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Dimensioning and Mobility of a Vehicle Combination

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Summary

This project is about the study and design of a combined vehicle, specifically a truck with a three-axle tandem trailer.

The project will calculate the driving resistances of this kind of vehicles and all the forces related to them. The driving resistances calculated will be used to find the power and torque that the truck engine needs. Then those parameters will be used to find an adequate engine for the truck that enables it to circulate and move properly.

In this project a gearbox is also going to be designed to calculate the power and torque that arrives at the driving wheels considering the loss of power. With this power that arrives at the driving wheels it will be possible to calculate the acceleration of all combined vehicle.

Finally there is presented some theory about vehicle dynamics and the force needed to unload the trailer using different unload systems will be determined and compared.

Contents

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1. Glossary

α	Inclination of the road angle
r	Radius of the wheels
C_{rr}	Rolling resistance coefficient
ρ	Density of the fluid
v	Speed of the set truck-trailer
A	Cross sectional area
C_d	Air resistance coefficient
F_h	Hill resistance force
F_y	Vertical component of the weight
F_{ac}	Acceleration resistance force
m, m_{set}	Total mass of the set truck-trailer
h_d	Distance between the ground and the hitch
ε	Distance between the application of the normal force at the wheel and its central point
P_{engine}	Power done by the engine
P_w	Power that arrives at the wheel
P_{loss}	Total power loss by the transmission system
η_{trans}	Efficiency of the transmission
η_i	Efficiency of the gearbox
η_{kar}	Efficiency of the cardan
η_0	Efficiency of the final gear
η_p	Efficiency of the differential
η_{clutch}	Efficiency of the clutch
i_{total}	Total gear ratio of the transmission system
i_{gear}	Gear ratio of each gear
i_{kar}	Cardan ratio
i_0	Final drive gear ratio
i_p	Wheel gear ratio
Γ_{wheel}	Torque that the wheel needs to do to move the set truck-trailer
F_w	Force that the wheel needs to do to move the set truck-trailer
Γ_{engine}	Torque that the engine does
μ	Friction coefficient between the tack and the wheels
EG	Self-steering gradient
SG	Angle gradient
F_{cf}	Centrifugal force on cornering
r_k	Radius of the curve
Tm	Tones (1000kg)

Specific of the trailer:

L_r	Distance between the hitch and the point of application of the trailer's wheels normal forces
L_ϕ	Distance between the hitch and the center of gravity of the trailer
h_a	Trailer high

h	Distance between the floor of the trailer and its center of gravity (CG)
h_2	Distance between the ground and the center of gravity of the trailer
m_2, m_3, m_4	Masses that the wheels of the trailer have to hold
m_b	Mass hold by the tandem of the trailer
N_1	Normal force at the hitch
N_2, N_3, N_4	Normal forces at the wheels of the trailer
F_1	Longitudinal force at the hitch
F_a	Trailer air resistance force
F_r	Rolling resistance force
P_a	Air resistance power
μ	Friction coefficient between the load and the trailer walls
γ	Density of the load
Z	Deep of the load
ø	Excluded volume of the load
R	Hydraulic radius
P_v	Vertical pressure
P_h	Horizontal pressure
L	Length of the trailer
B	Width of the trailer
F_{f1}	Friction force between the load and one wall of the trailer
F_{f2}	Friction force between the load and one wall of the trailer
F_{ch}	Force that the chains must do to move the load
$\mu_{belt\text{-}trrailer\ floor}$	Friction coefficient between the chains and the trailer floor
F_{pv}	Vertical force of the piston in the dumping unload system
F_p	Total force of the piston in the dumping unload system
β	Angle of inclination of the trailer in the dumping unload system
γ	Angle of inclination of the piston in the dumping unload system

Specific of the truck:

L_x	Distance between the axles of the truck
L_e	Distance between the driving axle and the center of gravity of the truck
L_d	Distance between the driving axle and the hitch
ICR	Instantaneous center of rotation of the wheels
h	Truck high
h_m	High of the center of gravity of the trailer
h_{a1}	Distance between the ground and the point of application of the truck air resistance
N_3	Normal fore at the driving wheels
N_4	Normal force at the front wheels of the trailer
F_{r3}	Rolling resistance force of the driving wheels
F_{r4}	Rolling resistance force of the front wheels of the truck
F_f	Friction force of the driving wheels
F_{a1}	Truck air resistance force
F_{truck}	Force that the set truck-trailer needs to be moved

Specific of the suspension:

F	Force of the string
A	Effective working area
p	Internal air pressure
p_a	Ambient air pressure
c	Spring rate
V	Air volume of the string

2. Preface

2.1. Introduction

This thesis will study the mobility of a combined vehicle. The research will be focused in the driving resistances, the engine system, considering the pollution emission, and all the components of the transmission part. However the project will study the break, suspension and tires that fit better in the vehicle.

On the other hand in this thesis theory of vehicle dynamics will be presented, relating it with the hard-duty combined vehicles shape and nature.

The principal way of research of this thesis will be by theory research.

2.2. Target

The main aim of this thesis is to design and determinate the parameters of the mechanical part of a truck and become familiar with the vehicle mobility and dimensioning of the vehicle combination.

The project will work on automotive vehicles, and because of that it has also the aim of let me improve and acquire more knowledge about automotive vehicles (a new field for me) learning how the weight, forces, velocities and another physical parameters affect the system and how it will react and which proceedings we can do .

The last aim of this project is learn other countries ways of working, meet different professors than the ones I had in Barcelona, acquire some of their knowledge and learn other ways of working. And finally improve my English.

3. Legislation

This thesis will be done considering the Spanish law and regulations for this kind of vehicles. That means that all the parameters that can be regulated, forces, distance between axles, length of the vehicle, maximum weight of the vehicle, etc. that will be used for the design of the truck, specially the trailer, will fit the Spanish regulations.

All automotive vehicles in Spain are regulated by the "Reglamento general de vehículos" -General vehicle regulation- approved on 23th of December 1998 by a Real Decreto. This legislation regulates the main general things of automotive vehicles and in this regulation a chapter called "Massas y dimensiones" -masses and dimensions-, which was modified on the 13 of October 2004 by the Order PRE/3298/2004, can be found. This part regulates all measures and weights that automotive vehicles, in Spain, must meet. Is this part were the vehicle must fit to be in the Spanish law.

This regulation indicates all measures masses that every kind of vehicle must have. And researching which vehicle is the most weight vehicle allowed to circulate in Spain without any extraordinary permission it can be seen that a combined vehicle is the best solution. That combined vehicle must be a set of a trailer and a truck.

3.1. Truck

The Spanish law defines a truck as a vehicle designed to carry material with a cabin with no more capacity than 9 people counting the driver.

The kind of tractor that the combined vehicle has can have 2 or 3 axles (one or two driven axles). Its maximum weight cannot be higher than 18000kg for the 2 axle one or 26000kg for the three axle one. That truck cannot be taller than 4m (Figure 2) and cannot measure more than 2,55 m of with.

Each axle of the truck has a maximum weight that it can hold. Non-drive axles can hold 10 tones each and drive axles can hold 11,5 tones. This numbers are the same for 2 or 3 axle trucks.

3.2. Trailer

The Spanish law defines a trailer as a non powered vehicle build to be coupled to an automobile holding a substantial part of its weight.

The maximum weight that the trailer can have, considering the empty weight and the load weight, is 24 Tm for the three axle trailers. The maximum weight that a tandem with three axles can hold is 24 Tm (8 Tm per axle) if the axle distance is between 1,30m and 1,40m or 21 Tm (7 Tm per axle) if the axle is equal or inferior to 1,30m.

The trailer must do 13,6m as a maximum length, its width can be 2,55m and the maximum high of the trailer from the ground must be 4m.

The hitch part must be closer than 12m to the rear part of the trailer and every front part of the trailer cannot be farther than 2,04m from the hitch.

(All of these measures are represented at Figure 2)

3.3. Combined vehicle

The whole length of the set cannot be superior than 16,5 meters and the maximum weight of it cannot exceed 40 Tm.

The union point between the tractor and the trailer (fifth wheel) must be between 1,15 and 1,30 meters from the floor and less than 4,5 meters from the front part of the truck.

The set truck-trailer has to be able to describe, for both sides, a completely circular trajectory inside an area defined by two concentric circles with interior and exterior radius of 5,30m and 12,50m without that any extreme point of the vehicle passes the line outside the circle.

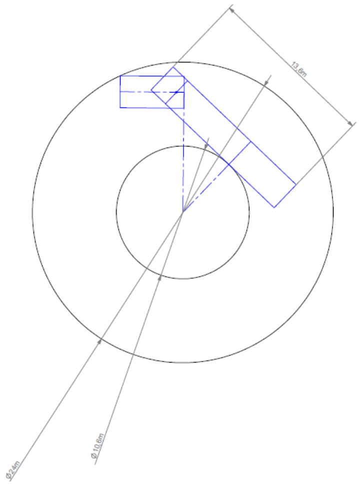


Figure 1 Truck unit in the circular area as stipulated by the Spanish regulations.

(In the figure 1 it can be seen that is possible, with some specific sizes, for the combined vehicle to describe the mentioned circular trajectory)

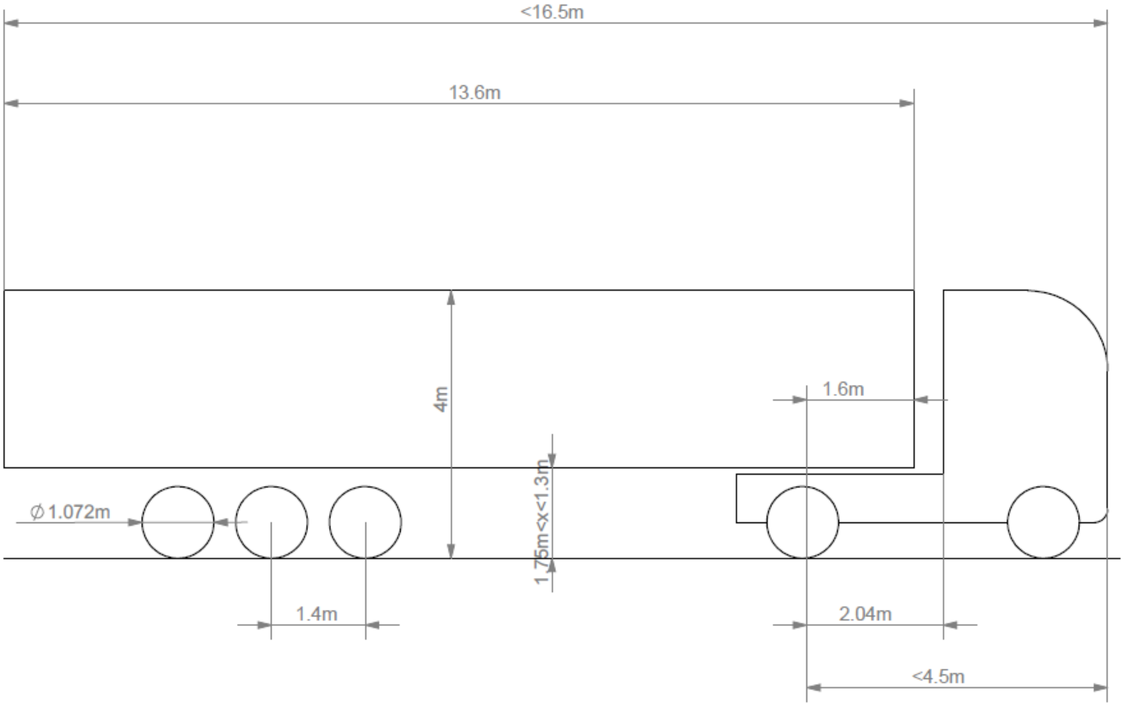


Figure 2 Dimensions of the set truck-trailer.

4. Truck design

4.1. Previous calculations

The aim of this section is do a correct design of the trailer and the truck to fit the legislation. First of all the static calculations must be done to know if every axle has the weight that the Spanish rules allows or overcomes it.

4.1.1. Static calculations

To calculate the load that every axle holds and to know if they fit in the Spanish law is recommendable to separate the trailer and the truck of the combined vehicle to make the calculations easier and clearer.

Trailer calculations:

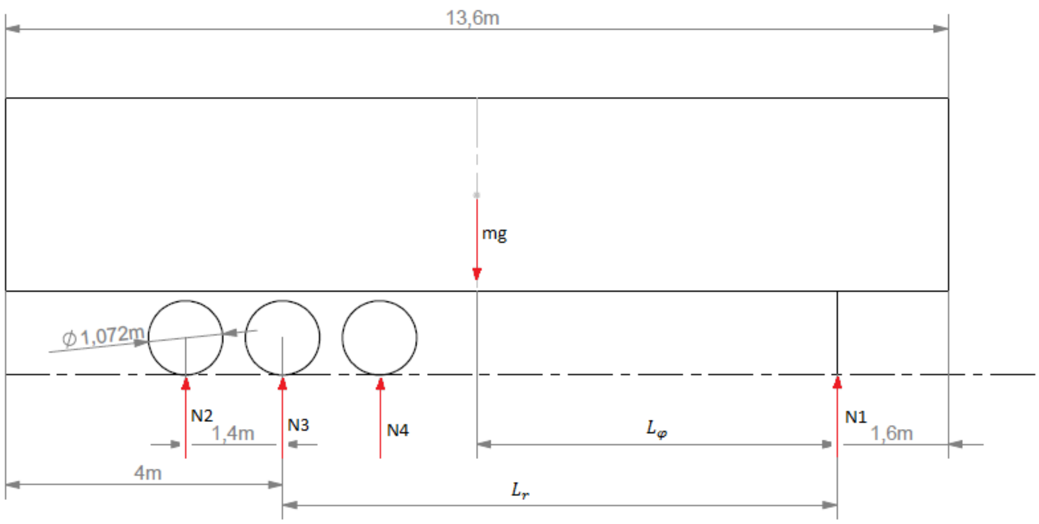


Figure 3 Dimensions of the trailer.

The previous design of the trailer that has to be done to do the static calculations is the figure 3 model. Looking at this model it can be considered that the forces N_2, N_3 and N_4 are equal because of the suspension system and the characteristics of the chassis. To calculate these forces in the axles it will be used the vertical force sum, the sum of moments in the point where N_1 is applied and the mentioned simplification N_2=N_3=N_4.

$mg = 24000 \cdot 9.81 = 235440N$ $\quad L_r = 5{,}2m \quad L_\phi = 8m$

$$\begin{pmatrix} eq.\ 1 \\ eq.\ 2 \\ eq.\ 3 \end{pmatrix} \quad \begin{cases} \Sigma F_x = N_1 + N_2 + N_3 + N_4 - m \cdot g = 0 \\ \Sigma M_{y(N1)} = m \cdot g \cdot L_{\varphi} - (N_2 + N_3 + N_4) \cdot L_r = 0 \\ N_2 = N_3 = N_4 \end{cases}$$

N_1 = 82404N = 8400kg

N_2 = N_3 = N_4 = 51012N = 5200kg

Truck calculations:

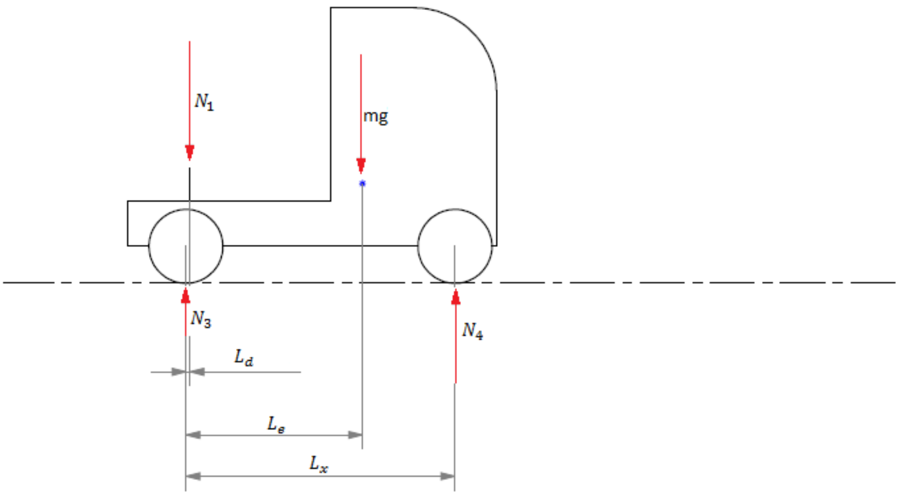


Figure 4 Dimensions of the truck.

To calculate the forces in the axles the vertical force sum and the sum of moments in the point where N_3 is applied will be used.

mg = 10000 · 9,81 = 98100N N_1 = 82404N

L_e= 2,8m L_x = 3,9m L_d = 0,06m

$$\begin{pmatrix} eq.\ 4 \\ eq.\ 5 \end{pmatrix} \quad \begin{cases} \Sigma F_x = N_3 + N_4 - m \cdot g - N_1 = 0 \\ \Sigma M_{y(N3)} - m \cdot g \cdot L_e - N_1 \cdot L_d + N_4 \cdot L_x = 0 \end{cases}$$

N_4 = 71698,5N = 7308,7kg N_3 = 108805,5N = 11091,3kg

As it can be seen the static analysis of the truck is correct. Any axle of trailer triaxle tandem hold more than 8000kg, the driven axle of the truck does not have to hold more than 11500kg and the other axles hold less than 10000kg. These values are not the maximum weight that the Spanish law allows and because of that it can be assumed that the design of the truck sizes and shape is the correct to fit the low.

4.2. Driving resistances

To start the determinations and design of the engine characteristics, all the forces that the engine will have to overcome to move and give acceleration to the set truck-trailer have to be known. All the vehicles that are circulating have four different kind of driving resistances. These resistances affect the force that the driving wheels need to do to move the truck.

There are four types of driving resistances: air resistance, rolling resistance, hill resistance and acceleration resistance.

The driving resistances will be calculated separately to know how they evolve.

4.2.1. Air resistance

The air resistance is a type of friction that refers to forces acting opposite to the relative motion of any object. This kind of friction is different of other friction forces, such as dry friction, which are nearly independent of velocity, air resistance forces depend on velocity. Air resistance forces have a square relation with the velocity for a turbulent flow and it is proportional to the velocity for a laminar flow. In the studied case it will only be considered the turbulent flow because the working velocities have high values.

Air resistance depends on the properties of the fluid and on the size, shape, and speed of the object. One way to express this is by means of the air resistance equation:

$$(eq.\ 6) \quad F_a = \frac{1}{2} \cdot \rho \cdot C_d \cdot A \cdot v^2$$

F_a is the air resistance force.

ρ is the density of the fluid.

v is the speed of the object relative to the fluid

A is the cross sectional area

C_d is the air resistance coefficient and is a dimensionless number.

Air resistance coefficient C_d is obtained by laboratory experiments. This coefficient is very depending on the shape and size of the vehicle. To have a low coefficient and less air resistance is important to do a design of the vehicle that reduces the surface with the boundary layer separated.

The studied vehicle is a combined vehicle. The aerodynamics of these kinds of vehicles is not easy to change. It will be considered that the air resistance coefficient of the truck is 0,9 and the air resistance coefficient of the trailer is the half of that, 0,45.

The air resistance that the truck will experiment during its movement has to be known.

The driving resistances are going to be calculated separately to know how they are evolving depending on the velocity of the vehicle. The calculations will start by the different values of the air resistance depending, as it has been said, on the velocity for the whole combined vehicle (figure 5).

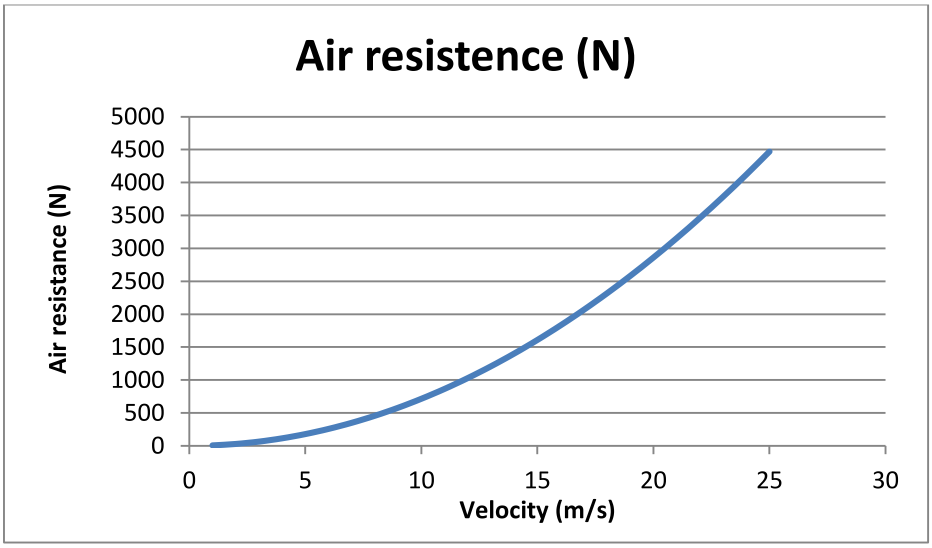


Figure 5 Air resistance depending on the velocity in road conditions.

As it was expected (because the equation is $F_a = C_d \cdot \frac{1}{2} \cdot \rho \cdot v^2 \cdot A$) the relation between the air resistance and the velocity is a square relation but to know if this relation is the correct it will checked using the following expression:

$$(eq.\ 7) \quad \frac{F_a(20)}{F_a(10)} = \left(\frac{v_{20}}{v_{10}}\right)^2$$

Air resistance force at the velocity of 10m/s: $F_a(10) = 715{,}2$ N

Air resistance force at the velocity of 20m/s: $F_a(20) = 2860{,}38$ N

$$(eq.\ 8) \qquad \frac{F_a(20)}{F_a(10)} = \left(\frac{v_{20}}{v_{10}}\right)^2 = \frac{2860{,}38}{715{,}2} = \left(\frac{20}{10}\right)^2 = 4$$

It is important to know the power that the engine will have to spend to beat the air resistances.

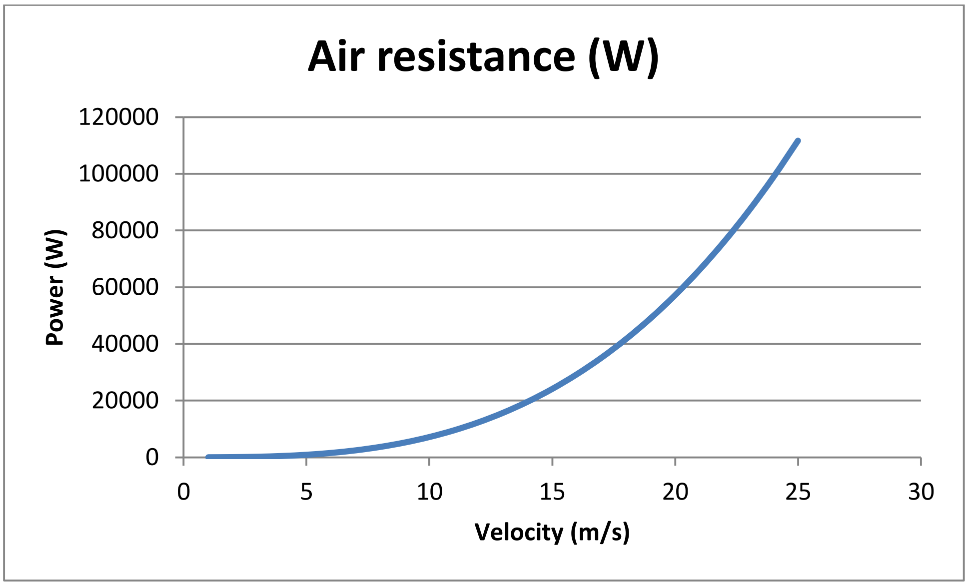


Figure 6 Representation of the power needed to beat the air resistances depending on the velocity.

Doing the same check as in the force of the air resistance, the relation between the velocity and the power must be cubic (because the equation of the force is $F_a = A \cdot C_d \cdot \frac{1}{2} \cdot \rho \cdot v^2$ and the equation of the power is $P = F \cdot v$).

$$(eq.\ 9) \qquad \frac{P_a(20)}{P_a(10)} = \left(\frac{v_{20}}{v_{10}}\right)^3$$

Air resistance power at the velocity of 10m/s: $P_a(10) = 7152$ W

Air resistance power at the velocity of 20m/s: $P_a(20) = 57207{,}6$ W

$$(eq.\ 10) \qquad \frac{P_a(20)}{P_a(10)} = \left(\frac{v_{20}}{v_{10}}\right)^3 = \frac{57207{,}6}{7152} = \left(\frac{20}{10}\right)^3 = 8$$

4.2.2. Rolling resistance

Rolling resistance is the force resisting the motion when a wheel rolls on a surface. It is mainly caused by non-elastic effects; that is, not all the energy needed for deformation of the wheel is recovered when the pressure is removed.

Rolling resistance is often expressed as a coefficient times the normal force. This coefficient of rolling resistance is generally much smaller than the coefficient of friction.

Any wheeled vehicle will gradually slow down due to rolling resistance including that of the bearings, but a train car with steel wheels running on steel rails will roll farther than a bus of the same mass with rubber tires running on tarmac. Factors that contribute to rolling resistance are the amount of deformation of the wheels, the deformation of the roadbed surface, and movement below the surface. Additional contributing factors include wheel diameter, speed, load on wheel, surface adhesion, sliding, and relative micro-sliding between the surfaces of contact.

The rolling coefficient depends on the velocity, on the pressure of the wheel and on the tire construction. The increase on this coefficient is directly proportional to the level of deformation and inversely proportional to the radius of the tire. This coefficient will increase in response to greater loads, higher speeds and lower tire pressure. In accelerations and breakings this coefficient is also incremented. For the nature of the equation that governs the coefficient it will be considered that this coefficient is constant under the speed of 150 km/h of the vehicle.

We can draw the rolling resistance in the following way:

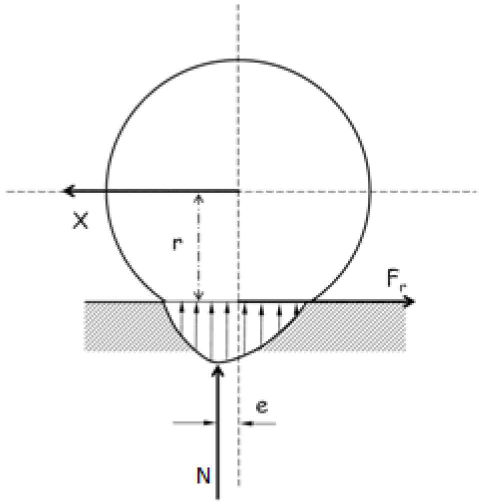


Figure 7 Rolling resistance scheme

If the sum of moments is done, without considering the torque, in the center of the wheel it is possible to find the friction coefficient.

$$(eq.\ \ 11) \qquad -F_r \cdot r + e \cdot N = 0$$

$$(eq.\ \ 12) \qquad \frac{F_r}{N} = \frac{e}{r} = C_{rr}$$

The friction coefficient that will be considered for the truck is 0,008. It is considered that all wheels have the same coefficient.

Then the different values of the rolling resistance depending on the velocity for the whole combined vehicle are going to be calculated (figure 8).

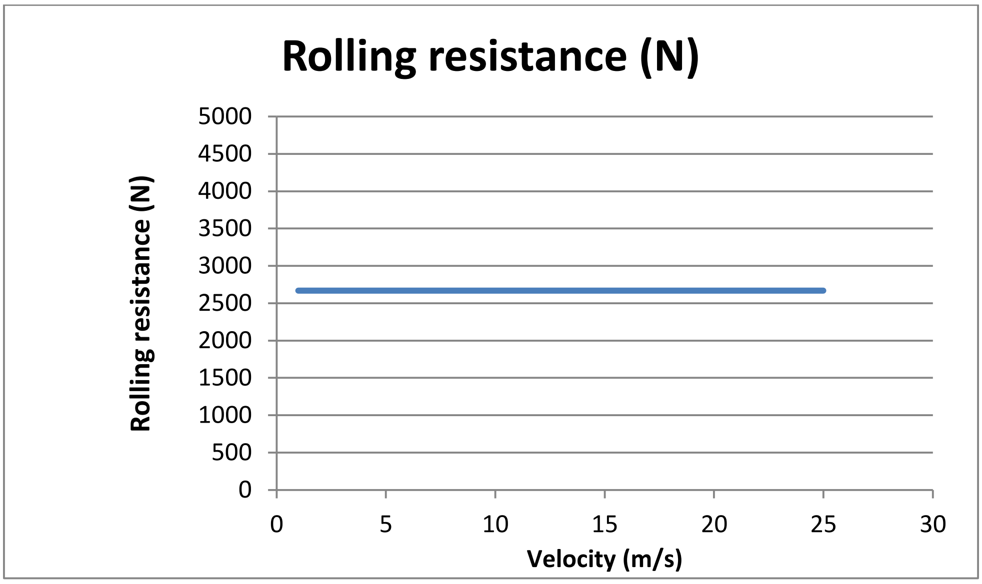


Figure 8 Rolling resistances depending on the velocity in road conditions.

As it can be seen, in this case (figure 8) the rolling resistance is not depending on the velocity but in practically cases it is inclined when velocity is increasing because the rolling resistance coefficient can depend on the velocity. Checking the rolling resistance equations it can be seen that this is correct because the rolling resistance forces only depend on the normal force of each wheel and on the rolling resistance coefficient that, as it has been said, is constant. As in this study has not been considered (because in a truck are negligible) the forces that the air does in the vertical axle the normal forces at wheels are always the same and the rolling resistances are always the same even the velocity changes. Because of that the power that the truck needs to beat the rolling resistances is proportional to the velocity (figure 9).

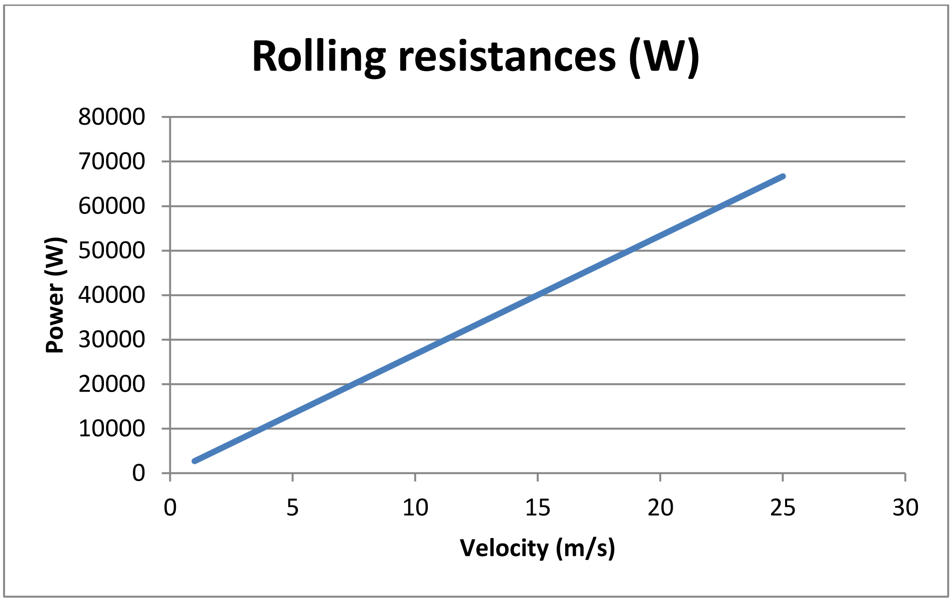


Figure 9 Representation of the power needed to beat the rolling resistances depending on the velocity.

To know the variation of the rolling resistance a graph will be built but now with the rolling resistances in function of the inclination. The inclination is the elevation gain for each 100 meters. It is expressed by %. The inclination of the figure 10 is P% of inclination.

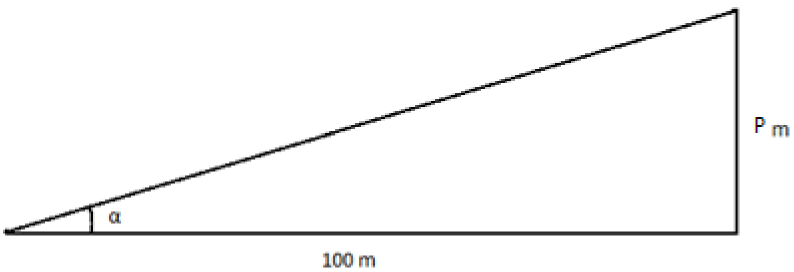


Figure 10 Draw of the inclination of the road.

The angle α is calculated by trigonometry:

$$(eq.\ \ 13) \qquad \tan\alpha = \frac{X[m]}{100[m]} = P\%$$

To do that graph, the sum of normal forces will be used, because the expression of the rolling resistances is directly related with it, in each inclination and the coefficient C_{rr} that the wheels of the truck have (0,008). The sum of these normal forces must have a similar evolution like a cosine because are only affected by the inclination of the road.

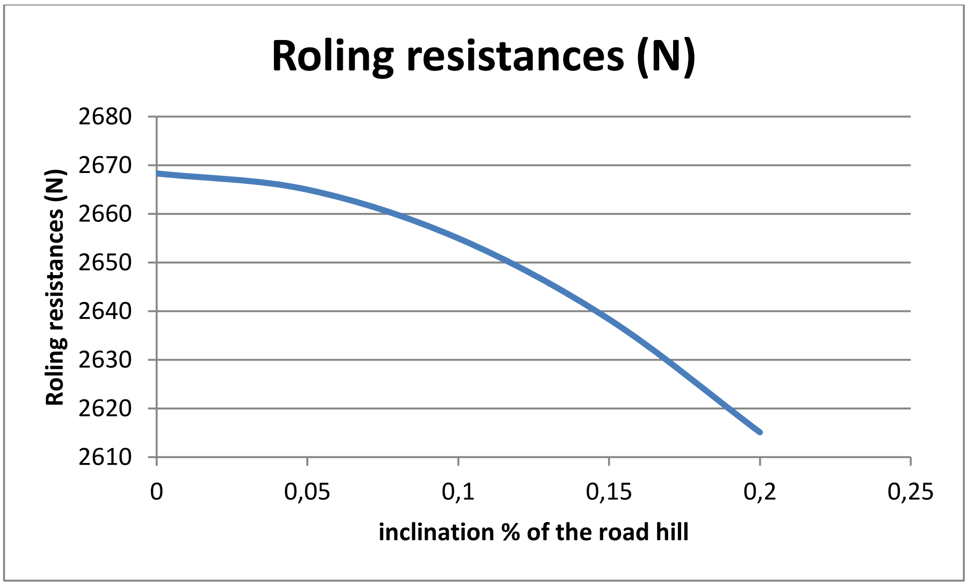


Figure 11 Rolling resistances depending on the inclination.

We can see in the figure 11 that the result is similar at the expected (cos α). That mean than the rolling resistance calculated is correct.

These results mean that when the road is gaining inclination the resistance that the spinning of wheels do is decreasing. But in the other side the component of the weight that the engine has to pull is increasing with the inclination (figure 13) with a sinus shape.

4.2.3. Hill resistance

The hill resistances only appear when the vehicle is moving in inclined roads. These kind of driving resistances are not depending on the velocity and other road conditions. The resistance is the weight component of the truck that because of the inclination of the road has a horizontal component and it has to be hold by the torque that the driving wheels do, instead of being hold by the normal force.

The equation of this hill resistance is:

$$(eq.\ \ 14) \qquad F_h = m_{set} \cdot g \cdot \sin \alpha$$

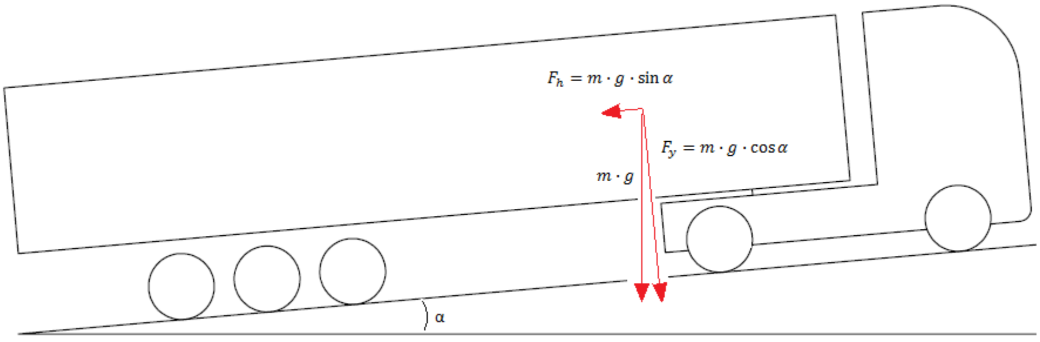


Figure 12 Components of the weight depending on the inclination of the road.

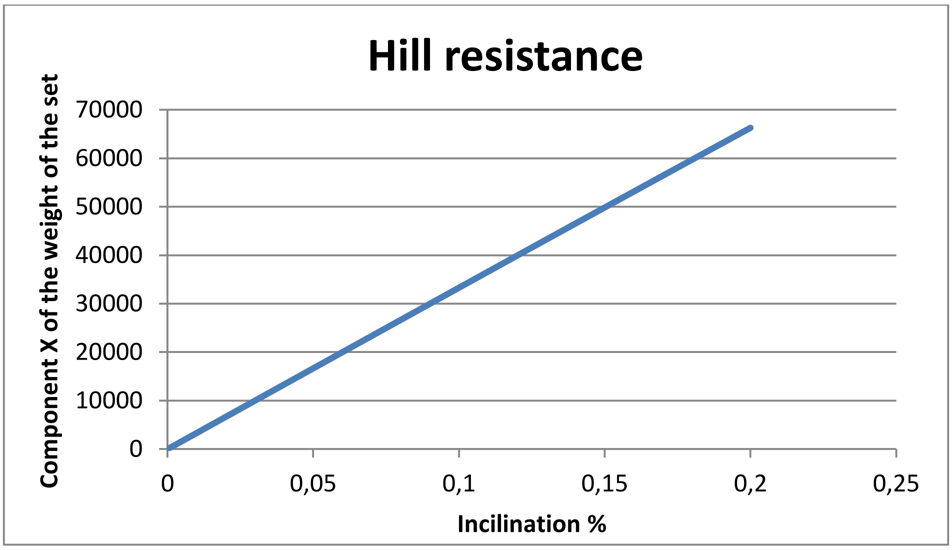


Figure 13 Represtentation of the of the hill resistances depending on the inclination of the road.

In the figure 13 is not easy to see but it can be seen that the representation of the hill resistance has the shape of a sinus. If the graph was draw until a bigger angle the sinus shape could be seen.

4.2.4. Acceleration resistance

This resistance appears when the vehicle is accelerating. The resistance is explained by the Second Law of Newton, it is the force that the wheels need to do to give the desired acceleration to the set truck-trailer. This resistance is the mass of the vehicle (mass of the truck and mass of the trailer) multiplied by the correspondent acceleration.

Applying the second Newton's second law he equation of this driving resistance is:

$$F = m \cdot a$$

$$F - m \cdot a = 0$$

$$(eq.\ \ 15) \qquad F_{ac} = m_{set} \cdot a$$

4.3. Driving resistances values

Knowing how the four resistances evolve separately depending on the different values of the velocity inclination and acceleration, it is necessary to calculate all the driving resistances together. Doing this it will be easy to know which is the engine that can run better the truck without significant torque and power lacks.

To do this calculations all the equations that will be used in the model of the trailer and the model of the truck have to be defined. These equations will consist in sum of forces, sum of moments and the equations of the driving resistances.

Once the equations will be together it will be possible to calculate all the driving situations with the correspondent driving resistances, the force and torque that the driving wheels will have to do and the power that the engine will need in each situation.

4.3.1. The trailer model

To calculate the trailer driving resistances and forces it is necessary to consider, as in the static calculations, that the three normal forces of every axle have the same value each other.

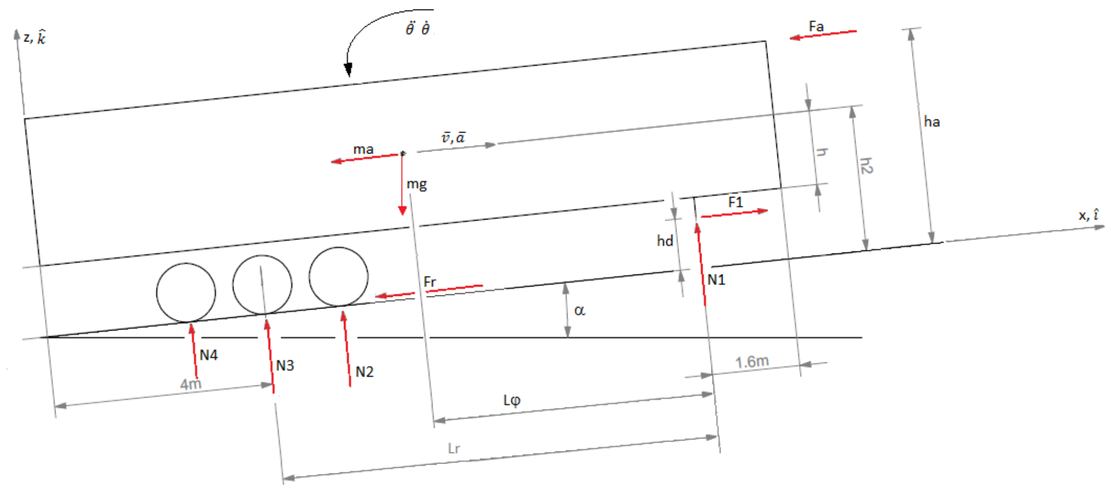


Figure 14 Forces of the trailer in road conditions.

The trailer forces that are in the illustration (figure 14) can be calculated with the following equations: sum of horizontal forces, sum of vertical forces, sum of moments about the point where N_1 crosses the floor, the specific equations of rolling resistance and air resistance and the relation between the normal forces of each axle.

$$(eq.\ sys.\ 1) \quad \begin{cases} \Sigma F_x = ma \\ \Sigma F_z = ma \\ \Sigma M_{N1} = I\ddot{\theta} \\ F_r = C_{rr} \cdot N \\ G = m \cdot g \\ F_a = C_d \cdot \frac{1}{2} \cdot \rho \cdot v^2 \cdot A \\ \frac{m_2}{m_b} \approx \frac{m_3}{m_b} \approx \frac{m_4}{m_b} \end{cases}$$

Breaking down the sum of forces and sum of moment equations these components appears:

$(eq.\ sys.\ 2)$

$$\begin{cases} \Sigma F_x = -G \cdot \sin\alpha - F_r + F_1 - F_a - m \cdot a = 0 \\ \Sigma F_z = N_1 + N_2 + N_3 + N_4 - G \cdot \cos\alpha = 0 \\ \Sigma M_{y(N1)} = -(N_2 + N_3 + N_4) \cdot L_r - F_1 \cdot h_d + F_a \cdot h_a + m \cdot a \cdot h_2 + \\ + G \cdot \sin\alpha \cdot h_2 + G \cdot \cos\alpha \cdot L_\varphi = 0 \end{cases}$$

Then the rest of equations are added and it can be simplified. And finally a system of 5 equations with 5 variables is done and it can be solved.

$$(eq.\ sys.\ 3)\quad \begin{cases} -G\cdot\sin\alpha - F_r + F_1 - F_a - m\cdot a = 0 \\ N_1 + N_2 + N_3 + N_4 - G\cdot\cos\alpha = 0 \\ \dfrac{N_2+N_3+N_4}{G} = \varphi_x\cdot\cos\alpha + \varphi\cdot\sin\alpha + \varphi\cdot z + k_a\cdot\varphi_a - \dfrac{F_1}{G}\cdot\varphi_d \\ F_r = C_{rr}\cdot(N_1+N_2+N_3) \\ F_a = C_d\cdot\dfrac{1}{2}\cdot\rho\cdot v^2\cdot A \end{cases}$$

The parameters used in the third equation are the following parameters:

$$(eq.\ sys.\ 4)\quad \begin{cases} \varphi_x = \dfrac{m_2+m_3+m_4}{m} = \dfrac{L_\varphi}{L_r} \\ \varphi = \dfrac{h_2}{L_\varphi} \\ z = \dfrac{a}{g} \\ k_a = \dfrac{F_a}{G} \\ \varphi_a = \dfrac{h_a}{L_r} \\ \varphi_d = \dfrac{h_d}{L_r} \end{cases}$$

4.3.2. The truck model

To calculate the truck driving resistances and forces it is necessary to consider the calculated trailer forces to know which value have the forces F_1 and N_1 (figure 15).

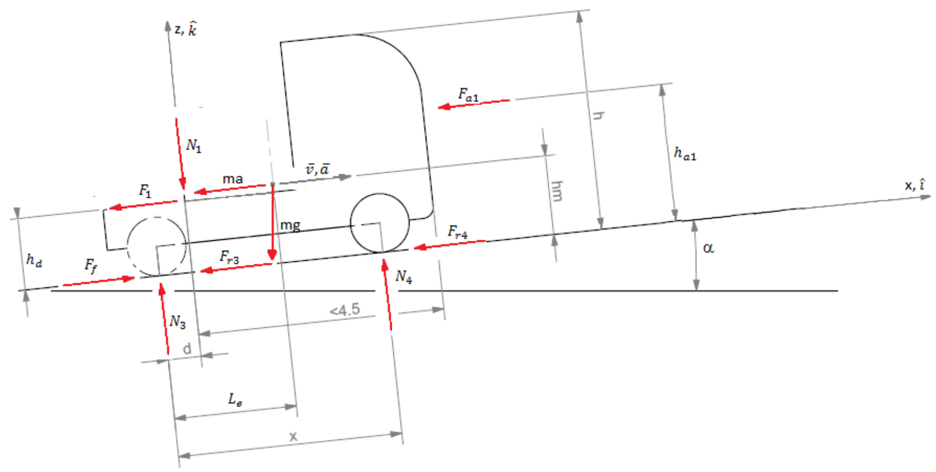


Figure 15 Forces of the truck in road conditions.

The truck forces can be calculated using the sum of horizontal forces, the sum of vertical forces, the sum of moments about the point where the rear wheel of the truck touches the ground and the equations of rolling resistance and air resistance.

$$(eq.\ sys.\ 5) \quad \begin{cases} \Sigma F_x = ma \\ \Sigma F_y = ma \\ \Sigma M_G = I\ddot{\theta} \\ F_r = C_{rr} \cdot N \\ F_a = A \cdot C_d \cdot \dfrac{1}{2} \cdot \rho \cdot v^2 \end{cases}$$

To calculate the moment equation it must be considered the reaction torque that the wheel gives to the truck. The force that needs the truck to be moved is the same force that the friction does. So the torque that the wheel must have to move the truck is:

$$(eq.\ \ 16) \qquad \Sigma M_{ICR} = I\ddot{\theta}$$

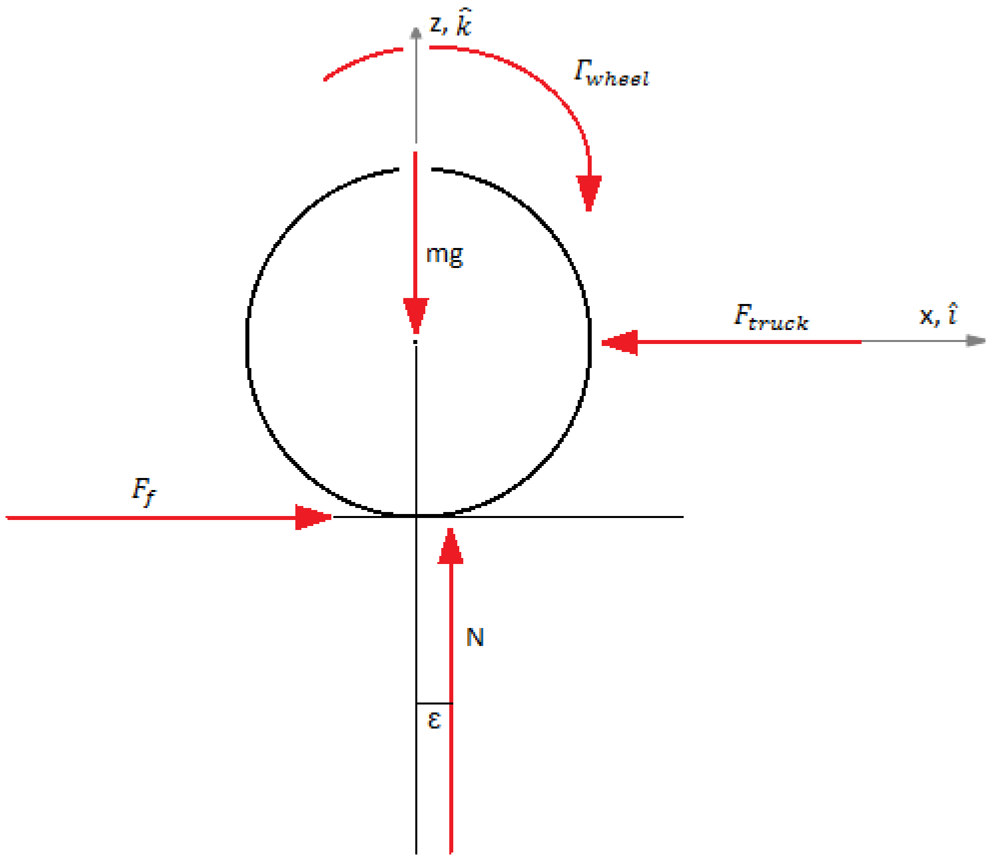


Figure 16 Forces in the driving wheels.

In the figure 16 it is possible to see how these forces and torques are distributed on the wheel.

In practical circulation is assumed $I\ddot{\theta} \approx 0$ and $\varepsilon \approx 0$ then:

$$(eq.\ 17) \qquad \Sigma M_{ICR} = F_{truck} \cdot r - \Gamma_{wheel} = 0$$

$$(eq.\ 18) \qquad \Gamma_{wheel} = F_{truck} \cdot r = F_f \cdot r$$

The force equations of the truck model are:

$(eq.\ sys.\ 6)$

$$\begin{cases} -m\cdot g\cdot \sin\alpha + F_f - F_{a1} - F_{r3} - F_{r4} - F_1 = m\cdot a \\ N_3 + N_4 - m\cdot g\cdot \cos\alpha - N_1 = 0 \\ N_4 + F_{a1}\cdot \varphi_{a1} + F_1\cdot \varphi_d + m\cdot g\cdot \sin\alpha\cdot \varphi_m - N_1\cdot \varphi_{d2} - m\cdot g\cdot \cos\alpha\cdot \varphi_e + F_f\cdot \varphi_r = 0 \\ F_{r3} = N_3\cdot C_{rr} \\ F_{r4} = N_4\cdot C_{rr} \\ F_{a1} = \frac{1}{2}\cdot C_d\cdot A\cdot \rho\cdot v^2 \end{cases}$$

The parameters that conform the sum of moment equation are:

$$(eq.\ sys.\ 7) \quad \begin{cases} \varphi_{a1} = \frac{h_{a1}}{x} \\ \varphi_d = \frac{h_d}{x} \\ \varphi_m = \frac{h_m}{x} \\ \varphi_{d2} = \frac{L_d}{x} \\ \varphi_e = \frac{L_e}{x} \\ \varphi_r = \frac{r}{x} \end{cases}$$

With the results of the trailer equations and solving this system too, it is possible to find the friction force that the truck needs to advance therefore the force that the drive wheels should do. This forces will bring us the wheel torque and with the gear reductions it is possible to determinate the engine torque.

With the wheel torque and the angular velocity of the wheel or with the friction force and the lineal velocity of the truck is not difficult to find how much power needs the truck to run in the previous conditions.

4.3.3. Hypothetic case

To explain how the system of equations and how all the general driving resistances will be calculated here there is an example of a particular situation with the steps explained.

A hypothetic road situation will be presented and the forces and torques involved on it are going to be calculated.

To calculate this situation a 5 axle set truck-trailer will be taken. The trailer is a 3 axle trailer, its weight is 24 tones and its center of gravity is on the middle of its length. The truck is a 2 axle tractor, its weight is 10 tones and its frontal area is 9,8m^2 (see figure 17). The tractor drive wheels are located on its rear axle (see figure 17).

The union system between truck and trailer is located 1,3m from the floor and it is a *fifth wheel* system. This hitch system consists of a kingpin, a 50.8 or 88.9 mm diameter steel pin on the front of the trailer, and a horseshoe-shaped coupling device called a *fifth wheel* on the rear of the tractor vehicle. The surface of the trailer hitch part (with the kingpin at the center) rotates against the surface of the fixed fifth wheel, which does not rotate. To reduce friction, grease is applied to the surface of the fifth wheel. This system provides more stability to the trailer than other coupling systems.

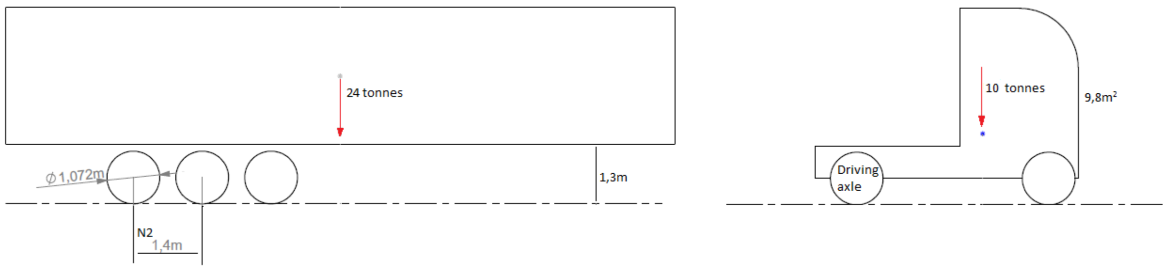


Figure 17 Masses and coupling of the set truck-trailer.

The situation that will be studied is the set, truck-trailer, circulating in a road with a certain inclination. As driving resistances, apart from the hill resistances and acceleration resistances, the air resistance and the rolling resistance will be considered.

As the vehicle is a combined vehicle, the parts are going to be calculated separately: first the force that the trailer needs to be moved in some conditions is going to be calculated and then, with the same conditions, the force that the driving wheels need to do to move the trailer and the truck will be calculated.

For example, the case that is calculated is the truck going full (weight of the trailer 24000kg and weight of the truck 10000kg) and circulating on a road with an inclination of 5% at the velocity of 54 km/h without acceleration.

The first things that have to be calculated, using the equations showed in the point 4.3.1. and the known variables, are all the results of the unknown variables of the trailer.

Trailer variables:

C_{rr}=0,008 A=6,885m^2 a=0 h=1,2m h_2=2,5m m=24000kg

α=0,05rad C_d=0,45 z=0 ϕ=0,481 ϕ_a=0,5 ϕ_d=0,144

ϕ_x=0,65 v=15 m/s

$$(eq.\ sys.\ 8)\quad \begin{cases} -G\cdot\sin\alpha - F_r + F_1 - F_a - m\cdot a = 0 \\ N_1 + N_2 + N_3 + N_4 - G\cdot\cos\alpha = 0 \\ \dfrac{N_2+N_3+N_4}{G} = \varphi_x\cdot\cos\alpha + \varphi\cdot\sin\alpha + \varphi\cdot z + k_a\cdot\varphi_a - \dfrac{F_1}{G}\cdot\varphi_d \\ F_r = C_{rr}\cdot(N_1+N_2+N_3) \\ F_a = C_d\cdot\dfrac{1}{2}\cdot\rho\cdot v^2\cdot A \end{cases}$$

Replacing the variables for the correspondent values, the forces that the trailer will need to do to be moved can be known and the normal forces that the wheels will have to hold too.

Results:

F_1=13439,6N F_r=1254,2N N_2=156779,2N **N_1=78366,6N**

F_a=418,3N k_a=0,0017765

Then, using the results of the trailer calculations, the known variables of the truck and the equations shown at the point 4.3.2., it is possible to calculate the truck unknown variables and know which force and power the engine need to move the truck.

Truck variables:

$A=9{,}8m^2$ $C_d=0{,}9$ $C_{rr}=0{,}008$ $F_1=13439{,}6N$ $N_1=78366{,}6N$

$L_d=0{,}6m$ $h_{a1}=2m$ $x=3{,}9m$ $a=0$ $h_d=1{,}15m$ $h_m=1{,}3m$

$m=10000kg$ $r=0{,}536m$ $v=15m/s$ $\alpha=0{,}05rad$ $L_e=2{,}9m$ $\phi_{a1}=0{,}5128$

$\phi_d=0{,}333$ $\phi_m=0{,}31$ $\phi_{d2}=0{,}015$ $\phi_e=0{,}5128$ $\phi_r=0{,}137$

$(eq.\ sys.\ 9)$

$$\begin{cases} -m\cdot g\cdot\sin\alpha+F_f-F_{a1}-F_{r3}-F_{r4}-F_1=m\cdot a \\ N_3+N_4-m\cdot g\cdot\cos\alpha-N_1=0 \\ N_4+F_{a1}\cdot\varphi_{a1}+F_1\cdot\varphi_d+m\cdot g\cdot\sin\alpha\cdot\varphi_m-N_1\cdot\varphi_{d2}-m\cdot g\cdot\cos\alpha\cdot\varphi_e+F_f\cdot\varphi_r=0 \\ F_{r3}=N_3\cdot C_{rr} \\ F_{r4}=N_4\cdot C_{rr} \\ F_{a1}=\dfrac{1}{2}\cdot C_d\cdot A\cdot\rho\cdot v^2 \end{cases}$$

Knowing the forces of the trailer and the values of the parameters of the trailer, it is possible to calculate the torque that the driving wheels will have to do to move the truck and the trailer in those specific conditions.

Results:

$F_{r3}=804{,}15N$ $F_{r4}=606{,}6N$ $\mathbf{F_f=20944N}$ $F_v=1190{,}7N$

$N_3=100518{,}9N$ $N_4=75825{,}03N$

As it can be seen, these results say that to move the truck, in these conditions, the driving wheels will have to do a total force of 20944N whence it will be necessary to search a combination of an engine and a gearbox that can assume these conditions and give to the driving wheels these forces.

4.3.4. General case

If the range of forces and power that the driving wheels and the engine will have to do to move the combined vehicle needs to be known better, it will be necessary to determinate the four different driving resistances together.

Firstly the rolling resistance and the air resistance are going to be summed (figure 18). Then the other two driving resistances, hill resistance and acceleration resistance, only will increase the force values of the resistances but the dependence on the velocity will be the same.

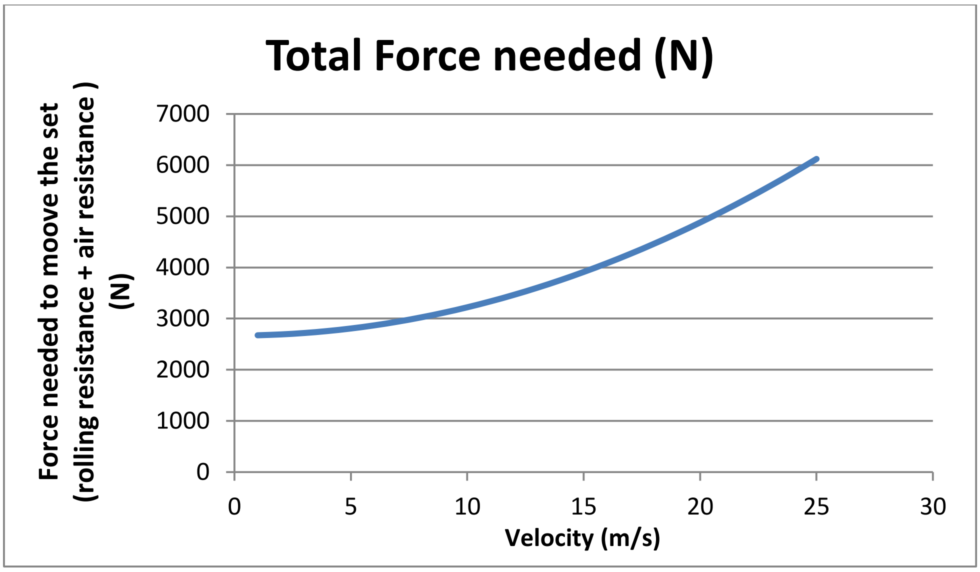


Figure 18 Total force needed to move the truck with an inclination of 0%.

To know the real effort that the engine will have to do (power and force) is necessary to sum all the driving resistances and the component of the weight that the truck must pull. Having put all together there is a graph that says the force that the wheels need to move the truck in a certain velocity and in a certain inclination of the road.

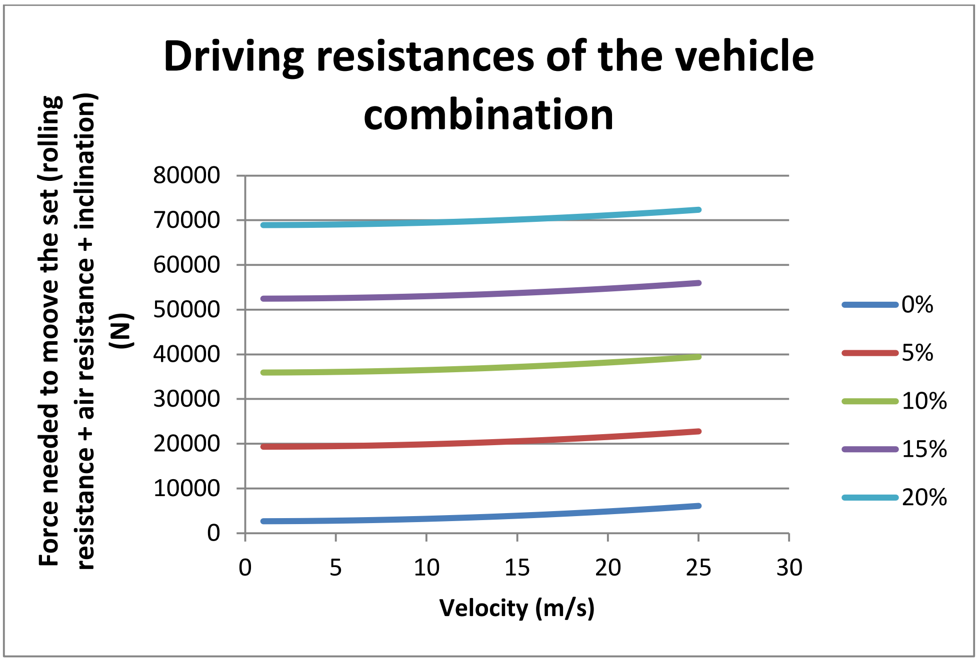


Figure 19 Total driving resistances (rolling resistance, air resistance and hill resistance) depending on the velocity.

Looking at the graph (figure 19) it is possible to see that the most relevant parameter is the inclination of the road (the different color lines). When this parameter increases the force that the truck needs to do to be moved increases significantly. On the other hand, even the air resistance is square related to de velocity (v^2), it can be considered, when the inclination is not 0%, that it is not much significant.

In the shown graph (figure 19) the representation of the acceleration driving resistance is not drawn. The acceleration in this graph is 0. This is because of the difficulty of drawing a graph with 4 variables. But as it has been said the gaining acceleration will increase the total diving resistances proportionally to the acceleration.

Now is possible to know the power that the wheel of the truck needs to move the vehicle in different velocities. As is known the equation of the power is: $P = F \cdot v$. The product between the previous graph (figure 19) and the velocity will show the power needed to beat that resistances.

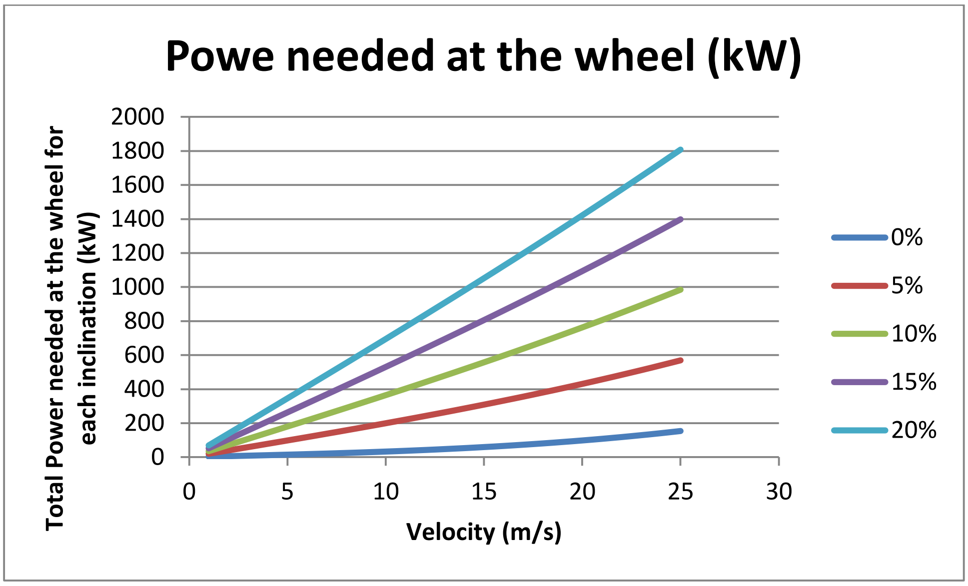


Figure 20 Total power needed at the driving wheels in road conditions.

Here (figure 20) is possible to see the influence that the inclination (every different color) and the velocity have on the force and the power needed to move the truck. In this power graph is highlighted that the velocity and the inclination have an important influence in the power needed to move the truck in the different situations. The relation between power and velocity is cubic (v^3). By this fact, is possible that the engine will have some power limitations when it will run at high velocity and in some roads with important inclinations. To solve these limitations and to keep circulating the truck will only need to reduce its velocity and in consequence the engine will spend less power to move.

4.4. Engine

Once the range of power and torque that the set will need to be moved in normal roads is bounded it is possible to try to find the engine that will allow the truck circulate without power problems in the majority of situations. It is true that the best engine would be the most powerful with the biggest force and torque but the point is finding a useful and real engine for the truck.

Continuing with the example of the point 4.3.3. it will be possible to find the power needed in the presented situation.

As it is known the power expression is: $P = F \cdot v$ and continuing with the previous calculations, the force has the following value: F = F_f = 20944N and velocity is 15m/s. It is easy to calculate the power needed to stay in this situation.

$$(eq.\ \ 19)\quad P_w = F \cdot v = F_f \cdot v = 20944 \cdot 15 = 314159{,}85W = 314{,}16kW = 428\ hp$$

This is the power that the wheels will need to move the set truck-trailer in this situation, but it is known that the transmission has a relevant loss of energy ($\eta_{trans} \approx 0{,}85$). For this reason it is necessary to calculate the power needed to the engine as:

$$(eq.\ \ 20)\quad P_{engine} = P_w + P_{loss}$$

$$(eq.\ \ 21)\quad P_{loss} = P_w \cdot \left(\frac{1}{\eta_{trans}} - 1\right)$$

$$\eta_{trans} \approx 0{,}85$$

$$(eq.\ \ 22)\quad P_{engine} = P_w + P_{loss} = P_w + P_w \cdot \left(\frac{1}{\eta_{trans}} - 1\right) = \frac{P_w}{\eta_{trans}} = \frac{314.16}{0{,}85} = 369{,}6\ kW = 504\ hp$$

These results show that the truck needs a powerful engine but not out of normality. Most truck companies are doing this kind of engines for tractors. The set truck-trailer is a heavy truck and because of that it will need to use a powerful engine to be moved.

To know all the parameters that need to be considered when the engine has to be chosen, it is necessary to know which torque must provide the engine to the wheel to move the set.

The expression of the torque, considering that the wheel has a diameter of 1,072m (see point 4.6.), is the following:

$$(eq.\ \ 23)\quad \Gamma_{wheel} = F \cdot r = F_f \cdot r = 20944 \cdot 0{,}536 = 11225{,}98Nm$$

This torque is the engine torque increased by the gear box velocity reductions. To know the exactly engine torque the procedure would be finding the reduction relation of the gearbox and the loss of energy that the gearbox and all the transmission system have.

$$(eq.\ 24) \qquad \eta_{trans} = \eta_i \cdot \eta_{kar} \cdot \eta_o \cdot \eta_p \cdot \eta_{clutch}$$

$$(eq.\ 25) \qquad \Gamma_{wheel} = \eta_{trans} \cdot i_{total} \cdot \Gamma_{engine}$$

$$(eq.\ 26) \qquad F_f = \frac{\Gamma_{wheel}}{r} = \eta_{trans} \cdot i_{total} \cdot \frac{\Gamma_{engine}}{r}$$

$$(eq.\ 27) \qquad \Gamma_{engine} = \frac{F_f \cdot r}{\eta_{trans} \cdot i_{total}}$$

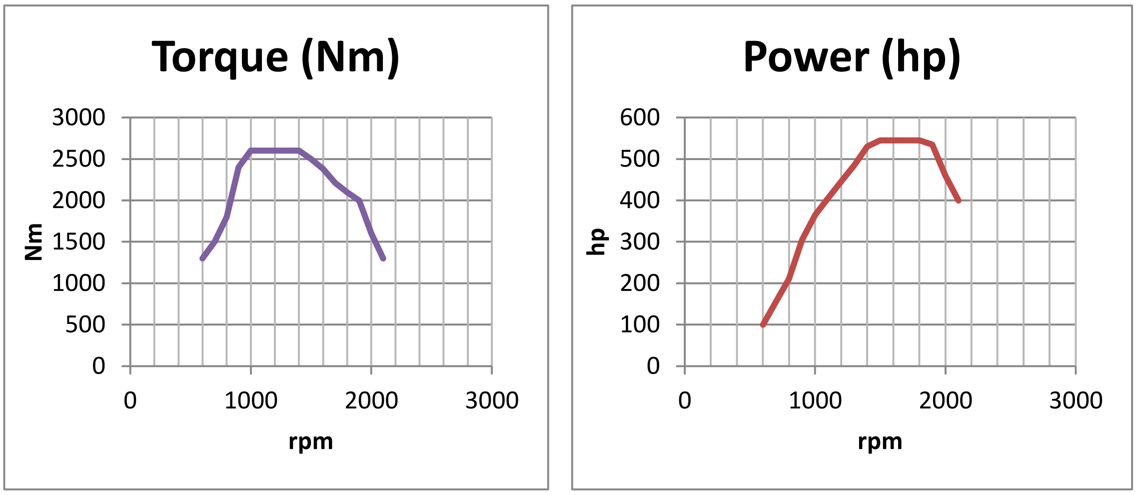
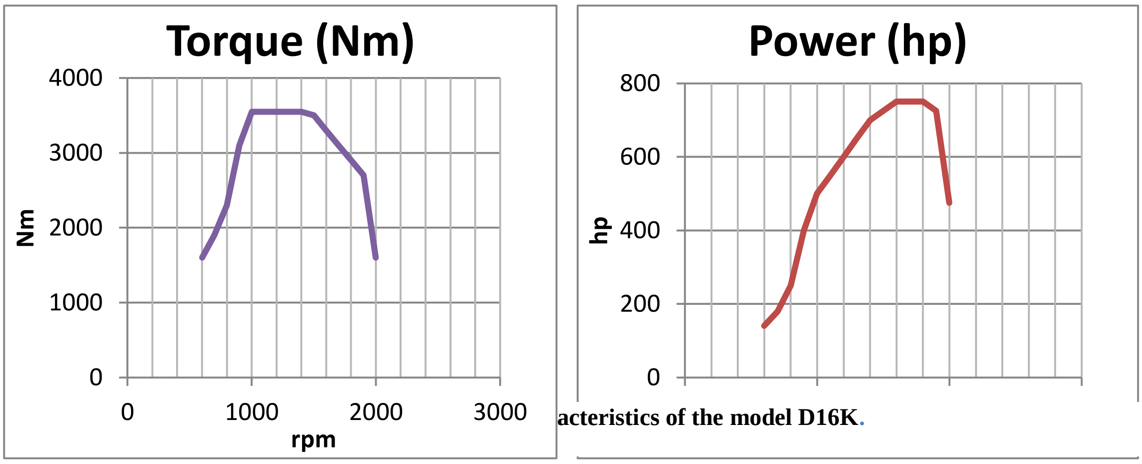


Figure 21 Engine characteristics of the model D13K.

An example of a good engine for our truck it would be the Volvo FH D13K or Volvo FH D16K. These two engines can provide the power needed at the wheel and a wide rpm interval with the maximum torque and the maximum power. Here there are the map of the power and torque that these two engines can give.



acteristics of the model D16K.

D13K engine (figure 21) has its maximum torque between 950rpm and 1500 rpm and it is 2600Nm. The maximum power of this engine is between 1450rpm and 1900rpm and it is 540hp (397kW).

D16K engine (figure 22) has its maximum torque between 950rpm and 1450 rpm and it is 3550Nm. The maximum power of this engine is between 1600rpm and 1600rpm and it is 750hp (551kW).

Following the hypothetic case presented in the point 4.3.3. if the *RTS2370A Gearbox* ,build for *Volvo,* is taken as a gearbox, it is easy to see that to run at the velocity of 54 km/h (29,7 rad/s wheel spinning) with an appropriate engine revolutions the gear that give a ratio of 4,13/1 should be taken. That means if the wheel is spinning at 29,7 rad/s, the engine will do it at 122,661 rad/s (1171,32 rpm). And if 11226 Nm are needed at the wheel, with this reduction, at the engine is going to be required a torque higher than 2718,15 Nm because of the loss of energy in all transmission system. If it is considered that the loss of torque in the transmission is about 15% ($\eta_{trans} \approx 0{,}85$) the engine will have to do a torque of 3200Nm.

It is important to take note that with other road inclinations or other velocities the torque on the wheel will easily increase and the power needed to. The truck will must have to reduce its velocity reducing gears, therefore having a bigger gear box relation, and increasing the difference between engine torque and wheel torque.

4.4.1. Minimization of the pollution in diesel engines

Nowadays the problem of the environmental contamination is a topic that affects all the society. Diesel engines have an excellent reputation for their low fuel consumption, reliability, and durability characteristics. They are also known for their extremely low hydrocarbon and carbon monoxide emissions. However, they have also been rejected by many for their odorous and sooty exhaust that is also characterized with high nitric oxide and particulate matter emissions.

4.4.1.1. Legislation

The pollutant emissions from road vehicles in Europe are regulated separately for light-duty vehicles (cars and light vans) and for heavy-duty vehicles (trucks and buses). Now the standard that regulates the values of the exhaust pollution on the vehicles is the standard

EURO VI. This standard says the exhaust emission limit values of different kind of molecules (figure 23). Moreover the manufacturers should demonstrate that engines comply with that emission limit values for useful life periods which depend on the vehicle category (figure 24).

Table 1. Exhaust emission limit values (European Commission, 2015)

CO	HC	NO_x	PM	PN
		g/kWh		*1/kWh*
1,5	0,13	0,40	0,01	$8x10^{11}$

Figure 23 Exhaust emission limit values for different kind of molecules.

Table 2. Exhaust limit values for useful life (European Commission, 2015)

Vehicle category	Euro VI
N1 and M2	160 000 km / 5 years
N2 **N3 ≤ 16 ton** **M3 Class I, Class II, Class A, and Class B ≤ 7.5 ton**	300 000 km / 6 years
N3 > 16 ton **M3 Class III, and Class B > 7.5 ton**	700 000 km / 7 years

Figure 24 Emission limit values for useful life periods depending on the vehicle category.

4.4.1.2. Systems to reduce the pollution

(Prof. Rr.-Ing Konrad Rief, Dipl--Ing. Karl-Heinz Dietsche, 2014)

(Automotive Handbook, 2014)

Since performance, fuel consumption, and emitted pollutants result from the combustion process, it is necessary first to understand the mechanisms of combustion in diesel engines.

To perform the diesel combustion, oxygen must be made available to the fuel in a specific manner to facilitate it. One of the most important aspects of this process is the mixing of fuel and air, which is a process often referred to as mixture preparation.

In diesel engines, the liquid fuel is usually injected at high velocity as one or more jets through small holes in the injector tip. It atomizes into small droplets and penetrates into the combustion chamber. The atomized fuel absorbs heat from the surrounding heated compressed air, vaporizes, and mixes with the surrounding high-temperature high-pressure air.

As the piston continues to move closer to top dead center (TDC), the mixture (mostly air) temperature reaches the fuel's ignition temperature. Instantaneous ignition of some premixed fuel and air occurs after the ignition delay period. This instantaneous ignition is considered the start of combustion. Increased pressure resulting from the premixed combustion compresses and heats the unburned portion of the charge and shortens the delay before its ignition. It also increases the evaporation rate of the remaining fuel. Atomization, vaporization, fuel vapor-air mixing, and combustion continue until all the injected fuel has combusted.

A rapid combustion close to the TDC is a good factor to rise the efficiency of the engine, but this fact is also associated with a local peak combustion temperatures, which results in the formulation of nitrogen oxides (NO_x). On the other hand, the majority of the air inducted into the cylinder of a diesel engine is compressed and heated, but never engages in the combustion process. Oxygen in the excess air helps oxidize gaseous hydrocarbons and carbon monoxide, reducing them to extremely small concentrations in the exhaust gas.

To reduce the peak of temperatures that a rapid combustion gives, the engine can be designed with Exhaust-air recirculation (EGR). This system lowers the peak combustion temperature though the higher amount of gas and the slower rate of combustion, thereby reducing the amount of nitrogen oxides (NO_x). With this system it is possible to reduce the NO_x gases without rising disproportionately the emission of particles.

Another way to reduce the pollution is treating the exhaust-gas. In diesel engines with excess air the three-way catalytic converter can be used to reduce HC, CO and NO_x. HC and CO can be easily reduced by oxidation reactions. The reduction of NO_x in the presence of oxygen is more complicated, the way to do it is with an NO_x accumulator-type catalytic converter or and Selective Catalytic Reduction (SCR)

4.4.1.2.1. Reduction of HC and CO

An effective way to reduce the emission of HC and CO is using a Diesel oxidation-type catalytic converter (DOC). This mechanism consists of a ceramic substrate structure, an oxide mixture and catalytically active metals: platinum, palladium and rhodium.

The DOC process performs different functions: oxide the CO and the HC to CO_2 and H_2O and oxides NO to NO_2 thing that is important to reduce the NO_x with the NSC or SCR process.

4.4.1.2.2. Reduction of NO_x

There are two systems to reduce the NO_x emissions. The first one is the NSC (accumulator-type catalytic converter) and the second one is the SCR (selective, catalytic reduction).

NSC:

The NSC system only reduces the NO_2 but no the NO, for that reason is important to do the DOC treatment before doing the NSC.

First of all the NO_2 is stored and reacts with the components of the catalyst face and oxygen and the NO_2 is reduced directly to N_2.

SCR:

Selective Catalytic Reduction is an advanced active emissions control technology system that injects a liquid-reductant agent through a special catalyst into the exhaust stream. The reductant source is usually automotive-grade urea, otherwise known as Diesel Exhaust Fluid (DEF). The DEF sets off a chemical reaction that converts nitrogen oxides into nitrogen, water and tiny amounts of carbon dioxide (CO2).

This system works reducing certain nitrogen oxides in the presence of oxygen using reducing agents that prefer to oxidize selectively with the oxygen contained in the nitrogen oxides instead of with the molecular oxygen present in the exhaust gas. The actual reducing agent is the non-toxic catalyst called urea [$(NH_2)_2CO$] and is used solved with water.

Urea first forms ammonia before the SCR reaction starts:

$(NH_2)_2CO \rightarrow NH_3 + HNCO$

$HNCO + H_2O \rightarrow NH_3 + CO_2$

The ammonia produced reacts in the SCR catalytic converter:

$4NO + 4NH_3 + O_2 \rightarrow 4N_2 + 6H_2O$

$NO + NO_2 + 2NH_3 \rightarrow 2N_2 + 3H_2O$

$6NO_2 + 8NH_3 \rightarrow 7N_2 + 12H_2O$

If more reducing agent is dispensed than the necessary it may result in NH_3 leakage. That thing is avoidable placing an additional oxidation-type catalytic converter.

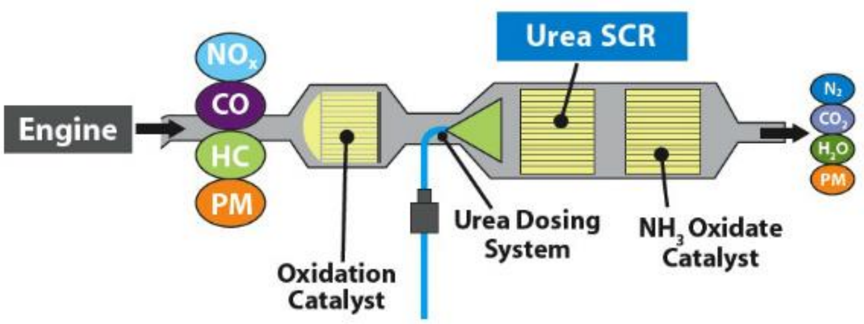


Figure 25 Scheme of an SCR system. (Diesel Technology Forum)

The system that regulates all the reactions is called DENOXTRONIC and its components are: the delivery module that brings the water-urea solution in the required pressure to the dossing module; the dosing module that implements the mixture with compressed air; the spraying pipe that atomizes and distributes the urea-water solution in the exhaust pipe and the control unit that regulates the dosing volume in accordance with the dosing strategy.

4.5. Transmission system

The truck needs to have the maximum torque that the engine can give to the wheel, especially in road velocities, to help it pass the inclinations without slowing down and to have more capability reaction against danger situations. This is possible designing a transmission system with a gearbox that enables change the gear when the torque, which the engine gives to the wheel, starts to decrease. To have this conditions is necessary a gearbox with a considerable number of gears.

In the transmission system is important to consider the different parts: the clutch (clutch), the gearbox (gear), the cardan (kar), the final gear (0) and the differential or planet gear (p). All of these parts have looses of power and torque.

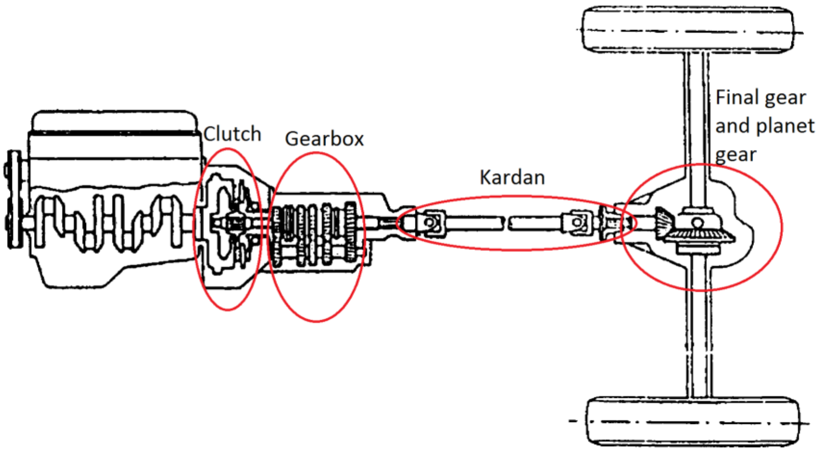


Figure 26 Scheme of the transmission.

The efficiency of the whole transmission is the product of the efficiency of each component and its expression is:

$$(eq.\ \ 28)\quad \eta_{trans} = \eta_{clutch} \cdot \eta_{gear} \cdot \eta_{kar} \cdot \eta_0 \cdot \eta_p$$

Then the torque that the engine will give to the truck is:

$$(eq.\ \ 29)\quad \Gamma_w = \eta_{trans} \cdot \Gamma_{engine} \cdot i_{gear} \cdot i_{kard} \cdot i_0 \cdot i_p$$

To know the real power and torque that will arrive at the wheel is necessary to do a study of each component.

4.5.1. Clutch

The clutch is an important part of every motor vehicle, but these kind of heavy vehicles need to have a well-designed clutch to avoid headings and posterior slipping.

The most advanced technology in clutch is the double clutch. This kind of clutch allows doing the change of the gears without losing power in any moment of it. During normal operation, a clutch freewheels while the other is coupled. The clutch and shifter are controlled by a pneumatic system. Basically, the dual clutch can preselect the next gear while driving in the current gear. When changing gear, the engaged clutch is released at the time while the other clutch that was previously free is engaged. This implies that the shift is performed without interruption in the power output.

This system of clutch is controlled by software. There is different kind of software depending on the characteristics of the terrain or the road and the use that the set truck-trailer will have. With this kind of clutch the gearbox spend almost no time doing the gearshift. The estimated time is 0.08s.

The efficiency of this kind of clutch is $\eta_{clutch} = 0{,}95$

4.5.2. Gearbox

After trying different number of gear gearboxes, doing the correspondent hypothesis and studies of each one it has been concluded that the best option and a good one to fit in both studied engines is a 13 gear gearbox. With that combination of gears that the gearbox offers is possible to have in all the range of velocities the maximum torque that the engine can give. With the help of the shift software the truck will always has the maximum torque at the driving wheels.

The relation of transmission of the gears is the following relation:

Table 3. Gears with their gearratios.

Gear	Relation	Gear	Relation
1	15,67:1	**7**	1,92:1
2	7,46:1	**8**	1,68:1
3	4,48:1	**9**	1,5:1
4	3,36:1	**10**	1,34:1
5	2,69:1	**11**	1,22:1
6	2,24:1	**12**	1,12:1
		13	0,86:1

Figure 27 Number of the gear and its reduction.

The reduction that the gearbox gives is complemented with the reduction that the final gear, situated in the driving axle has.

The efficiency of the gearbox is normally elevated. In this paper it is going to be considered that the gearbox of the truck has an efficiency of 97% ($\eta_{gear} = 0{,}97$).

The combination of the gearbox and the double clutch makes the gearshift very fast. This is important because when the engine is gear shifting the wheels do not receive torque and the vehicle stars to slow. In the case of fast gearshift, the short time that the gear is shifting is very short and the velocity of the vehicle after doing the gearshift is very similar to the velocity that the vehicle had before.

4.5.3. Cardan axle

The cardan joint is the tool that allows transmitting an angular movement with a concrete direction without transmitting the other two directions angular movement. This tool is necessary in the majority of vehicles to not lose power when the driving axle is moving by the suspension sinking.

This part of the transmission does not present any reduction of the spinning ($i_{kar} = 1$) and its efficiency is almost 100% ($\eta_{kar} = 0{,}98$).

4.5.4. Final gear

This gear is situated in the middle of the driving axle and its function is to reduce the angular velocity that the cardan has and increasing its torque. Doing this, the cardan, does not need to be subject to all the torque that the wheels need to move the set truck-trailer.

When a truck is sold the owner can choose which final gear prefers. To build this transmission system the final gear chosen has been the one with a reduction relation of 3,76:1 ($i_0 = 3{,}76$). This value is because it is one value from the middle of the range of possibilities that can be chosen and the expected truck is a flexible truck. This reduction relation means that the cardan will turn 3,76 times faster than the final gear and it will transmit 3,76 times more torque than the cardan.

The efficiency of this part of the transmission is around 97% ($\eta_0 = 0{,}97$).

4.5.5. Differential gear

The differential compensates the different angular velocity of the driving wheels: between the wheel that is inside of the curve and the wheel that is outside of it. The interesting thing of this system is that the wheels always receive the same torque. The system divides by two the torque that the final gear has and gives it to each wheel.

If because of the road conditions the wheels have different friction coefficients, the maximum torque that the differential can give to the wheels is the normal force that the ground gives to the wheels multiplied by the minor coefficient of friction.

This system's efficiency is around 97% ($\eta_p = 0{,}97$)

4.5.6. Map of the transmission system

Knowing all the characteristics of the transmission of the vehicle is the time to draw a map showing the force that the combination of the engine with this transmission system can give to the wheels.

The determination of the efficiency of the total transmission system the following equation is going to be used:

$$(eq.\ \ 30) \quad \eta_{trans} = \eta_{clutch} \cdot \eta_{gear} \cdot \eta_{kar} \cdot \eta_0 \cdot \eta_p = 0{,}95 \cdot 0{,}97 \cdot 0{,}98 \cdot 0{,}97 \cdot 0{,}97 = 0{,}85$$

Then using the reductions of each gear and the following equation it is easy to draw the map of the engine.

$$(eq.\ \ 31) \quad \Gamma_{wheel} = \eta_{trans} \cdot \Gamma_{engine} \cdot i_{gear} \cdot i_{kar} \cdot i_0 \cdot i_p$$

$$(eq.\ \ 32) \quad \Gamma_{wheel} = F_w \cdot r_w$$

The graph that will show the force that the engine gives to the wheel in each different gear and the forces that the wheel needs to move the truck in different inclinations will be drawn. To draw this graph it will be used the torque (figure 21 and figure 22) that the two engines presented in the point 4.4. have.

In the first graph (figure 28) it is possible to see how this gearbox fits in the D13K engine and the force at the wheel that each gear transmits to the wheel. It can be seen that in roads without inclination the truck will not have any force problems. But with a 5% of inclination the truck will have to reduce some gears and it will not be able to run at its maximum velocity. And the same will happen in roads with more inclination.

In the second one (figure 29), it is possible to see the forces that the gearbox can give to the wheel using the D16K engine. As this engine is more powerful and hard, it is normal to see that the force at the wheel that gives each gear is superior to the force that gives the D13K. This means that the D16K engine truck, if the inclination of the road is 5%, will be faster (only reducing one gear will be able to continue without problems) than the D13K engine truck. In harder inclinations, obviously, it will be to reduce more gears and run slower.

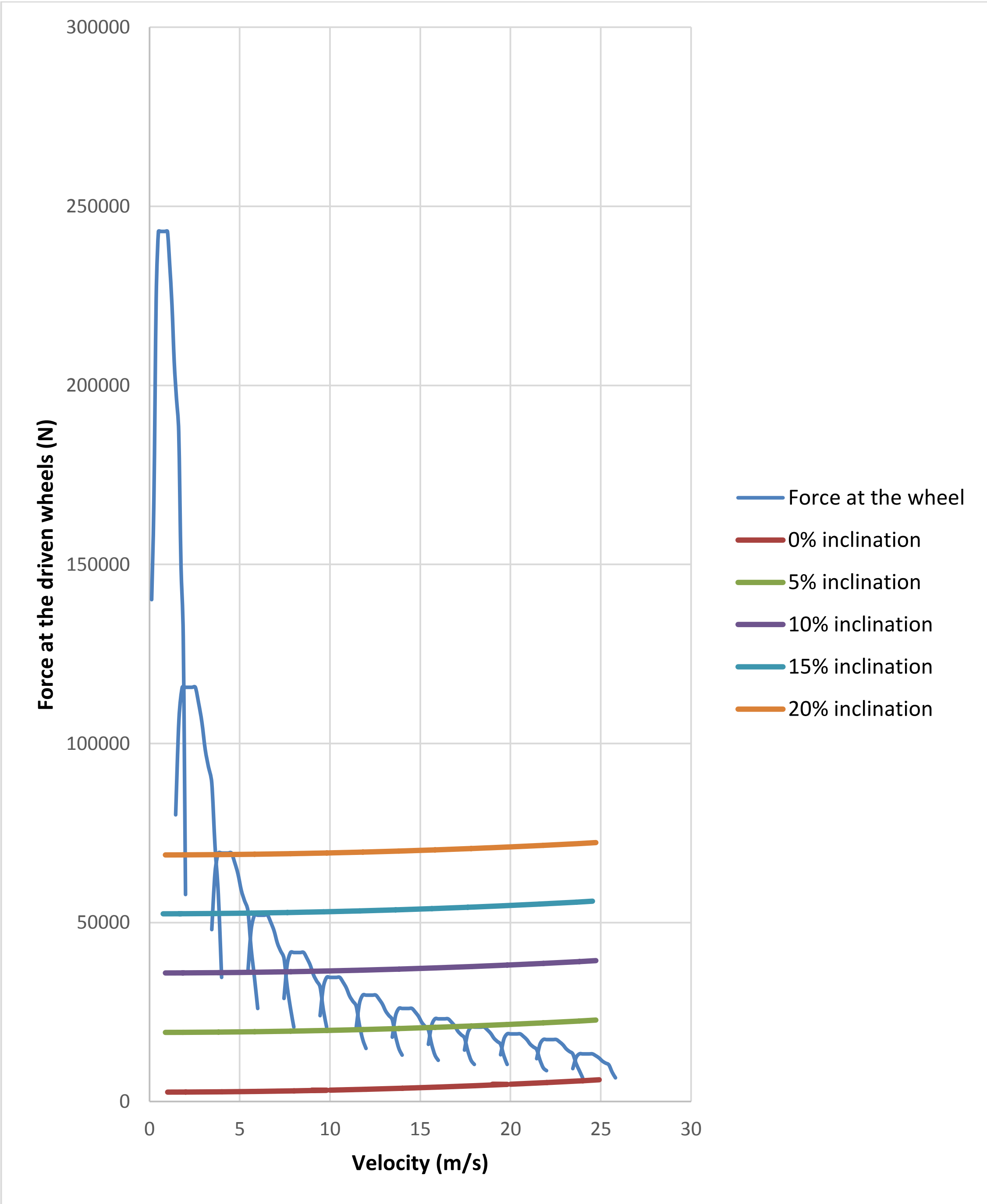


Figure 28 Representation of the force at the wheel of the engine D13K and the driving resistances for different inclinations.

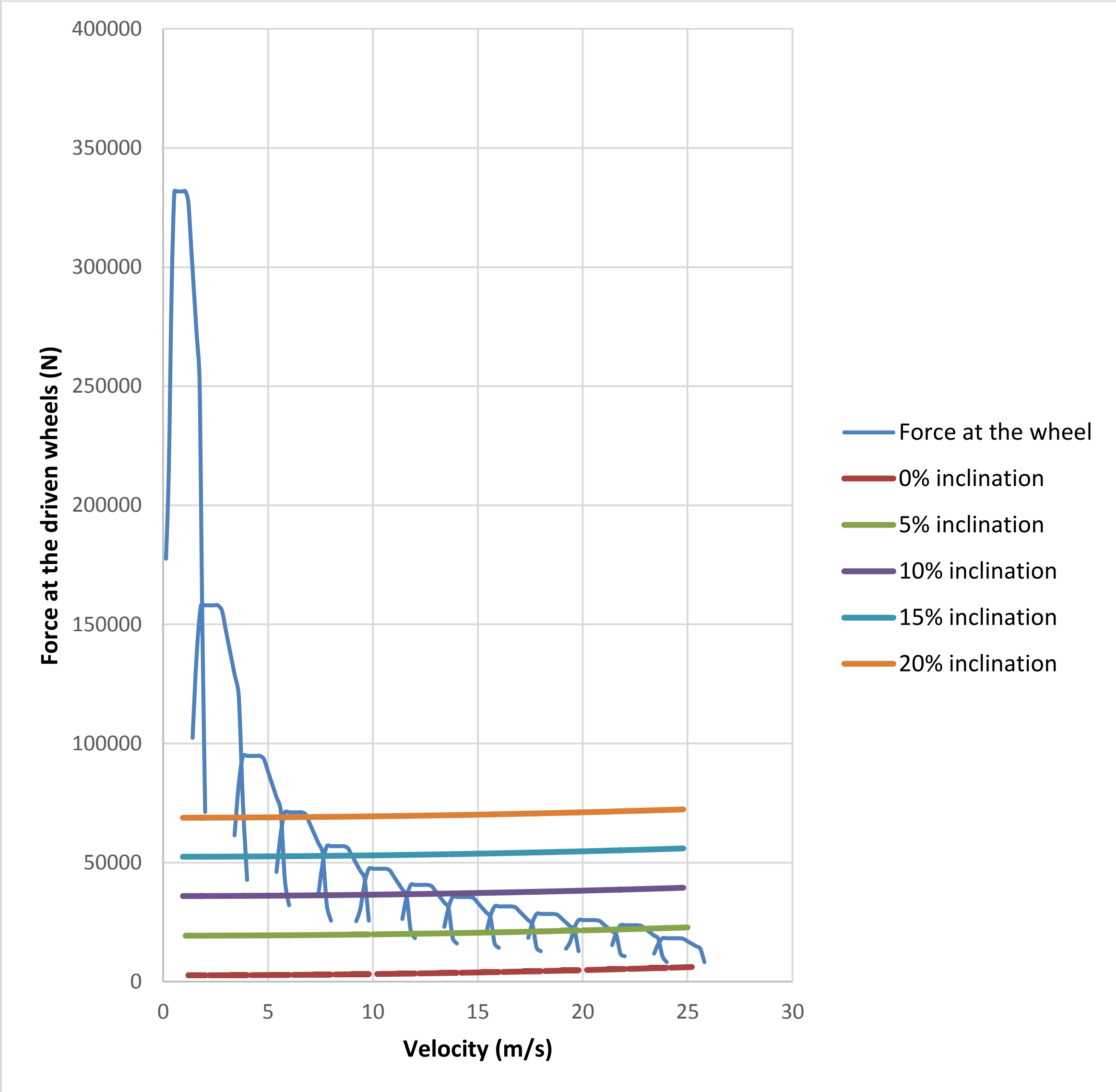


Figure 29 Representation of the force at the wheel of the engine D16K and the driving resistances for different inclinations.

4.6. Wheels and tires

The tires are ones of the most important parts of the vehicle because that part is in contact with the road and is in that part where the truck interacts with the medium.

To choose appropriate set of tires for the trailer and the truck it is necessary to know the needs they must meet. Basically these needs are two: the maximum weight they can hold and the maximum velocity they can run.

As have been said, the trailer cannot weight more than 24000kg in total. Considering that the whole weight of the trailer is held mostly by the axles and then by the fifth wheel, solving the equations with different inclinations and accelerations and it is possible to see that every trailer axle may hold at most less than 8000kg. On the other hand the legislation says that the maximum weight per axle in triaxle tandems cannot exceed 8000kg. As a consequence it can be said that the maximum weight that every trailer tire will must hold is 4000kg.

In Spain this type of heavy trucks, by law, cannot run faster than 90 km/h. For that reason the chosen tire will be a tire that cannot run over 90km/h or 100km/h.

On the other hand there is the truck. One of its axle, the drive axle, can hold 11,5 tones and in this axle the best option is to put double wheel because the tires with that weight coefficient cannot run fast. The others axles, in our case only the front one, can hold 10 tones but doing the hypothesis it was seen that, rarely, this axle should hold more than 8 or 8,5 tones. Considering this and the difficulty of putting double wheel at the direction axle it has been decided, for this truck, to put the maximum weight of this axle at 8,5 tones.

Basically there are two different tires that fit the needs of the truck. Both types of tires have the same velocity limit (J), the same diameter and the same diameter of wheel. But one is more width, and can hold more weight, than the other.

The dimensions of the tires and rims are:

385/65 R 22,5 160/158 J

315/80 R 22,5 156/153 J

(The tables of the parameters of the tires are in the Annex 1)

Both the tires have the same external diameter, 1072mm, and can go at 100km/h. The first one can hold 4500kg and the second one can hold 4000kg.

According to our model it can be said that the trailer can run with the second type of tires (315/80 R 22,5 156/153 J) without problems of weight. At the time to decide which tires are the best for the truck it has to be considered that the direction axle can hold 8500kg of weight. This means that it only can use the first tire type (385/65 R 22,5 160/158 J). The driving axle can go with double wheel and use the second type of tire.

On the other hand it is necessary to know the friction coefficient that the tires of the driving axle must have to let the set move when it will run in different conditions. Using the normal force in that axle and the friction force needed to move the set in different conditions it is possible to calculate the minimum friction coefficient needed to move the set truck-trailer at different inclinations and velocities.

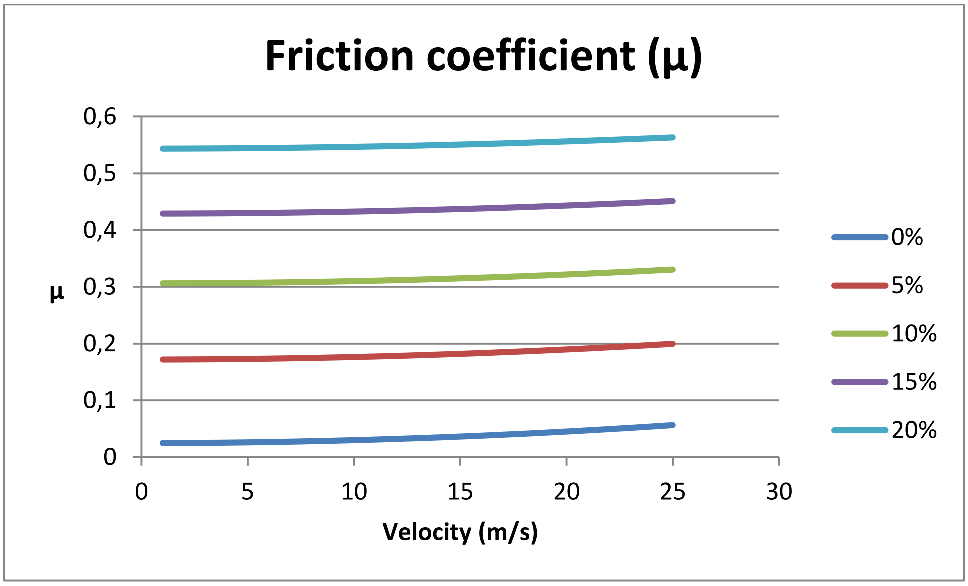


Figure 30 Required friction coefficient between tires and road surface.

In the figure 30 it is shown that the friction coefficient is smaller than 0,6 in all common asphalted roads and in main driving situations (inclinations of the road under 20%).

With these values of the friction coefficient, it can be said that the truck will be able to move in common asphalted roads without having grip problems because main of the truck tires have a maximum friction coefficient μ=0,8 on dry asphalt and μ=0,55 on wet asphalt.

The truck will not have problems in conditions of dry asphalt. But in wet asphalt conditions and in important road inclinations, the set can have some grip problems especially in hard accelerations or in hard breakings.

4.7. Suspension

The suspension is fundamental to stabilize the vehicle in bad road conditions and distribute to all the wheels the weight of the vehicle in bad road conditions.

Most of heavy vehicles are built with pneumatic springs.

A pneumatic spring is a spring that contains a defined gas volume which is compressed with the spring deflection. The spring force results from the effective working area and the differences between the internal and external air pressures:

$$(eq.\ \ 33) \qquad F = A \cdot (p - p_a) \qquad \begin{cases} F = force\ of\ the\ string \\ A = effective\ working\ area \\ p = internal\ air\ pressure \\ p_a = ambient\ air\ pressure \end{cases}$$

There are two basic types of gas springs in use: constant gas volume spring and constant gas mass spring.

The spring that can be more advantageous for the trailer is the constant gas volume spring because that kind of spring provides a nearly constant natural frequency with changing load. This type of spring normally uses ambient air. That thing makes the system extremely simple. If the static load is charged a system add more air with a compressor and the gas volume remains constant. As a consequence the internal pressure increases (according to the previous equation) proportionally to the load and the spring rate increases more or less like the following equation:

$$(eq.\ \ 34) \qquad c = \frac{F^2 p^n}{(p - p_a)^2 V} \qquad \begin{cases} 1 < n \leq 1{,}4 \\ V = air\ volume\ of\ the\ string \\ F = force\ of\ the\ string \\ p = internal\ air\ pressure\ of\ the\ string \\ p_a = ambient\ air\ pressure \\ c = spring\ rate \end{cases}$$

The other type of spring, with constant gas mass, hydraulic fluid is the working medium to transfer the pressure to the gas volume. Movement of the fluid with wheel travel is usually controlled by an integral damping device. The working area is defined by the diameter of the piston road. As the gas volume V, in contrast to the case of the constant-volume spring, decreases with growing load, the spring rate increases by a greater amount than the load, and hence the natural frequency also, so a constant-mass gas spring behaves in a non-linear manner.

As it has been said before, the truck will use the constant gas volume spring because that spring provides a constant natural frequency with different loads and that is what it needs. The truck has to have a similar behavior with the trailer full or empty.

4.8. Brake system

This kind of vehicles usually uses air brakes because the several advantages for large trailer vehicles that these brakes has. The main one is that this kind of brake system can never run out of its operating fluid, as hydraulic brakes can, and minor leaks do not result in brake failures.

In these brakes compressed air is used as the energy medium. The brake pedal effort of the operator is used only to modulate the air pressure applied to the brake chambers. To avoid failures brake air systems must have a dual air brake system so that in the event of one circuit failure, emergency braking function is maintained.

The basic tractor trailer system function is divided in two air circuit to have a second option in case of failure of the principal circuit and it works in this way: The compressor charges a wet supply reservoir from which two tractor reservoirs are fed, namely one front and one rear circuit reservoir. This compressor also charges two reservoirs of the trailer (the trailer service reservoir and the spring brake reservoir). When the front brake circuit fails, a valve immediately close off the front circuit to protect the rear circuit, which continues to function normally. A similar protection is installed in case of a rear brake circuit failure. If the front and the rear brake systems become inoperative, spring brakes will apply automatically when the air pressure drops below approximately 275kPa.

Apart of that the tractor will have an engine brake. This brake works transforming the engine to a compressor and, to compress the air, using the kinetic energy that the truck has making circulate it slower. This system is to help the other brakes and it does not have wear. Because of that it is possible to use it during long periods of time and log downhill.

Is important to point that this system only works to slow the truck, not to stop it. This system only can help the driver in long downhill where the other brakes heat and lose the capacity of braking but newer in slow velocities because to use this kind of break the truck needs a minimum amount of kinetic energy.

4.9. Vehicle Acceleration

Once the force and the torque that the two different truck engines can give to the driving wheels and the driving resistances that the set truck-trailer has are known it is possible to calculate the maximum acceleration that the vehicle can have, in different velocities. These maximum accelerations will be calculated considering that the shifting of the gears is done in the most effective way.

These values of acceleration have been calculated considering an inclination of 0% and a gear shifting time of 0.08s. This short time of gearshift is because of the vehicle's double clutch system and makes that in the graph of the velocity depending on the time (figure 31 and figure 32) the gearshift times are not appreciable.

In the figure 31 and the figure 32 it is possible to see how much time will spend the D13K and D16K engines, respectively, to raise the velocity of 90km/h. The D16K, the most powerful engine, can arrive at 90km/h easier than the D13K engine.

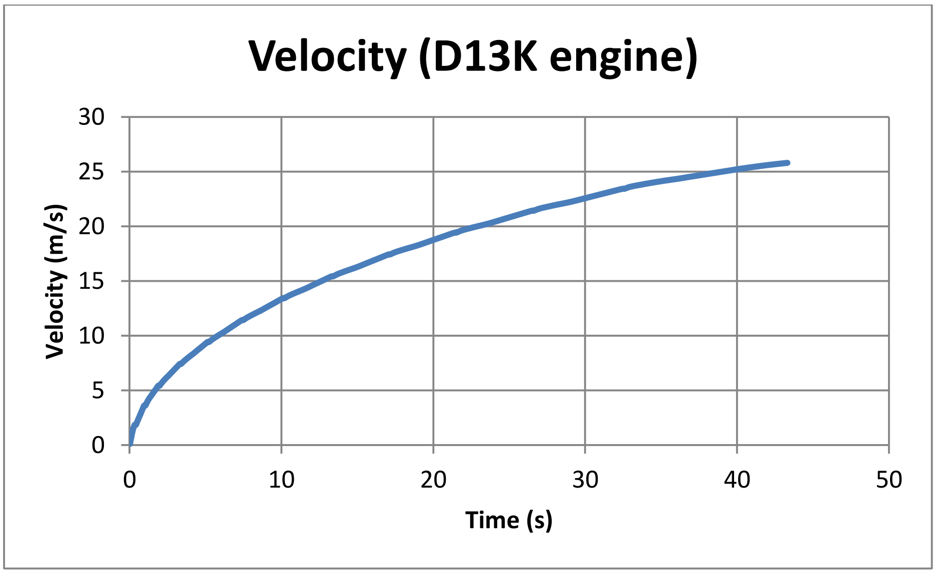


Figure 31 Acceleration of the engine D13K. Velocity of the set truck-trailer depending on the time.

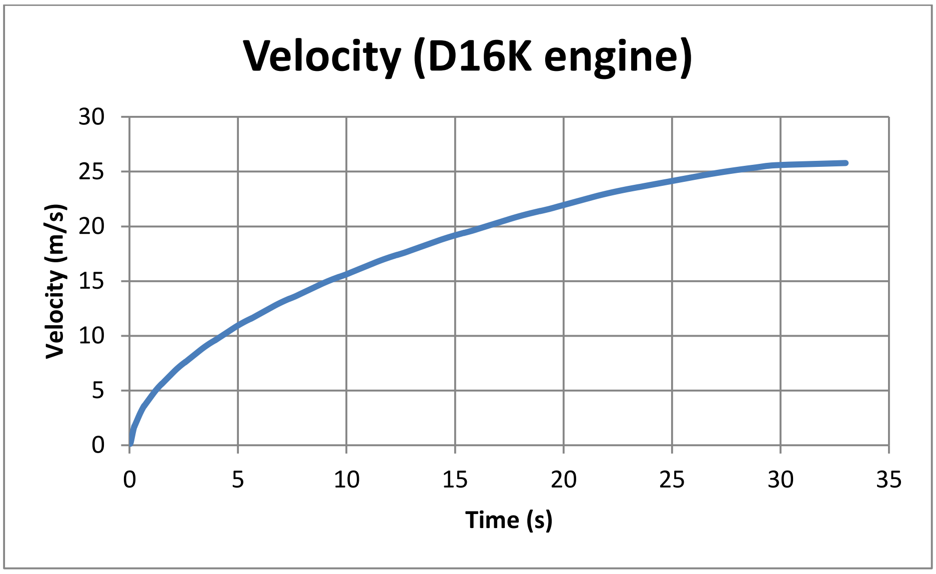


Figure 32 Acceleration of the engine D16K. Velocity of the set truck-trailer depending on the time.

On the other hand it is possible to calculate the distance that the truck will have to run to raise that velocity. To do that the only necessary thing is integrate the line of the velocity. But this function has no equation and because of that the only way to calculate this is calculating the area below the line.

$$S_{ac} = \int dS_{ac} = \int_{t_0}^{t} v \cdot dt$$

The engine D13K (figure 31) needs 637,5 meters to run at 90 km/h (25 m/s) in a road without inclination and full loaded. The engine D16K (figure 32) needs 462,5 meters do run at 90 km/h (25 m/s) with the same conditions.

$$\frac{637{,}5}{462{,}5} = 1{,}37 = 137\%$$

The vehicle with the engine D13K needs a 37% more distance to go at 90 km/h starting from zero.

5. Vehicle dynamics

(Prof. Rr.-Ing Konrad Rief, Dipl--Ing. Karl-Heinz Dietsche, 2014)

In this section some theory and study the vehicle accelerations and forces related to the steering, cornering and lateral inclinations of the road will be present. The parameters used will be the parameters shown in the following figure 33:

Table 4. Vehicle dynamics parameters.

Quantity		Unit
δ	Axle steering angle	rad
δ_H	Steering-wheel angle	rad
α_v	Slip angle of front axle	rad
α_h	Slip angle of rear axle	rad
β	Float angle	rad
ψ	Yaw angle	rad
ω_e	Undamped natural frequency	s^{-1}
l	Wheelbase	m
l_v	Distance between front axle and center of gravity	m
l_h	Distance between rear axle and center of gravity	m
v	Longitudinal velocity	m/s
v_r	Resulting wind impact velocity	m/s
C_v	Rear cornering stiffness Front axle	N/rad
C_h	Rear cornering stiffness Rear axle	N/rad
D	Damping factor	1/rad
m	Total mass (weight)	kg
i_l	Steering ratio	–
F_{SV}	Lateral force on front axle	N
F_{SH}	Lateral force on rear axle	N
a_y	Lateral acceleration	m/s^2
θ	Yaw moment of inertia	Nms^2
ρ	Air density	kg/m^3
A	Frontal area	m^2
τ	Angle of impact	rad
F_S	Crosswind force	N
M_Z	Crosswind yaw moment	Nm

Figure 33 Parameters used in the vehicle dynamics. (Prof. Rr.-Ing Konrad Rief, Dipl--Ing. Karl-Heinz Dietsche, 2014) (Automotive Handbook, 2014 9th edition)

The vehicle has different balance axles and directions that will be used to study and explain the different phenomenon of its dynamics. This axles are the axles that can be seen at the picture in the figure 34.

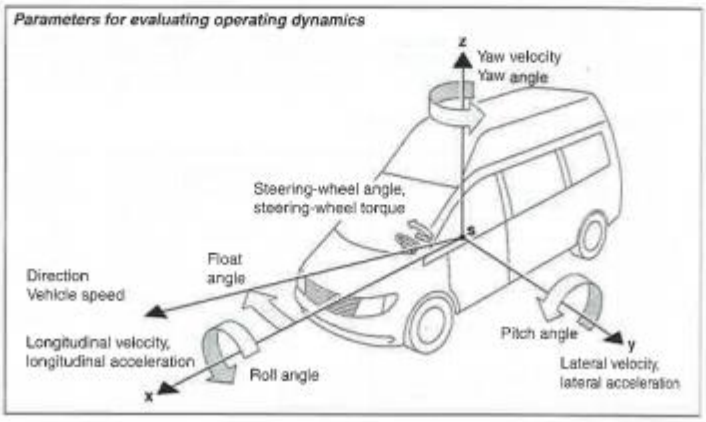


Figure 34 Balance axles of the vehicle. (Prof. Rr.-Ing Konrad Rief, Dipl--Ing. Karl-Heinz Dietsche, 2014) (Automotive Handbook, 2014 9th edition)

5.1. Lateral acceleration

The passenger vehicles that are circulating nowadays are able to reach lateral accelerations of up to 10 m/s^2. The studied truck does not have these values, the maximum lateral acceleration that this kind of vehicles can reach without having stability problems is near 5 or 6 m/s^2 but this is dependent on the load of the trailer.

There are different ranges of lateral acceleration. The range between 0 and 0,5 m/s^2 is known as the small-signal range. The lateral accelerations that have these values are caused by the changing crosswind and the irregularities of the road. The range from 0,5 to 4 m/s^2 is known as the linear range, as the vehicle behavior that occurs in this range can be described with the aid of the single-track model. This range of accelerations occurs in actions as changing driving lanes or combinations of maneuvers. In the lateral acceleration range between 4 to 6 m/s^2 is a transition range because some vehicles start to lose its stability. For example for the case of the studied truck that range is the limit of lateral acceleration that can have. Another kind of vehicles as sport cars can have this range of lateral accelerations without experimenting any stability problems. The last lateral acceleration range is above 6 m/s^2, these lateral accelerations only occurs in extreme cases or in racing circuits. Usually when a vehicle experiment this accelerations it finish in accident in a normal traffic road.

The most common lateral accelerations that the vehicles experiment when they are circulating are the accelerations around 4 m/s^2.

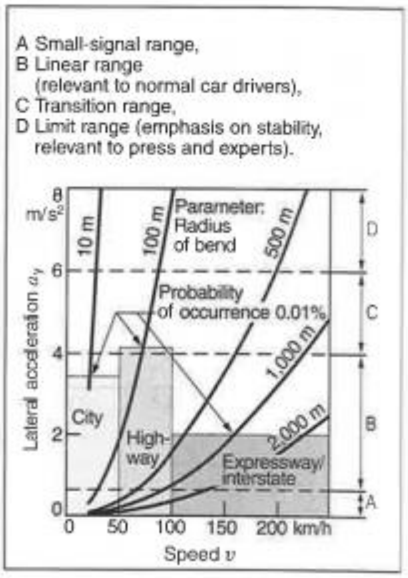


Figure 35 Kinds of lateral acceleration. (Prof. Rr.-Ing Konrad Rief, Dipl--Ing. Karl-Heinz Dietsche, 2014) (Automotive Handbook, 2014 9th edition)

5.2. Self-steering effect

The slip angle is the angle that is formed between the longitudinal axis and the velocity vector of the tire and the angle β is the angle between the velocity vector at the center of gravity and the longitudinal axis of the vehicle. With the torque balance and these angles, it is possible to calculate the change in the steering-wheel angle in connection with increasing lateral acceleration for the skid-pad maneuver at a constant radius. The self-steering definition comes from that:

$$(eq.\ \ 35) \qquad EG = \frac{m}{l} \cdot \left(\frac{l_h}{C_v} - \frac{l_v}{C_h}\right)$$

All passenger cars are designed to under-steer in the linear lateral acceleration range. The self-steering gradient (EG) value for the passenger vehicles is closer to 0,25 degrees·s^2/m. This self-steering gradient improves the stability and damping of the vehicle and draws the driver's attention to the increasing lateral acceleration.

The float angle gradient (SG) should be as low as possible to increase the stability of the vehicle. That angle can be calculated from the figure 36:

$$(eq.\ \ 36) \qquad SG = \frac{m \cdot l_v}{C_h \cdot I}$$

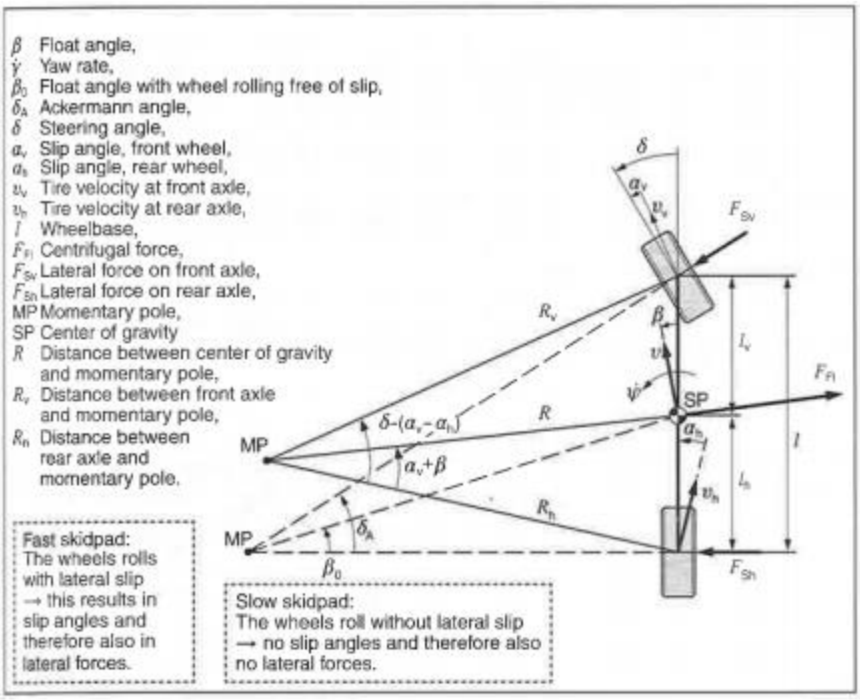


Figure 36 Self steering scheme. (Prof. Rr.-Ing Konrad Rief, Dipl--Ing. Karl-Heinz Dietsche, 2014) (Automotive Handbook, 2014 9th edition)

Heavy duty commercial vehicles with air suspension, the studied case, are normally equipped with solid or rigid axles controlled with suspension arms and links. Such axles are generally designed to ensure that the proportion of self-steering properties in axle control is virtually constant for all laden states since there is no difference in level between unloaded and loaded.

Wheel loads at the truck's rear axle vary dramatically, depending on whether the truck is loaded or unloaded. This makes the combined vehicle responding to reductions in load with more under-steer.

5.3. Damping factor

The damping factor of a vehicle can be identified from the yaw response of sudden steering or step input. All the vehicles are designed to ensure that damping is as high as possible.

A self-steering gradient is required if the vehicle is to have good straight-running characteristics but a low self-steering gradient is necessary to facilitate a high damping factor, particularly at high speeds.

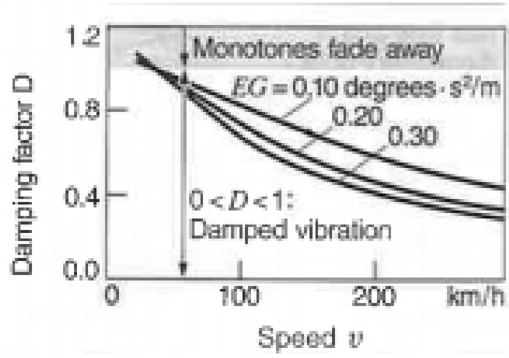


Figure 37 Damping factor. (Prof. Rr.-Ing Konrad Rief, Dipl--Ing. Karl-Heinz Dietsche, 2014) (Automotive Handbook, 2014 9th edition)

5.4. Cornering behavior

The cornering of a vehicle is affected by the centrifugal force, which pulls the vehicle away from the path of travel. The equation of this force is the following:

$$(eq.\ 37) \qquad F_{cf} = \frac{m \cdot v^2}{r_k}$$

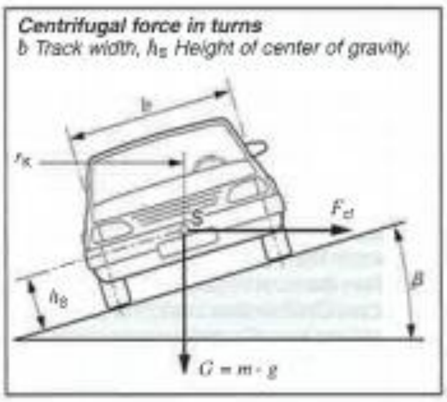


Figure 38 Car forces in cornering. (Prof. Rr.-Ing Konrad Rief, Dipl--Ing. Karl-Heinz Dietsche, 2014) (Automotive Handbook, 2014 9th edition)

The magnitude of the tilt away that the vehicle experiments during the turn, depends on the rates of the springs and their response to alternating compression. The roll axis is the body's instantaneous axis of rotation relative to the road surface. This axis has the longitudinal direction of the vehicle. Like all rigid bodies the vehicle does a screwing or a rotation motion. This motion is supplemented by a lateral displacement. The most stability position of the instantaneous axis of rotation of the car is it passing through the center of gravity. That means that the lateral displacement of the vehicle is null. When the instantaneous axis of rotation is not so close to the center of gravity the vehicle will slip more in turns. In the case of the trucks the roll angle is measured and evaluated not only at the center of vehicle gravity, because due to the torsional weak frame and the separated driver's cab mounting it is measured in several fixed points.

This lateral displacement is caused by the cornering force. The cornering force is generated by tire slip and is proportional to slip angle at low slip angles. The rate at which cornering force appears is described by relaxation length. Slip angle describes the deformation of the tire contact patch, and this deflection of the contact patch deforms the tire in a fashion akin to a spring.

The deformation of the tire contact patch, as it is like a spring, generates a reaction force in the tire; the cornering force. Integrating the force generated by every tread element along the contact patch length gives the total cornering force.

Having the instantaneous axis of rotation near the center of gravity generally implies a corresponding upward displacement of the instantaneous axes of wheels, resulting in a change in track width. For this reason, design seeks to combine a high instantaneous roll center with minimal track change. The best option is to put the instantaneous axes of the wheels as high as possible and place them as far as possible from the body of the vehicle.

Critical speeds (numerical value equations)

	Flat curve	Banked curve
Speed at which the vehicle exceeds the limit of adhesion (skid)	$v \leq 11{,}28\sqrt{\mu_r \cdot r_K}$ km/h	$v \leq 11{,}28\sqrt{\dfrac{(\mu_r + \tan\gamma)\cdot r_K}{1-\mu_r \cdot \tan\gamma}}$ km/h
Speed at which the vehicle tips	$v \geq 11{,}28\sqrt{\dfrac{b \cdot r_K}{2 \cdot h_S}}$ km/h	$v \geq 11{,}28\sqrt{\dfrac{\left(\dfrac{b}{2 \cdot h_s} + \tan\gamma\right)\cdot r_K}{1 - \dfrac{b}{2 \cdot h_S}\cdot \tan\gamma}}$ km/h

h_S Height of center of gravity (in m), μ_r Max. coefficient of friction, b Track width (in m), r_K Curve radius (in m), γ Curve banking.

Figure 39 Critical speeds in cornering. (Prof. Rr.-Ing Konrad Rief, Dipl--Ing. Karl-Heinz Dietsche, 2014) (Automotive Handbook, 2014 9th edition)

6. Trailer

Once all the truck parts have been designed and decided is the moment to start thinking on the trailer design and its unload systems.

The unload systems that will be studied are three different unload systems. The first one will be unload by chains system, then unload by ejector system and finally dump unload system. These three different systems will be compared and analyzed. In this point the force that each system, for one specific material (same viscosity and coefficients of friction), needs to do to perform the unload procedure will calculated.

To have an equal point of view of each unload system, all the unload systems will be studied using the same trailer shape and size. These measures will be the ones used during the design of the truck. Only it can be some differences in the length of the trailer because the things to transport are heavy and it is not necessary to have the longest trailer allowed. The trailer that will be used will have a length of 8 meters. Despite of this the effective measures of the box of the trailer will be 7m x 2,15m x 1,8m of length, width and deep.

6.1. Force calculations

To calculate the forces involved inside the trailer is necessary to use the theory of granular solid because the load that the trailer will transport are different kind of grain.

First of all, the forces that the load does to the trailer container have to be calculated. The equations of the granular solid will be used to calculate the pressure that the load does to the walls and the ground of the trailer.

$$(eq.\ \ 38)\qquad P_v = P_\infty \cdot \left(1 - e^{-\frac{\mu K Z}{R}}\right)\qquad (\text{C. J. Brown, J. Nielsen, 1998})$$

$$(eq.\ \ 39)\qquad P_\infty = \frac{\gamma R}{\mu}$$

$$(eq.\ \ 40)\qquad P_h = K \cdot P_v$$

$$(eq.\ \ 41)\qquad K = \frac{1 - \sin^2 \emptyset}{1 + \sin^2 \emptyset}$$

Where μ is the $friction\ coeficient$ between the load and the trailer, γ is the *density of the load*, *Z is* the *deep of the load*, ∅ is the *excluded volume* (the excluded volume is the division between the real volume of the granular solid and the volume that this solid has when it is in one place, counting the air holes in the middle of the grains) and R is the hydraulic radius.

$$\begin{cases} \mu = friction\ coeficient \\ \gamma = density\ of\ the\ load \\ Z = deep\ of\ the\ load \\ \emptyset = excluded\ volume \\ R = \dfrac{wet\ area}{wet\ perimeter} = hidraulic\ radius \end{cases}$$

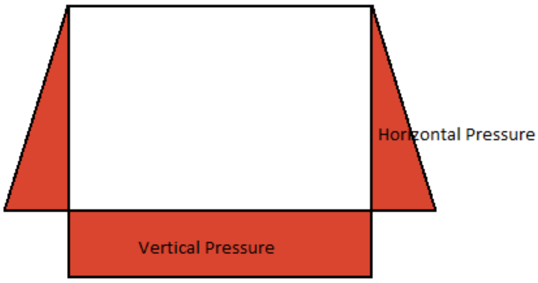


Figure 40 Pressures at the walls and the floor of the trailer.

For example, if the load is rice: $\gamma = 580 kg/m^3$, $\emptyset = \frac{1}{6}$, Z = 1,5m, $\mu \approx 0{,}2$ and the R of the trailer $\frac{7 \cdot 2{,}15}{(7+2.15) \cdot 2} = 0{,}822m$.

$$(eq.\ \ 42) \qquad P_\infty = \frac{\gamma R}{\mu} = \frac{580 \cdot 0{,}822}{0{,}2} = 2383{,}8 \frac{kg}{m^2}$$

$$(eq.\ \ 43) \qquad K = \frac{1 - \sin^2 \emptyset}{1 + \sin^2 \emptyset} = 0{,}9464$$

$$(eq.\ \ 44) \qquad P_v = P_\infty \cdot \left(1 - e^{-\frac{\mu K Z}{R}}\right) = 2383{,}8 \cdot \left(1 - e^{-\frac{0{,}2 \cdot 0{,}9464 \cdot 1{,}5}{0{,}822}}\right) = 696{,}2\ kg/m^2$$

$$(eq.\ \ 45) \qquad P_h = K \cdot P_v = 0{,}9464 \cdot 696{,}22 = 658{,}9\ kg/m^2$$

As it can be seen the vertical pressure and the horizontal pressure at the same deep are not the same, this is because the characteristics of granular solid.

The P_h is the pressure in the point near the ground of the trailer, the maximum horizontal pressure. The distribution of horizontal pressure has a triangular distribution (Figure 40), starts with 0 kg/m^2 and finishes with the value of 658,9 kg/m^2 in its deepest part. To work with the horizontal distribution of pressures the mean of the pressures will be used.

With these values it is possible to calculate the specific values that each unload system has and then compare it.

6.1.1. Unload by chains

This unload system is used mostly in agricultural transportations as grain. This system has been used for a long period and some companies are still building trailers with it.

The system consists in different transversal metal bars set in the floor of the truck. These bars are pulled through the length of the trailer by chains. The bars make the load move to the rear part of the trailer and this load gets out by the rear door of the mentioned trailer.

To facilitate the calculations and theories the chains will be assumed as a belt conveyor.

With the pressures that have been calculated in the point 6.1. is possible to calculate the normal forces that the load does to the trailer and the friction force between the trailer walls and the load.

With the pressure is possible to calculate the total force that the load does to the trailer:

$$F = P \cdot S$$

$$(eq.\ \ 36) \qquad N_{one\ side} = \frac{P_{h(Z=0)} + P_{h(Z=1.5)}}{2} \cdot (L \cdot Z)$$
$$= \frac{0 + 658{,}9}{2} \left[\frac{kg}{m^2}\right] \cdot 9{,}81 \left[\frac{N}{kg}\right] \cdot (7[m] \cdot 1{,}5[m]) = 3{,}39\ kN$$

With that normal force and the friction coefficient is possible to calculate the friction force between the load and the trailer walls. As the trailer is symmetric calculating once the normal force in one wall and then multiply per two.

$$F_{f\ one\ side} = \mu \cdot N_{one\ side} = 0{,}2 \cdot 33934{,}99[N] = 6{,}8\ kN$$

Then as friction force in one side is 6786,99N, the force that the unload chains must do (F_{ch}) to move the load is the double, 13573,99N, plus the friction that the chains have with the ground of the trailer.

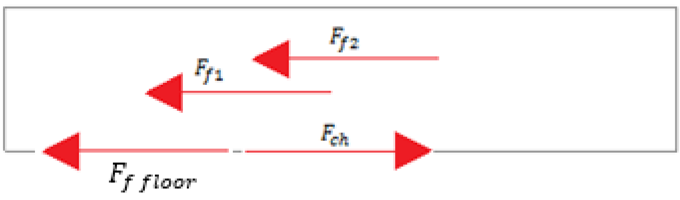


Figure 41 Forces at the trailer in during the chains unload.

Now is the time to calculate the force that the truck must do to the chain considering its friction with the trailer's floor.

If the trailer has some cylinders under the belt conveyor the belt-trailer floor friction coefficient can be reduced: $\mu_{belt-trailer\ floor} = 0{,}05$.

Using the same equations as before it is easy to find the friction coefficient:

$$F = P \cdot S$$

$$(eq.\ \ 47) \qquad N_{ch} = P_v \cdot (L \cdot B) = 696{,}22 \left[\frac{kg}{m^2}\right] \cdot 9.81 \left[\frac{N}{kg}\right] \cdot (7[m] \cdot 2{,}15[m]) = 102{,}79\ kN$$

$$(eq.\ \ 48) \qquad F_{f\ floor} = \mu \cdot N_{floor} = 0{,}05 \cdot 102790{,}27[N] = 5{,}1\ kN$$

Summing the force that the load needs to be moved and the resistance force that the friction between the chains and the ground of the truck does, it is easy to know which will be the force that the chains have to do to unload the trailer.

$$(eq.\ \ 49) \qquad F_{f\ one\ side} + F_{f\ one\ side} + F_{f\ floor} = 6786{,}99 + 6786{,}99 + 5139{,}51 = 18{,}7\ kN$$

6.1.2. Unload by ejector

This system has been implemented the last years and is quite new. Is a very versatile system, it can transport a large range of materials.

The system consists on an ejector, that is moving, with the help of hydraulic pistons, and push the front part of the trailer load. Because of that pressure, the rear part of the load starts moving and the unloading process starts. The idea of this unload system is very similar a syringe.

To know all the forces involved in this unload system is necessary to calculate the normal forces in the walls and in the floor of the truck. To calculate this is necessary to use the pressures previously calculated:

$$(eq.\ 50) \qquad N_{one\ side} = \frac{P_{h(Z=0)} + P_{h(Z=1.5)}}{2} \cdot (L \cdot Z)$$
$$= \frac{0 + 658{,}9}{2} \left[\frac{kg}{m^2}\right] \cdot 9{,}81 \left[\frac{N}{kg}\right] \cdot (7[m] \cdot 1{,}5[m]) = 33{,}9\ kN$$

$$(eq.\ 51) \qquad N_{floor} = P_v \cdot (L \cdot B) = 696{,}22 \left[\frac{kg}{m^2}\right] \cdot 9{,}81 \left[\frac{N}{kg}\right] \cdot (7[m] \cdot 2{,}15[m])$$
$$= 102{,}8\ kN$$

Considering that the friction coefficient is 0,2 is not difficult to calculate the force that the material needs to be unload.

$$(eq.\ 52) \qquad F_{f\ one\ side} = \mu \cdot N_{one\ side} = 0{,}2 \cdot 33934{,}99[N] = 6{,}8\ kN$$

$$(eq.\ 53) \qquad F_{f\ floor} = \mu \cdot N_{floor} = 0{,}2 \cdot 102790{,}27[N] = 20{,}6kN$$

Summing the friction forces, the maximum force that the ejector will have to do to unload the trailer will be known.

$$(eq.\ 54) \qquad F_{f\ one\ side} + F_{f\ one\ side} + F_{f\ floor} = 6786{,}99 + 6786{,}99 + 20558{,}05$$
$$= 34{,}1\ kN$$

To unload the trailer with the ejector system the mechanism will have to do a maximum force of 34,132kN.

6.1.3. Dump unload

This kind of unload system is mostly used in sand transportation and in other kind of dry materials.

The system consists in incline the trailer in a concrete angle until the mass of the load overcomes the friction forces and the load stars moving. The angle that will be necessary to dump the load will depend on the friction coefficient between the load and the trailer walls.

Considering the same values used in the point 6.1.1. it can be calculated the normal force and the critical angle where the unload starts.

With the pressure it is possible to calculate the total force that the load does to the trailer:

$$F = P \cdot S$$

Now the deep of the trailer has increased because of the inclination of it. To calculate the pressure in the walls and in the floor of it will be necessary to correct that deep adding a corrective factor: $1/\cos\beta$

$$(eq.\ \ 55) \qquad N_{one\ side} = \frac{P_{h(Z=0)} + P_{h(Z=1.5)}}{2} \cdot \frac{1}{\cos\beta} \cdot (L \cdot Z)$$

$$= \frac{0 + 658{,}9}{2} \cdot \frac{1}{\cos\beta}\left[\frac{kg}{m^2}\right] \cdot 9{,}81\left[\frac{N}{kg}\right] \cdot (7[m] \cdot 1{,}5[m]) = 33{,}9 \cdot \frac{1}{\cos\beta}\ kN$$

To calculate the normal force that the floor of the trailer does to the load the inclination of the trailer must be considered. That normal force will be the cosines component of the mass.

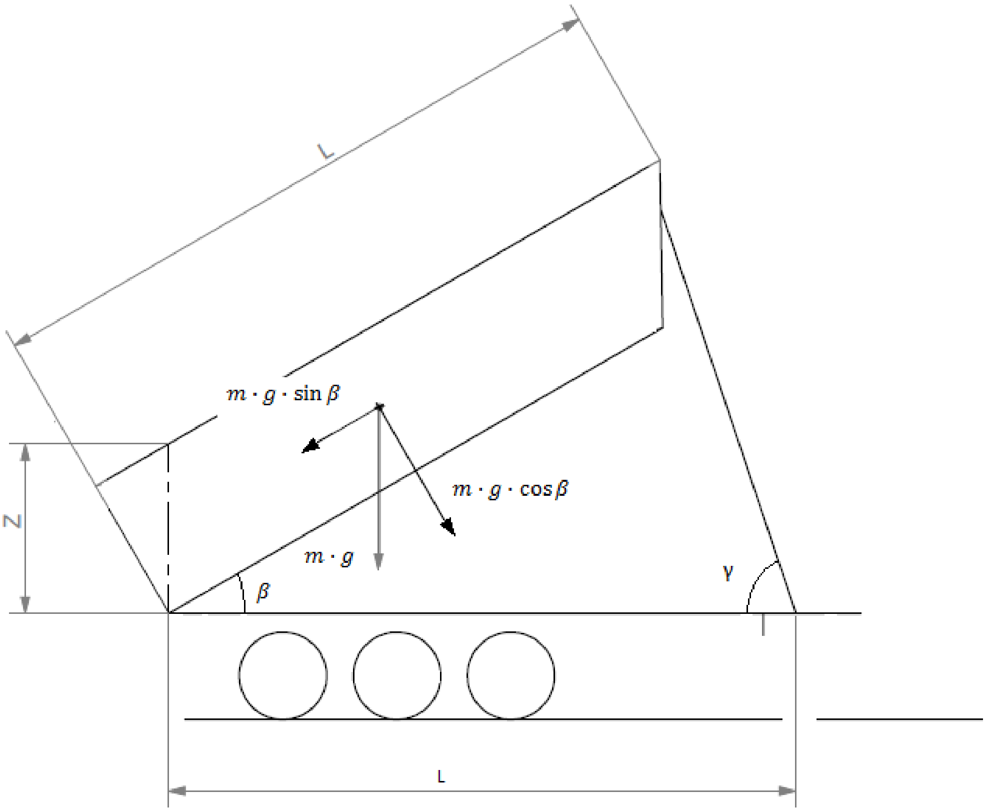


Figure 42 Forces in the dumping trailer.

$$(eq.\ 56) \qquad N_{floor} = P_v \cdot \frac{1}{\cos\beta} \cdot \cos\beta \cdot (L \cdot B)$$

$$= \frac{696{,}22}{\cos\beta} \cdot \cos\beta \cdot \left[\frac{kg}{m^2}\right] \cdot 9{,}81 \left[\frac{N}{kg}\right] \cdot (7[m] \cdot 2{,}15[m]) = 102{,}8\ kN$$

Now, to calculate the angle where the load starts moving and when the unload process starts is possible. That angle will be the angle that makes the friction force and the weight force have the same value.

$$(eq.\ 57) \qquad N \cdot \mu = m \cdot g \cdot \sin\beta$$

$$\left(2 \cdot N_{one\ side} + N_{floor}\right) \cdot \mu = volume \cdot \gamma \cdot g \cdot \sin\beta$$

$$\left(2 \cdot 33934{,}99 \cdot \frac{1}{\cos\beta} + 102790{,}27\right) \cdot 0{,}2 = 22{,}575 \cdot 580 \cdot 9{,}81 \cdot \sin\beta$$

$$\beta = 15{,}65\ deg$$

The force needed to unload using this system is the force that will be necessary to lift the trailer doing the dumping process.

Considering that the trailer is lifted by a hydraulic piston situated in the front part of it, is possible to calculate the vertical force (F_{pv}) that the piston may do using the sum of moments:

$$\Sigma M_C = 0$$

$$(eq.\ \ 58) \qquad m \cdot g \cdot \frac{L}{2} = F_{pv} \cdot L$$

$$F_{pv} = \frac{m \cdot g}{2} = \frac{volume \cdot \gamma \cdot g}{2} = \frac{22{,}575 \cdot 580 \cdot 9{,}81}{2} = 64{,}2\ kN$$

Once the vertical force is known it is possible to calculate the force that the piston must do using the geometry of the trailer. To know the piston force is necessary to know the angle γ.

The triangle that formed by the two parts of the trailer and the piston is isosceles and knowing that is easy to calculate the γ angle:

$$\mathrm{L} \cdot \sin\frac{\beta}{2} = \frac{piston\ longitude}{2} = L \cdot \cos\gamma$$

$$\gamma = 82{,}175^{o}$$

Knowing the vertical force that the piston must do and the inclination angle of it, the total force that it have to do can be calculated.

$$F_p = \frac{F_{pv}}{\sin\gamma} = \frac{64{,}224}{0{,}99} = 64{,}8\ kN$$

The total force that the piston will have to do to unload the trailer is 64,83 kN.

It is possible to see that the easiest way to unload the trailer, talking about forces, is using the chain system because is the system that requires less force. Then is the ejector system and finally the hardest system to unload, in force terms, is the trailer is the dumping system.

Every system has its advantages depending on the material that the trailer has inside but it can be said that the less powerful needed unload system is the unload by chains system.

Conclusions

As a conclusion of the work done during the evolution of this thesis, the engines that have been studied during will be compared in the different situations explained on the thesis.

The engines D13K and D16K are able to run in roads without inclination without having problems of acceleration independently on the velocity they are running.

When the truck is running in a roads with a positive inclination of 5%, very common roads, the engine acquire more importance because if the trucks are circulating in their maximum velocity (25 m/s or 90 km/h) they have to reduce one or more gears. The truck with the D13K engine will have to reduce 4 gears and circulate near 18 m/s (65 km/h). On the other hand the truck with the D16K engine will only have to reduce one gear, and it can run near 23 m/s (82 km/h). In this situation the D16K engine is advantageous.

If the thing to compare is the acceleration in different velocities it can be seen that in low velocities both engines has a similar acceleration: the D16K engine spends 8 seconds to accelerate between 0 m/s and 15 m/s (54 km/h) and the D13K engine spends 12 seconds to do the same. But in accelerations in faster velocities the difference between the powers of the two engines is more significant: the D13K engine spends 36 seconds to accelerate between 15 m/s (54 km/h) and 25 m/s (90 km/h) and the D16K engine spends 19 seconds to do the same. These differences in accelerations, especially in high velocities, is significant when the driver wants to do an overtaking or do an incorporation in a fast road.

In conclusion it can be said that if the driver who buy the truck has to run in irregular roads with inclinations and more traffic the ideal engine is the D16K engine because it provides more power to the driving wheels and because of that more reaction capacity. If the driver is most of the time driving in roads without important inclinations and at constant speed the D13K engine is a good engine because it is also powerful, it can do the work without problems and it is cheaper.

Having done the comparison of the two engines, a good way to continue through the topic and improving the thesis would be focusing and studying more the vehicle dynamics, especially the vehicle dynamics of this kind of combined vehicles, and go farther in the study of the different unload systems calculating the unload times and which materials are better for each unload system.

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And finally I would like to say that I truly enjoy doing this project and I hope that all the knowledge I have acquired doing it will help me in future projects and jobs. This topic is one of my favorite topics in engineering and in a near future I would really like to work or study more that topic.

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Annex

a. Annex 1

Tables of the parameters of the tires:

No.	Size	PR	Type	Standard Rim	Inflated Dimensions (mm)		Load Index (single/double)	Speed	Inflated Pressure (kPa)	
					SW	OD			Single	Double
1	7.50R16	14	TT	6.00G	215	805	122/118	K	770	770
2	8.25R16	16	TT	6.5	235	855	128/124	M	770	870
3	900R20	16	TT	7	259	1019	144/142	K	900	900
4	10.00R20	18	TT	7.5	278	1054	149/146	J	930	930
5	11.00R20	18	TT	8	293	1085	152/149	J	930	930
6	12.00R20	18	TT	8.5	315	1025	154/149	K	840	770
7	11R22.5	16	TL	8.25	279	1050	146/143	M	830	830
8	12R22.5	18	TL	9	300	1050	152/148	M	850	850
9	13R22.5	18	TL	9.75	320	1124	156/150	L	850	850
10	295/80R22.5	15	TL	9	298	1044	152/148	M	850	850
11	315/80R22.5	20	TL	9	300	1085	156/153	J	850	850

12	385/65R22.5	20	TL	11.75	389	1072	160/158	J	900	900
13	1200R24	20	TT	8.5	315	1226	160/157	K	900	900
14	1100R22	18	TT	8.00	293	1135	154/151	K	930	930
15	315/70R22.5	18	TL	9.00	312	1014	152/148	M	850	850

Load index											
index	kg	index	kg	index	kg	index	kg	index	kg	index	kg
120	1400	130	1900	140	2500	150	3350	160	4500	170	6000
121	1450	131	1950	141	2575	151	3450	161	4625	171	6150
122	1500	132	2000	142	2650	152	3550	162	4750	172	6300
123	1550	133	2060	143	2725	153	3650	163	4875	173	6500
124	1600	134	2120	144	2800	154	3750	164	5000	174	6700
125	1650	135	2180	145	2900	155	3875	165	5150	175	6900
126	1700	136	2240	146	3000	156	4000	166	5300	176	7100
127	1750	137	2300	147	3075	157	4125	167	5450	177	7300
128	1800	138	2360	148	3150	158	4250	168	5600	178	7500
129	1850	139	2430	149	3250	159	4375	169	5800	179	7750

Velocity ratings							
Code	km/h	Code	km/h	Code	km/h	Code	km/h
A1	5	B	50	L	120	U	200
A2	10	C	60	M	130	H	210
A3	15	D	65	N	140	V	240

A4	20	E	70	P	150	Z	over 240
A5	25	F	80	Q	160	W	270
A6	30	G	90	R	170	(W)	over 270
A7	35	J	100	S	180	Y	300
A8	40	K	110	T	190	(Y)	over 300