gear shifts:	107	108	91
Average torque [Nm]:	1365.9	1311.0	1339.5
Average power [kW]:	109.2	106.2	111.4

	Higher ratio rear axle ++	**Higher target speed**	**Lower target speed**
Engine:	D13C460 EU5	D13C460 EU5	D13C460 EU5
Gearbox:	AT2512C	AT2512C	AT2512C
Rear axle:	RSS1344C	RSS1344C	RSS1344C
Rear axle ratio:	**3.08:1**	2.64:1	2.64:1
Tires:	315/70 R22.5	315/70 R22.5	315/70 R22.5
Number of tires:	12	12	12
Weight [tonnage]:	40	40	40
Drive axle pressure [tonnage]:	11	11	11
Frontal area [m^2]:	9.7	9.7	9.7
Air resistance:	0.53	0.53	0.53
Rolling resistance:	0.0051	0.0051	0.0051
Coefficient of friction:	0.8	0.8	0.8
Application:	Combination	Combination	Combination
Speed limiter [km/h]:	90	90	90
Wind speed [m/s]:	0	0	0
Target speed [km/h]:	85 (80)	**90**	**80**
Lv-Bo-Lv			
Fuel consumption [l/100 km]:	**31.67**	**31.85**	**30.15**
Average speed [km/h]:	84.75	89.06	80.10
Gear shifts:	10	24	32
Average torque [Nm]:	1104.6	1289.8	1253.3
Average power [kW]:	111.2	120.0	101.6
Regional distribution hilly			
Fuel consumption [l/100 km]:	**55.51**		
Average speed [km/h]:	48.71		
Gear shifts:	98		
Average torque [Nm]:	1423.1		
Average power [kW]:	112.6		

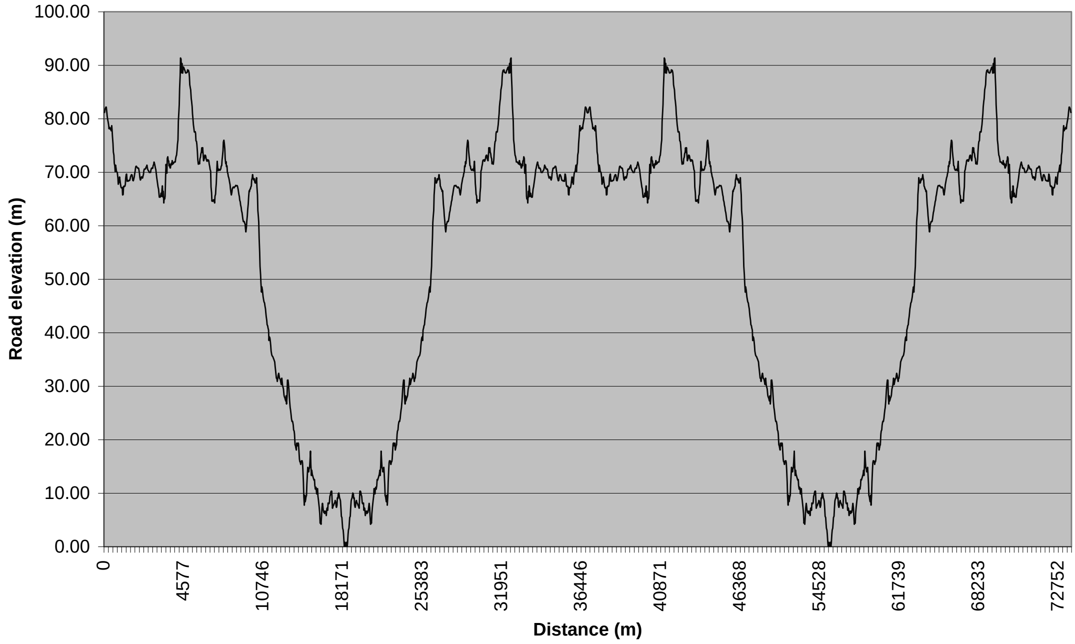
	2.5 m/s wind speed	**5.0 m/s wind speed**
Engine:	D13C460 EU5	D13C460 EU5
Gearbox:	AT2512C	AT2512C
Rear axle:	RSS1344C	RSS1344C
Rear axle ratio:	2.64:1	2.64:1
Tires:	315/70 R22.5	315/70 R22.5
Number of tires:	12	12
Weight [tonnage]:	40	40
Drive axle pressure [tonnage]:	11	11
Frontal area [m^2]:	9.7	9.7
Air resistance:	0.53	0.53
Rolling resistance:	0.0051	0.0051
Coefficient of friction:	0.8	0.8
Application:	Combination	Combination
Speed limiter [km/h]:	90	90
Wind speed [m/s]:	**2.5**	**5.0**
Target speed [km/h]:	85 (80)	85 (80)
Lv-Bo-Lv		
Fuel consumption [l/100 km]:	**32.82**	**34.93**
Average speed [km/h]:	84.33	84.04
Gear shifts:	28	30
Average torque [Nm]:	1297.1	1327.0
Average power [kW]:	117.1	124.5
Regional distribution hilly		
Fuel consumption [l/100 km]:	**56.25**	**57.32**
Average speed [km/h]:	48.52	48.41
Gear shifts:	104	104
Average torque [Nm]:	1428.3	1439.4
Average power [kW]:	113.4	115.5

Appendix K (Simulation results of optimized vehicles in PERF)

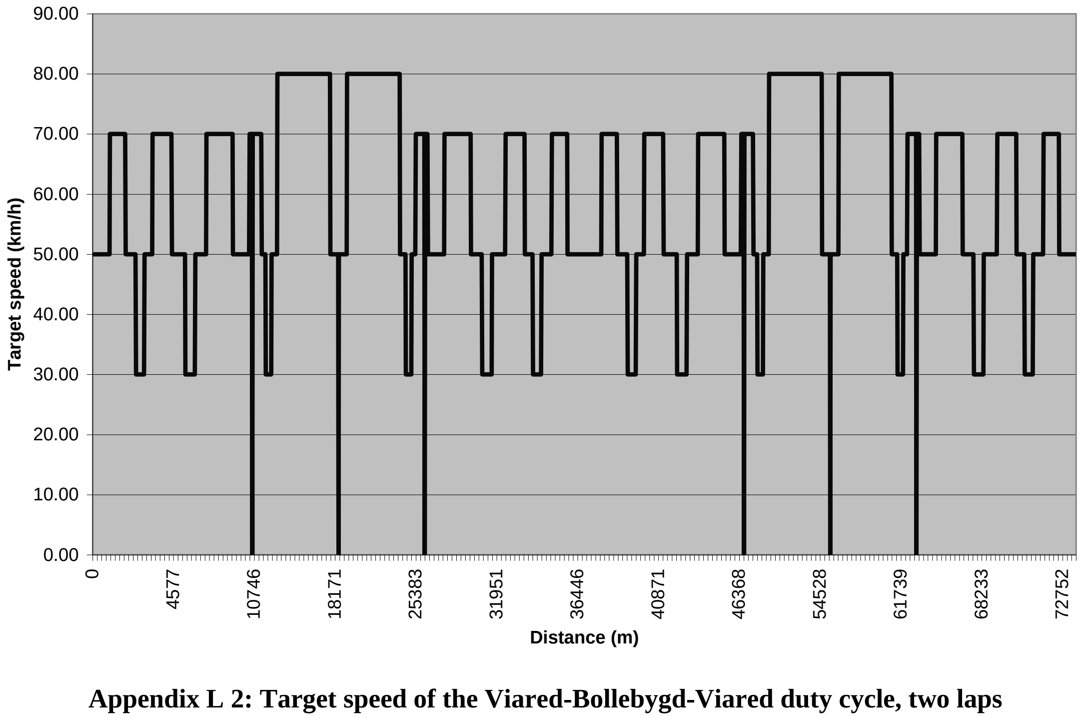
	Tested Volvo vehicle	**Optimized vehicle 460 hp**	**Optimized vehicle 380 hp**
Engine:	**D13C500 EU5**	D13C460 EU5	**D13C380 EU5**
Gearbox:	AT2512C	AT2512C	AT2512C
Rear axle:	RSS1344C	RSS1344C	RSS1344C
Rear axle ratio:	2.64:1	2.64:1	2.64:1
Tires:	315/70 R22.5	**315/80 R22.5**	**315/80 R22.5**
Number of tires:	12	12	12
Weight [tonnage]:	**40.12**	**39**	**39**
Drive axle pressure [tonnage]:	11	11	11
Frontal area [m^2]:	**10.08**	**9.45**	**9.45**
Air resistance:	0.53	**0.5**	**0.5**
Rolling resistance:	0.0051	**0.0041**	**0.0041**
Coefficient of friction:	0.8	0.8	0.8
Application:	Combination	Combination	Combination
Speed limiter [km/h]:	90	90	90
Wind speed [m/s]:	0	0	0
Target speed [km/h]:	85 (80)	85 (80)	85 (80)
Lv-Bo-Lv			
Fuel consumption [l/100 km]:	**31.44**	**28.23**	**28.25**
Average speed [km/h]:	84.93	85.02	83.81
Gear shifts:	24	30	35
Average torque [Nm]:	1293.3	1280.5	1233.2
Average power [kW]:	112.5	101.2	99.2
Regional distribution hilly			
Fuel consumption [l/100 km]:	**55.86**	**53.00**	**52.68**
Average speed [km/h]:	48.95	49.01	47.26
Gear shifts:	95	108	114
Average torque [Nm]:	1478.5	1383.2	1265.8
Average power [kW]:	113.7	107.6	102.4

	Low rolling resistance 460 hp	**Low rolling resistance 380 hp**
Engine:	D13C460 EU5	**D13C380 EU5**
Gearbox:	AT2512C	AT2512C
Rear axle:	RSS1344C	RSS1344C
Rear axle ratio:	2.64:1	2.64:1
Tires:	315/70 R22.5	315/70 R22.5
Number of tires:	12	12
Weight [tonnage]:	40	40
Drive axle pressure [tonnage]:	11	11
Frontal area [m^2]:	9.7	9.7
Air resistance:	0.53	0.53
Rolling resistance:	**0.0041**	**0.0041**
Coefficient of friction:	0.8	0.8
Application:	Combination	Combination
Speed limiter [km/h]:	90	90
Wind speed [m/s]:	2.5	5.0
Target speed [km/h]:	85 (80)	85 (80)
Lv-Bo-Lv		
Fuel consumption [l/100 km]:	**29.42**	**29.46**
Average speed [km/h]:	84.78	83.59
Gear shifts:	26	34
Average torque [Nm]:	1250.9	1203.7
Average power [kW]:	105.2	103.2
Regional distribution hilly		
Fuel consumption [l/100 km]:	**54.03**	**53.58**
Average speed [km/h]:	48.63	47.00
Gear shifts:	102	109
Average torque [Nm]:	1401.8	1300.2
Average power [kW]:	109.2	104.3

Appendix L (Viared-Bollebygd-Viared duty cycle)



Appendix L 1: Road elevations of the Viared-Bollebygd-Viared duty cycle, two laps



Appendix L 2: Target speed of the Viared-Bollebygd-Viared duty cycle, two laps

Appendix M (VFL parameters for simulations)

Appendix M 1: VFL parameters for the simulations

Parameter:	**Value and unit:**
Fuel heat factor	42.9 MJ/kg
Fuel air ratio (λ=1) mass	14.58
Fuel density liquid	0.8355 kg/dm^3
Fuel hydrogen/carbon ratio molar	1.853
Surrounding temperature	10 or 25 ºC

Appendix N (Vehicle specifications at road test)

Vehicle	**Scania R480**	**Mercedes Actros 1848**
Odo-meter	3’236 km	12’330 km
Engine	DC13/480 hp/EGR/EuroV	12L/476 hp/SCR/EuroV
Gearbox	GRS 905R opticruise	MB Powershift G271
Gears	12	12
Rear axle ratio	2.71:1	2.53:1
Tire maker	Michelin	Michelin
Tire type front/rear	XF/XD Energy	XDA 2+ Energy
Tire dimension	315/70 R22.5	315/60 R22.5
Roof deflector	Yes	Yes
Side deflectors	Yes	Yes
Chassis skirts	No	Yes
Front spoiler	No	Yes
Trailer tire maker	Michelin	Michelin
Trailer tire type	XTA 2 Energy	XTA 2 Energy
Trailer tire dimension	385/55 R22.5	385/55 R22.5
Tire pressure tractor/trailer	8 / 9 bars	8 / 9 bars
Total length	16.5 m	16.5 m
Tractor height	4.06 m	3.93 m
Trailer height	3.87 m	3.91 m
Total width	2.6 m	2.6 m
Trailer gap	0.24 m	0.23 m
Side deflector distance to trailer	0.335 m	0.23 m
Weight AA trailer	40’320 kg	39’920 kg
Weight BL trailer	40’600 kg	40’720 kg
Outdoor temperature	-1 to +3 ºC	+13 ºC

Vehicle	**Volvo FH500**	**DAF XF105**
Odo-meter	28’788 km	120’730 km
Engine	D13C/500 hp/EuroV	MX12.9L/460 hp/EuroV
Gearbox	AT2612D	ZF/AS tronic
Gears	12	12
Rear axle ratio	2.64:1	2.69:1
Tire maker	Michelin	Michelin
Tire type front/rear	XF/XD Energy	XF/XD Energy
Tire dimension	315/70 R22.5	315/70 R22.5
Roof deflector	Yes	Yes
Side deflectors	Yes	Yes
Chassis skirts	Yes	Yes
Front spoiler	Yes	No
Trailer tire maker	Michelin	Michelin
Trailer tire type	XTA 2 Energy	XTA 2 Energy
Trailer tire dimension	385/55 R22.5	385/55 R22.5
Tire pressure tractor/trailer	8 / 9 bars	8 / 9 bars
Total length	16.5 m	16.5 m
Tractor height	4.07 m	4.06 m
Trailer height	3.91 m	3.92 m
Total width	2.6 m	2.6 m
Trailer gap	0.245 m	0.24 m
Side deflector distance to trailer	0.415 m	0.27 m
Weight AA trailer	40’120 kg	40’220 kg
Weight BL trailer	- kg	40’500 kg
Outdoor temperature	+5 to +7 ºC (slight rain)	-3.5 to +2 ºC

Vehicle	**Volvo FH13 Reference**
Odo-meter	30’640 km
Engine	D13C/500 hp/EuroV
Gearbox	AT2612D
Gears	12
Rear axle ratio	2.64:1
Tire maker	Michelin
Tire type front/rear	XZA2/XZD2 Energy
Tire dimension	315/70 R22.5
Roof deflector	Yes
Side deflectors	Yes
Chassis skirts	Yes
Front spoiler	Yes
Trailer tire maker	Michelin
Trailer tire type	XTA 2 Energy
Trailer tire dimension	385/55 R22.5
Tire pressure tractor/trailer	8 / 9 bars
Total length	16.7 m
Tractor height	4.05 m
Trailer height	3.90 m
Total width	2.6 m
Trailer gap	0.34 m
Side deflector distance to trailer	0.40 m
Weight AA trailer	40’360 kg (D), 40’380 kg(V), -(M) and 40’180 (S) kg
Weight BL trailer	40’520 kg (D), 40’500 kg(V), -(M) and 40’460 (S) kg

Note! (D) - DAF, (V) - Volvo, (M) - Mercedes and (S) - Scania