Design of a Semi-Trailer for Mining Industry

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ABSTRACT

This work intends to present the project of an innovative semi-trailer with high payload capacity, for the mining industry. In a first stage a market analysis is performed and the goal of breaching a gap in the dumper trucks market is announced. It follows the presentation of the mechanical design which is divided in 5 main categories: tyres, rims and axles; suspension; chassis; dumping container and hydraulic dumping system.

A special emphasis is given to the leaf suspension analysis and some recommended project methodologies are presented. All major structural analysis was performed with finite element software and besides this structural analysis, all the important aspects of this equipment subsystems like the hydraulic system, brake system, dumping door, etc. are scrutinized and justified.

Keywords: Dumping semi-trailer, Mining industry, Leaf spring, Finite element method

1. INTRODUCTION

This project is a result of a partnership between IST and Galtrailer, a company that in the past 12 years has been dedicated to the project and construction of specialized transport equipment, including semitrailers with high payload capacity.

The equipment presented in this project is a dumping semi-trailer destined to the open-air mining industry. This trailer is intended to be used in private roads and has a conventional construction so that it is possible to reduce its production time, also reduce the overall cost and to increase the ease of maintenance. To achieve this type of construction, previous projects were used as a starting point. One of the main innovations presented is the high payload capacity despite of the conventional construction.

1.1. BACKGROUNDS

In this section a brief exposure of the available mining industry market is presented. First, we start by noticing that there are two main categories in ore material transport equipment: articulated trucks and integral trucks. The first ones usually have a payload capacity of less than 40 metric tonnes and offer as a main advantage high mobility in ruff terrain. The second ones offer as a main advantage a high payload capacity usually superior to 100 metric tonnes. Table 1 shows the main characteristics of some integral trucks from a worldwide recognized manufacturer and Table 2 shows the same characteristics for articulated trucks.

Table 1 - Dumper trucks, Caterpillar © 2018 [1]

	785D	***785G***	***789D***
POWER [kW]	1005	1193	1468
PAYLOAD [METRIC TONNE]	129,7	164,8	225,2

Table 2 - Articulated dumper trucks, Caterpillar © 2018 [2]

	745	**730C2 EJ**	**740 EJ**
POWER [KW]	370	274	370
PAYLOAD [METRIC TONNE]	41	28	38

From this basic market analysis, we can say that our solution is innovative as there is no offer of a semitrailer option in todays market. One probable cause is that the transportation vehicles in the mining industry represent a very small fraction of the total investment cost. For this reason, the mining companies attribute more value to the dumpers high payload capacity and reliability and neglect their high cost. On the other hand, a conventional semitrailer option represents lower costs of production, maintenance and service, which can be appealing to small mining explorations. This solution also offers a higher level of service because when there is a problem in the truck the semitrailer can be uncoupled and used by another truck (and vice versa). Moreover, this solution supposes the use of road legal trucks which are built accordingly to tight environmental requirements that lead to a continuous reduction of fuel consumption and emission of pollutants gases. Whereas dumpers have a less rigorous regulation despite their intensive use. As a result, we can state that the presented solution as a higher efficiency.

1.2. Project Requirements

With the previous market analysis in mind we can stablish the necessary project requirements. We intend to develop a 3-axle dumping semitrailer with a payload capacity of 84 metric tonnes (roughly 100 metric tonnes of total weight). The construction must be robust in order to support the rough conditions present in mining fields. To achieve this type of construction it is necessary to use high quality materials and robust solutions such as a mechanical leaf suspension and drum brakes.

We establish a maximum speed of 40 km/h suitable for a safe transportation of ore material.

2. Design

In this section we present the methods behind the structural dimensioning and components selection.

2.1. Tyres, Wheels and Axles

The tyres are one of the most important components as they establish an upper limit in the equipment payload capacity. This is the reason why they are the first component to be defined. From the regular company supplier’s catalogue, we selected the tyre with the highest load capacity. Table 3 presents the specifications of the selected tyre.

Table 3 - Tyre specifications [3]

Designation	**Load Index/Speed Index**	**Wheel Width [in]**	**Load Capacity at 9 bar [kg]**
445/65 R22.5	169/K	14	11600

From Table 3 the maximum load allowable per axle is 11600 kg (2 tyres per axle). However, this capacity is defined for a maximum speed of 100 km/h (speed index K). Adjusting the speed to 40 km/h we obtain 15% more capacity resulting in 13340 kg per axle.

As we intend to use double tyres (4 tyres per axle) the load capacity limit is approximately 25000 kg per axle.

To use the previously selected tyre in a double setup we need to produce a special wheel that can withstand such high loads. The production of this wheel represents a problem because of the high thickness required for the wheel's centre disk. Most wheel suppliers don't have equipment that allows the metal forming of this thicker steel plate. Despite this difficulty we were able to find a supplier capable of manufacture the specified wheel, however, the maximum offset required for a double tyre setup that results from the forming process is 248 mm. Accounting the maximum tyre dimensions we verify that this offset is small, yet acceptable, with enough margin of space between the two tyres (Figure 1).

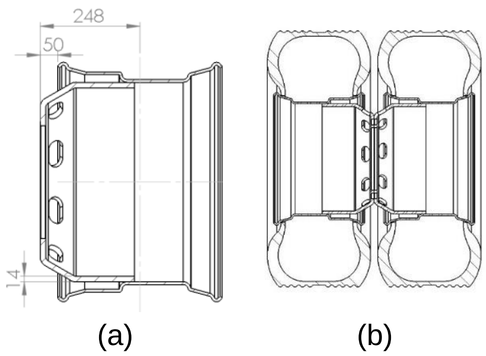


Figure 1 - (a) 14' Wheel with 248mm offset (b) Double tyre setup

The axles are also a very important component, and like the tyres, they impose a limit in the equipment's payload capacity. It is important to select an axle with a load capacity near the capacity of the tyres, to fulfil all its potential. The braking system is associated to the axle selection as they are sold together. Drum brakes offer more reliability in rough environments as they operate in a closed set. In contrast, disk brakes are exposed and can be damaged or contaminated with fine dust. For these reasons we selected an axle with a load capacity of 24000 kg at 40 km/h and equipped with drum brakes (designation SAE-SMB P204220CI).

2.2. SUSPENSION

Mechanical leaf suspensions have been traditionally used over hydraulic suspensions for high load applications due to their superior load capacity. Furthermore, they can be used as a structural member and offer reduced needs of maintenance making them the ideal choice for this type of application.

For this equipment a semi-elliptical leaf spring with plain end mountings was selected. The suspension sets on rotating supports that allow the load to be evenly distributed trough all the axles. It is assumed that the nominal load for the dimensioning of the suspension system is 12000 kg (half of the axle maximum capacity).

The first parameter to have in mind is the vertical stiffness. Spring deformation is directly related to comfort as well as geometric constraints. From the spring manual (by the Society of Automotive Engineers [4]), we can determine the standard intervals for static deflection (with nominal load) and for the additional margin required by dynamic loads. From Table 4 we determine that for the off-road case, static deflection shouldn't exceed 175 mm and that the additional margin needed for dynamic loads is, at most, 125 mm.

Besides spring deflection, we should also verify if the maximum stress present during deformation is below the yield strength of the material. Typically, leaf springs use a hardened steel with a high yield limit (above 1000 MPa). The stresses can derive from both vertical loads and braking efforts that will induce an “S” shape deformation called wind up.

Table 4 - Standard values for leaf spring deflection [4]

	Static Deflection (mm)	Margin for Dynamic Loads (mm)
Buses	100 – 200	50 – 125
Trucks (regular roads)	75 – 200	75 – 125
Trucks (off road)	25 – 175	50 – 125

For this type of projects, there is no practical way to perform a fatigue analysis because the equipment can be used in a variety of scenarios with unpredictable dynamic loads. To perform such analysis, it would be necessary to measure the amplitude of loads in real case applications making project cost and time increase significantly. To ensure that safety requirements are fulfilled, Grubisic and Fischer [5] and Grubisic [6] determined load factors that, accordingly to type of road use, ensure that stress levels stay below the fatigue limits.

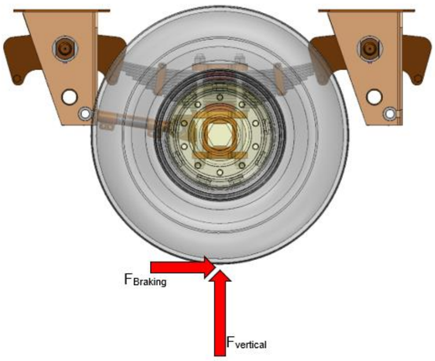


Figure 2 - Contact patch loads

The load factor a corresponds to a case of straight ahead driving and is given by equation 1.

$$F_{vertical} = a \cdot P \tag{1}$$

The load factors b and c correspond to an aggressive braking situation combined with vertical loading and are given by equations 2 and 3.

$$F_{vertical} = b \cdot P \tag{2}$$

$$F_{braking} = c \cdot P \tag{3}$$

Where *P* is the nominal vertical loading (typically one half of axle capacity).

The vertical displacement and maximum stress are given by eq.’s 4 and 5 accordingly to the SAE’s Spring Manual [4].

$$\sigma = \frac{P \cdot l \cdot h}{2 \cdot \sum I} \tag{4}$$

$$\delta = \frac{P \cdot l^3}{2 \cdot E \cdot SF \cdot \sum I} \tag{5}$$

$$\sum I = N \cdot \frac{b \cdot h^3}{12} \tag{6}$$

Where l is the span, h is the leaf thickness, E is the Young modulus, N the total number of leaves and P nominal vertical loading.

The project of this leaf spring suspension uses as reference an existing spring designed by Galtrailer with 13 leaves, 14 mm of thickness and 100 mm of width. For production reasons it was intended to maintain as much as possible the characteristics of the previous spring. From eq.’s 4, 5 and 6 we have opted for 13 leaves with a thickness of 14 mm and a width of 120 mm. The results are shown in Table 5.

Table 5 - Analytical calculations for 13 leaves with a cross section of 14 x 120 mm and nominal load of 12 tonnes

Cross Section [mm]	Stress [MPa]	Wind up Stress [MPa]	δ [mm]	K [N/mm]	SF
14 x 120	684	532	66.3	1775	2

The analytical calculations previously presented have some limitations since they represent a simplified approach that neglects important parameters such as the pre-load on the central bolt that maintains the leaves together and the friction between leaves. In this case, friction as a beneficial effect as it is responsible for providing the damping that will attenuate the indefinite oscillation of the spring (like regular shock absorbers).

To obtain a better model a linear static finite element simulation was created using the software Siemens NX 12.

In a first stage and to save computation time, a simpler model was created. This model was made only by three leaves and served the purpose of defining the element type, it's size and the number of elements along the thickness. Figure 3 illustrates the geometry and boundary conditions used in the simulations.

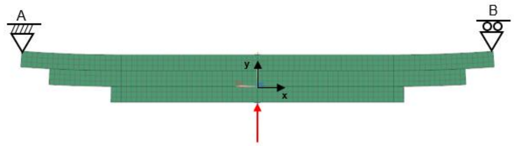
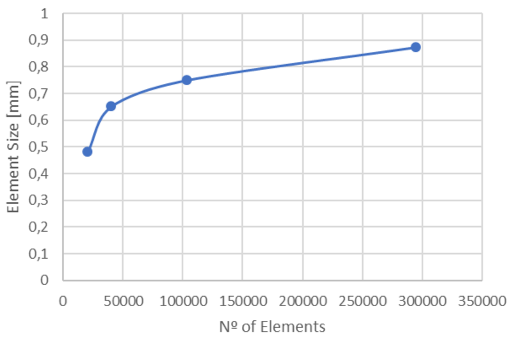


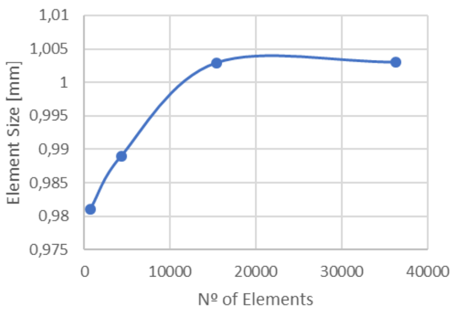
Figure 3 - Simplified Leaf spring finite elements model

The Coulomb friction model was used with a static friction coefficient of 0.3 as indicated in the works of Younesian, *et al*. [7] and Kong, *et al*. [8]. The central bolt was simulated through a 1D rod element with a pre-load of 50kN.

With this model it was possible to conclude that the best type of elements were 3D linear interpolation hexahedrons. This type of elemet present a converged mesh with a significantly lower number of elements when compared with tetrahedrons. Additionally, they have a good performance in contact problems [9]. For the selected type of element, the size that results in a converged mesh is 7 mm (2 elements along thickness).



(a)



(b)

Figure 4 - (a) Mesh convergence for linear tetrahedron elements (b) Mesh convergence for linear hexahedron elements

Figure 4 shows the mesh convergence for tetra and hexa elements. Notice that for the hexa elements, convergence is achieved with a significative lower number of elements. Moreover, tetra elements artificially increase the rigidity of the structure as for the same element size they present inferior displacements. This occurs because these elements have less nodes (less degrees of freedom) and the shape functions are of an inferior degree.

Once experience was obtained with the simplified model a more complex one was created. This time all the 13 leaves were included. The increased number of contact surfaces was responsible for the appearance of some convergence problems. These problems were attenuated with the implementation of different boundary conditions. Comparing with Figure 3 in point A a sliding support was used and in the central hole a cylindrical constraint was created. Additionally, some nodes of the XY exterior surface were constraint to prevent individual rotation of the leaf springs. Figure 5 and Table 6 present an overview of the finite element model.

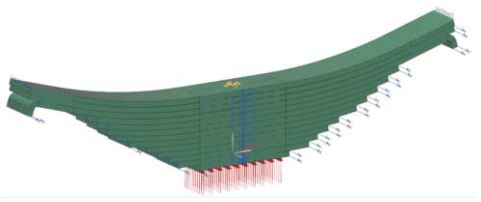


Figure 5 - 13 Leaves finite element model

Table 6 - Finite element model summary

Type of Element	CHEXA (linear hexahedron)
Elements Along Thickness	2
Total Number of Elements	49864
Total Number of Nodes	79932
Material	AISI_Steel_1005

Figure 6 presents the displacement result for the nominal load of 12000 kg. However, the maximum result presented in the colour scale is not the true displacement since there are small gaps between the leaves that will progressively get eliminate as the spring deforms. Therefore, inferior leaves present displacements larger than in reality. The true displacement is the one presented by the top leave and it as the value of 77 mm resulting in a vertical rigidity of 1514 N/mm. This rigidity will be used later as a boundary condition for the chassis finite element simulation.

Figure 7 presents the Von Mises stress result. There is no confidence in the maximum stress values as they urge from contact force peaks located in regions were there is penetration between leaves and therefore, some computational error. Observing the stress along the top leave, which is the expected region for maximum stress, it is possible to conclude that the maximum value is 844 MPa. This stress value represents a 1.7 factor of safety.

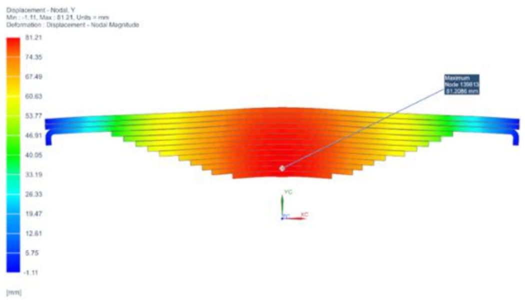


Figure 6 - Displacement result for the 13 leaves finite element model

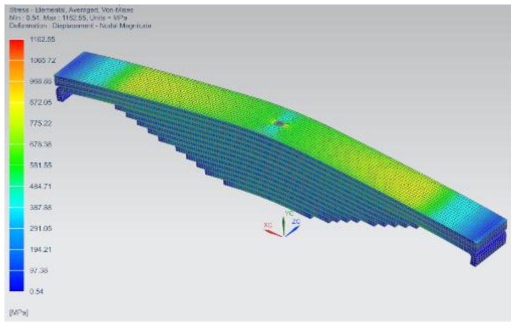


Figure 7 - Von Mises stress result for the 13 leaves finite element model

In Table 7 a comparison between the theoretical model and the computational one is presented. The difference between the theoretical model and the computational one in terms of vertical displacement is 8% and in terms of vertical rigidity is 9%.

Table 7 - Comparison between theoretical model and finite element simulation

	Load [kg]	***Vertical Displacement [mm]***	***Rigidity [N/mm]***	***Max. Stress [MPa]***
Computational Model	5000	37	1325	372
Theoretical Model	5000	34	1443	342

2.3. CHASSIS

The chassis is a very important component of this equipment and it should be in conformity with the standard ISO 1726 (2000-05-15) [10]. This standard defines the neck shape of the chassis and ensures that the trailer is compatible with the majority of trucks. Figure 8 illustrates the shape defined by the standard.

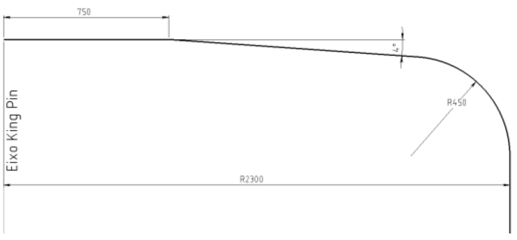


Figure 8 - Neck shape according to ISO 1726 (2000-05-15) [10]

The chassis frame structural analysis was performed trough finite element analysis with the software Siemens NX 12. 2D linear elements (trias and quads) were used since they produced a better representation of the structure due to the aspect ratio of the chassis components and because they reduced drastically computation time. Four load cases were considered: case I is simply vertical loading from the cargo weight (95 ton); case II is torsion produced in cornering by a lateral acceleration of 0.6g; case III is a combination of the previous cases and case IV represents a situation where the equipment is unloaded and supported by the landing gears. We concluded that case III is the most critical mainly because of the lateral acceleration. Figure 9 shows the boundary conditions used in case III, Figure 10 presents the global displacement results (in mm) and Figure 11 shows the Von Mises stress (in MPa).

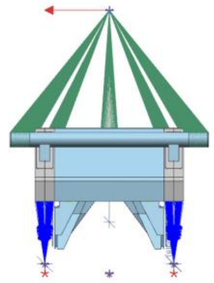


Figure 9 - Boundary conditions for load case III

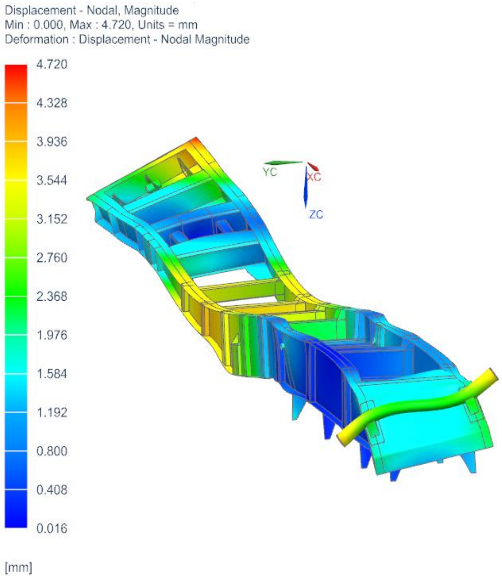


Figure 10 - Displacement result for load case III [mm]

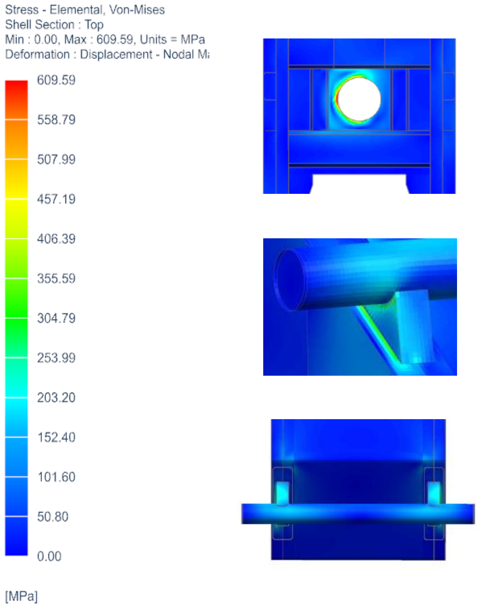


Figure 11 - Von Mises stress result for load case III [MPa]

The high stresses (610 MPa) presented in Figure 11 are due from the boundary condition applied to a circular edge in order to represent the king pin support. Here RBE2 elements constraint all translational DOFs. In reality, this load is distributed to a larger area (the fifth wheel plate) which contributes to a significative reduction of the stress value. Additionally, the king pin plate where these high stresses occur is made of a hardened steel with high yield strength. For these reasons such stresses were ignored and the focus went to the next highest value (318MPa) that occurred in the rotation point of the dumping container. Table 8 shows the results obtained from the chassis finite elements simulations.

Table 8 - Summary of the simulation results

Load Case	***Max Displacement [mm]***	***VM Stress [MPa]***	***SF***
I	3.75	210	2.2
II	4.55	318	2.0
III	4.72	318	2.0
IV	9.16	90	5.2

Once the structure of the semitrailer is defined we can perform an analysis to verify that the axles are not in an overload situation. The model to obtain the axle reactions considers that the load is evenly distributed between all three axles (Figure 12). Eq's 7 and 8 where used to obtain both axles and king pin reactions.

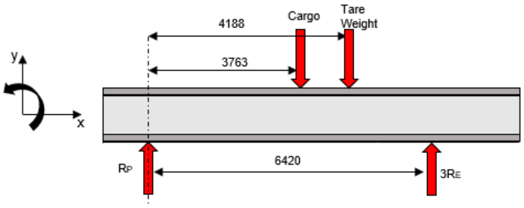


Figure 12 - Chassis beam model

$$\sum M_P = 0 \Leftrightarrow 3 \cdot R_E = \frac{C \cdot 3.763 + 17 \cdot 4.188}{6.42} \quad (7)$$

$$\sum F_y = 0 \Leftrightarrow R_P = C + 17 - 3 \cdot R_E \quad (8)$$

Where R_E represents the axle reation, R_P the king pin reaction, C the cargo weight. With a tare weight of 17 tonnes we obtain a reaction of 20 tonnes in each axle. Considering that the maximum allowed value is 24 tonnes we can conclude that the axles are safe. The king pin reaction is 41 tonnes and this value is distributed between the truck axles. For this reason, it is necessary to select an adequate truck. For high load applications it is common to use a 6x4 truck (3 axles with 4 driving wheels). The axle capacity for these trucks are roughly around 20 tonnes so we can conclude that the king pin reaction is not excessive.

2.4. Dumping Container

The width of the dumping container should not be larger than the width of the axles. The maximum defined value was 3200 mm.

Two types of dumping container were designed. The first one has a flat ground and it is destined to transport fine materials. Figure 13 illustrates this version of the equipment.

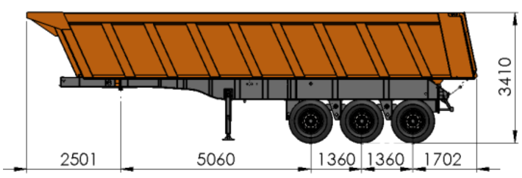


Figure 13 - Flat dumping container version

For the transport of large rocks, a version with an inclined bottom was created. This version is represented in Figure 14 and can be supplied with or without the back gate.

Both dumping containers offer a total volume of 46 m^3.

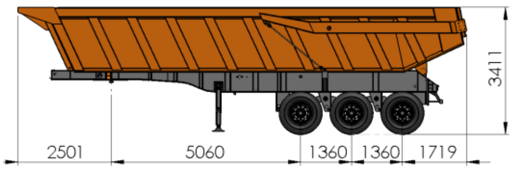


Figure 14 - Inclined dumping container version

2.5. Hydraulic Dumping System

The hydraulic dumping system is comprised of a pump, a hydraulic cylinder, a valve and a controller. It is necessary to design the system so that it can lift the required load and that the actuation length is enough to fulfil the defined dumping angle. For each type of material there is a suitable dumping angle at which all the cargo present in the container can be dumped to the exterior. If we go under that angle some material will remain stuck in the container and can create potentially dangerous situations. If we lift the container with stuck cargo, the centre of gravity can be shifted to an unstable configuration that can make the equipment roll over.

For the majority of ore material, the necessary dumping angle is 50º. From geometric calculations we derived that to fulfil this angle, we need a hydraulic cylinder with a total actuation length of 9450 mm. From the company's suppliers we selected a cylinder with suitable lifting capacity and with an actuation length near the value defined previously. Table 9 presents the technical specifications for the selected cylinder.

Table 9 - Technical specifications for the selected hydraulic cylinder [11]

Designation	***Lifting Capacity [ton]***	***Total Actuation Length [mm]***
FE A 191-6-08200-011-K1672	24-33	9872

Knowing that the combined weight of the dumping container and cargo is around 100 tonnes we can concluded that 2 cylinders are needed. Notice that the hydraulic cylinders need to lift around half of the total weight since the cargo is distributed between the cylinders and the rotation point. Since there are two cylinders it is also necessary to add a flow divider valve in order to correct any unbalance between the cylinders and avoid the overload off one cylinder.

3. Conclusions and Future Work

The equipment presented in this paper shows an innovative alternative to the traditional dumper trucks. With a conventional construction it is possible to reduce the costs of production, service and maintenance, pointing out the fact that a semitrailer solution offers higher level of service. Other advantage is the higher efficiency associated with the use of road legal trucks regulated by tight environmental laws.

Since dynamic loads are not well known and depend largely on the type of roads the equipment is destined to, fatigue analysis is not easy to perform. Estimation of this type of loads would require the existence of a similar equipment to perform acquisition *in loco*. This would increase significantly the project cost and duration. Therefore, load factors and increased safety factors are used to ensure that stress levels stay under the fatigue limits.

From the suspension analysis it was possible to conclude that the analytical formulation provides accurate results and it is a good approach for an initial stage. Nevertheless, the current practice in the semitrailer industry is to confirm the performance of a leaf spring with experimental tests performed by the suspension manufacturer. FEM models can present good results like in the cases of Karditsas, *et al*. [12] and Tank, *et al*. [13]. However, in cases where there are a large number of contact surfaces convergence problems may occur. In a future work a 2D approach could be explored as it could result in better convergence.

In the chassis finite element simulation, it was possible to conclude that the lateral acceleration is the critical load situation. The value of 0.6g comes from a static calculation, and it is the point at which the equipment would roll over.

To conclude, it is important to state that some of the design decisions derived from the experience accumulated by Galtrailer in its 12 years past. We have privileged modular solutions that can be used in various types of equipment, reducing project time and consequently reducing cost.

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