Alternative traction system for road-rail vehicles

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Alternative traction system for road-rail vehicles

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Abstract

Excavators have traditionally been diesel driven with hydraulic actuators, but with the ongoing electrification of vehicles, smaller electric excavators are getting more common. Larger electrical excavators are still in a prototyping or early production stage but can be expected to grow. The electrification of construction equipment creates new possibilities to replace previous hydraulic- with electric actuators.

This thesis covers the product development of an electric tractive system for a road-rail excavator using Ulrich and Eppinger´s method for product development. Current hydraulic actuators have different speed, torque and size characteristics while following other safety regulations compared to electrical motors. These differences create the need for a product development project to investigate potential problems and possibilities in the change from a hydraulic- to electric tractive system.

The project was made at CE Engineering Solutions (CEES). Different technical standards were studied to simplify a possible CE-marking of the product and the technical specifications were set using comparative data from competitors as well as data and knowledge from previous projects within CEES. The solutions were restricted to the use of components available today in small order quantities since the initial target was building and verification of a prototype. In the thesis, multiple concepts were generated and three different concepts with two different working principles are presented more in detail. A final solution was selected by concept scoring with weight factors decided together with CEES.

The final solution consists of a single motor, inverter, and reduction gear per wheel axle due to cost efficiency, differing from the more recent solutions at CEES with two independent motors per axle. The limitation of using “off the shelf” products available today resulted in a solution using an industry standard gearbox not fully adapted for the high-speed electrical motors often used in mobile applications, restricting the traction performance but resulting in a more cost-effective solution.

Keywords: Product development, electric traction, construction equipment, road-rail vehicles.

Sammanfattning

Grävmaskiner har traditionellt sett varit dieseldrivna med hydraulik men med den pågående elektrifieringen av fordon börjar mindre elektriska grävmaskiner bli vanligare. Större elektriska grävmaskiner finns idag främst i tidigt/prototyp-stadie men förväntas att bli fler. Elektrifieringen av anläggningsmaskiner skapar nya möjligheter för att ersätta tidigare hydrauliska funktioner med elektriska dito.

Det här examensarbetet beskriver produktutvecklingen av en elektrisk drivlina för väg- och rälsgående grävmaskiner med hjälp av Ulrich och Eppingers metod för produktutveckling. Dagens hydrauliska motorer har en annan karaktäristik gällande hastighet, vridmoment och storlek jämfört med elmotorer samtidigt som de följer andra standarder och regleringar. Sammantaget skapar detta ett behov av att undersöka potentiella problem och fördelar med att ersätta dagens hydraulikbaserade drivlina med en elektrisk.

Arbetet genomfördes tillsammans med CE Engineering Solutions (CEES). Inom arbetet studerades olika tekniska standarder för att förenkla en möjlig CE-märkning av lösningen. De tekniska specifikationerna för lösningen bestämdes med hjälp av jämförelsedata från konkurrenter samt data och erfarenhet hos CEES. De framtagna lösningarna begränsades till att enbart innehålla komponenter tillgängliga idag i små kvantiteter för att underlätta framtagandet av en eventuell prototyp. Inom arbetet togs ett flertal olika koncept fram och tre av dem, varav två med liknande arbetsprincip, är presenterade i detalj. En slutgiltig lösning valdes genom en viktad poängmatris där viktningen bestämdes tillsammans med CEES.

Den slutgiltiga lösningen består av en motor, växelriktare och reduktionsväxel per hjulaxel på grund av kostnadsskäl vilket skiljer den från de senare lösningarna från CEES vilka alla haft separata motorer för varje hjul. Begränsningen i att använda standardprodukter gav en lösning med en växellåda av industrimodell som inte till fullo harmoniserar varvtalsmässig med motorerna i dagens elektriska fordon. Detta innebär att transporthastigheten är begränsad av växellådan men skapade också en mer kostnadseffektiv lösning utifrån förutsättningarna.

Nyckelord: Produktutveckling, Eldrift, Anläggningsmaskiner, Tvåvägsfordon.

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Lund, February 2023

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1 Introduction

1.1 CE Engineering Solutions and the project

CE engineering solutions (CEES) is an engineering and manufacturing company, member of the Volvo group, working with special application solutions to Volvo construction equipment. The solutions include adaption of excavators, wheel loaders and haulers for applications such as tunneling, slag handling, heavy industries etc. One of the adaptions is modification of excavators for usage on both road and rail (road-rail machine/vehicle), commonly used for railway maintenance, Figure 1.

Figure 1 A road-rail excavator manufactured by CEES.

Excavators have traditionally been diesel driven with mainly hydraulic actuators. However, manufacturers are now starting to investigate electrification of the machines in the development of moving from fossil fuels ("Change starts here", n.d.). Larger electrical excavators are still in a prototyping or early production stage but can be expected to grow (Doyle, 2021). Since the currently available electrical excavators use traditional hydraulic actuators, it is possible to keep the conventional traction system of the machines. However, this project aims at investigating the possibilities and limitations of replacing the currently hydraulic driven transmission system of category 9A road-rail excavators with an electrical traction system, see chapter *3.1.1- Road-rail machines – General.* This will be done by developing an electric traction system adapted to an electrical base machine.

Previous products developed and manufactured at CEES, including hydraulic- and indirect driven road-rail construction equipment will act as a foundation to this project.

1.2 Project goal, process and scope

1.2.1 **Project goal**

The goal of the project is to develop an electric traction system for road-rail vehicles using technology and components available today. The solution must be adapted to be used with a 30-ton machine and it must be possible for the solution to fulfil relevant national and international regulations and specifications for road-rail vehicles.

1.2.2 **Process**

Since a product developing process is characterized by lots of uncertainties in its initial state, two different routes were set up prior to the project. Both routes included a first phase with literature study and research of solutions used today, thereafter a process of writing product specification, concept generation and lastly concept selection and evaluation. The first phase was planned for approximately half the project time and would end in a design review together with CEES and supervisors. After finishing the first phase, a decision would be made regarding the project continuing with either phase two or phase three.

The project would preferably continue with phase three, including preparation of concept for full scale prototyping, test of prototype and lastly evaluation of solution and inspection. If phase three is to be deemed impossible after the design review due to either complexity, time restriction or lead time for components, phase two would follow the first phase. Phase two would include further refinement and development of concept with finite element analysis (FEA), 2D-drawings and lastly a verification to the system specifications. The different routes and phases are illustrated in Figure 2.

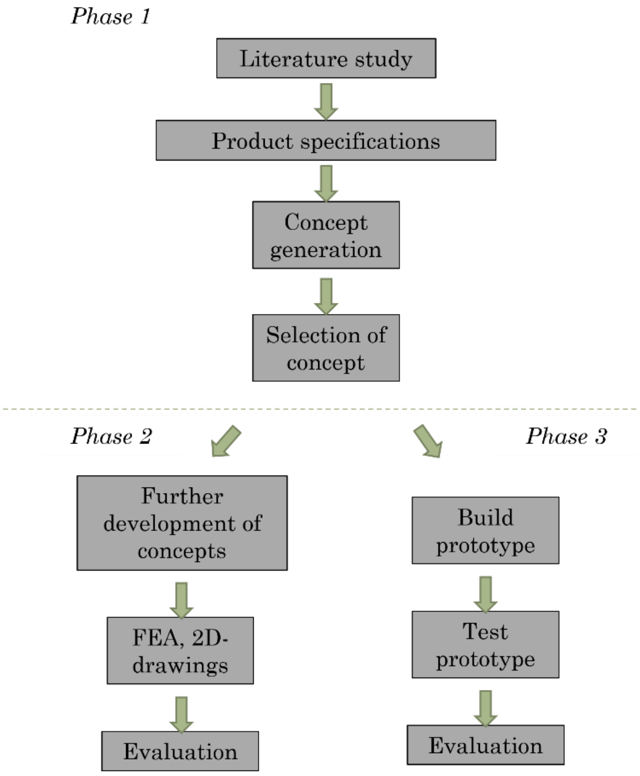


Figure 2 The planning of the project with three different phases in two routes.

1.2.3 Scope

The focus of the project will be on the electric drivetrain including technical requirements, packaging, and selection of concept with respect to several different criteria's such as price, robustness, performance, integration to Volvo CE. To make it able to sell the product within the EU, the product needs to fulfil the relevant EU directives. Therefore, the project will also include a literature study of different applicable technical standards to verify compliance and simplify CE-marking.

Other aspects concerning road-rail vehicles such as the number of rail wheel axles, diameter of rail wheels, mounting to base machine etc. is out of scope and will rely on previous solutions at CEES.

Due to the product being produced in small series and possible spare parts needed in the future, the design is limited to components available off the shelf.

2 Methodology

This section describes parts of Ulrich and Eppinger's (U&E) product development process and how it was adapted for this thesis.

2.1 Ulrich and Eppinger's product development process

As described in the project introduction, this project aims to develop an electric tractive system for road-rail machines. U&E describes a generic product development process consisting of six phases. (Ulrich & Eppinger, 2012, pp. 13-14) The phases are illustrated in Figure 3.

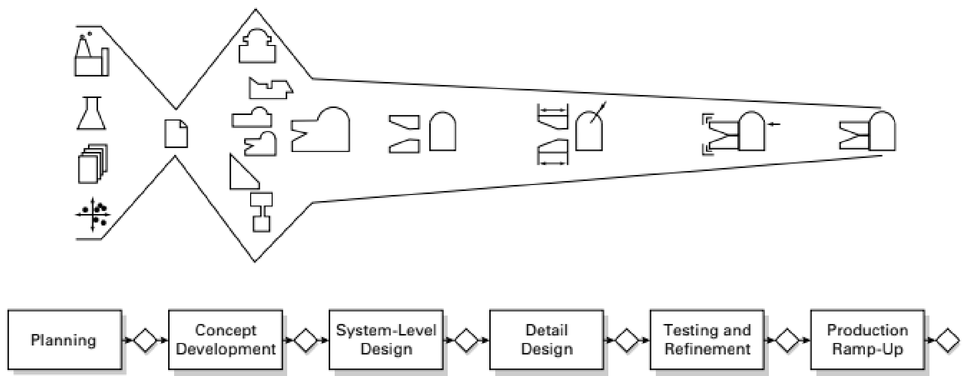


Figure 3 The six phases of the generic product development process according to U&E.

The generic product development process by U&E was chosen for this project due to the structural methods with a step-by-step approach. The approach reduces the risk of moving forward in the process with unsupported decisions and acts as a checklist in the development process, simplifying the documentation of the project.

Presented in the scope of this thesis, this project will mainly focus on the development, selection, and evaluation of concepts, activities mainly within the concept development phase presented in section *2.1.1 Concept development.* Therefore, will the focus of this project be on this phase.

2.1.1 **Concept development**

The concept development phase can be divided into several activities. Even though the activities presented in Figure 4 have a specific order, the concept development phase is generally an iterative process, and the activities may overlap in time according to U&E (Ulrich & Eppinger, 2012, p. 16).

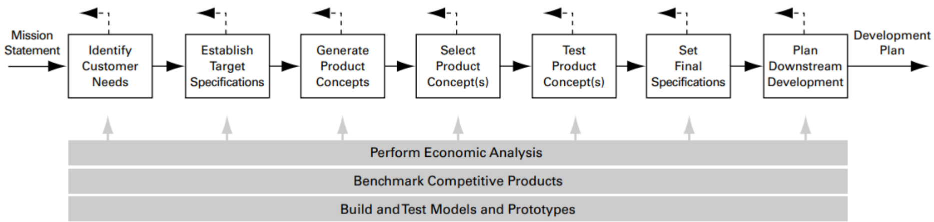


Figure 4 The activities comprising the concept development phase according to U&E.

Establish target specifications, generate product concepts, and *select product concepts* are activities that lie within the scope of this thesis and the methodology and process are described thoroughly in the report under section 4-6. The other activities are covered more briefly.

3 Literature study and state of the art

This section describes the initial process to get an understanding of previously used solutions, technical requirements, and customer needs. The process was mainly carried out using three different types of sources: product brochures/websites, technical national and international standards, and patent search for different road-rail machines. The study raised questions about electrical- and regenerative braking possibilities which were briefly studied.

3.1 Road-rail machines – general

3.1.1 **Road-machine categories**

To get a basic understanding of rail-road vehicles and as a preparation for further studies, the three main categories of road-rail vehicles were studied. Different regulations apply to different categories of vehicles and an understanding of the categories was crucial for understanding the following regulations and standards.

Today's road-rail vehicles, that cannot be incorporated into a train, are divided into three different categories depending on type of propulsion, class 9A/B/C. The different categories are defined in the standard EN 15746-1 (European Standard EN 15746-1, 2020).

3.1.1.1 Road-rail machine category 9A

The 9A category includes machines with braking and traction directly on rail wheels, today often with the use of hydraulic motors. The machine load is entirely on the rail wheels. An example of a category 9A excavator can be seen in Figure 5.

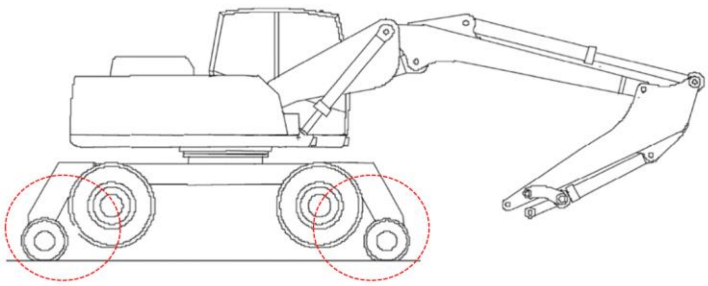


Figure 5 A category 9A road-rail excavator.

3.1.1.2 Road-rail machine category 9B

The 9B category includes machines with traction and braking indirect from road wheels to rail wheels. The machine load is entirely on rail wheels. An example of a category 9B excavator can be seen in Figure 6.

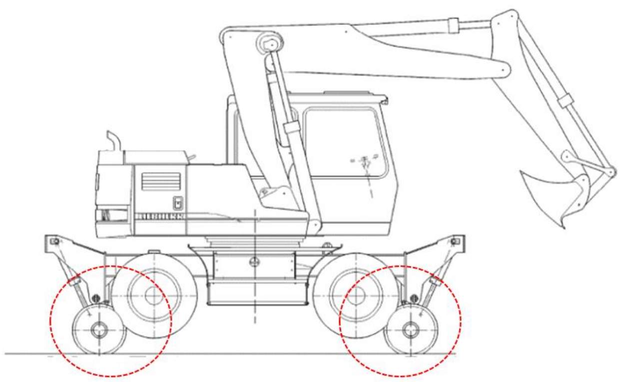


Figure 6 A category 9B road-rail excavator.

3.1.1.3 Road-rail machine category 9C (Low rail)

The 9C category, also called low rail, includes machines with braking and traction on road wheels. The machine load is divided between road and rail wheels. An example of a category 9C machine can be seen in Figure 7

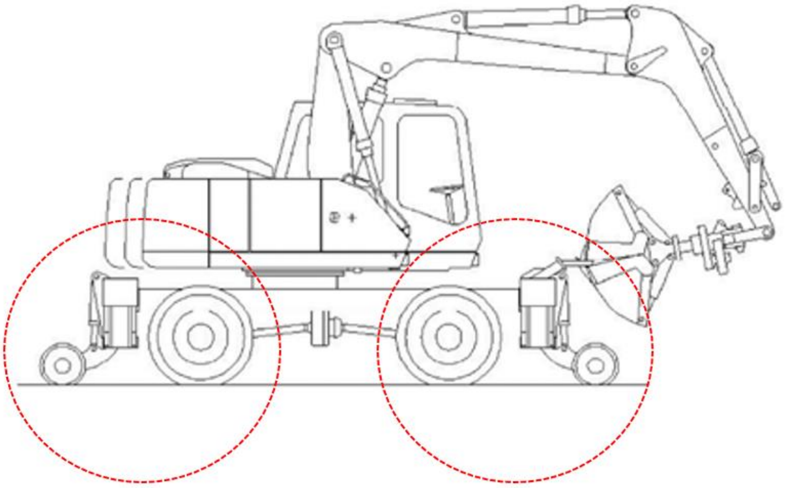
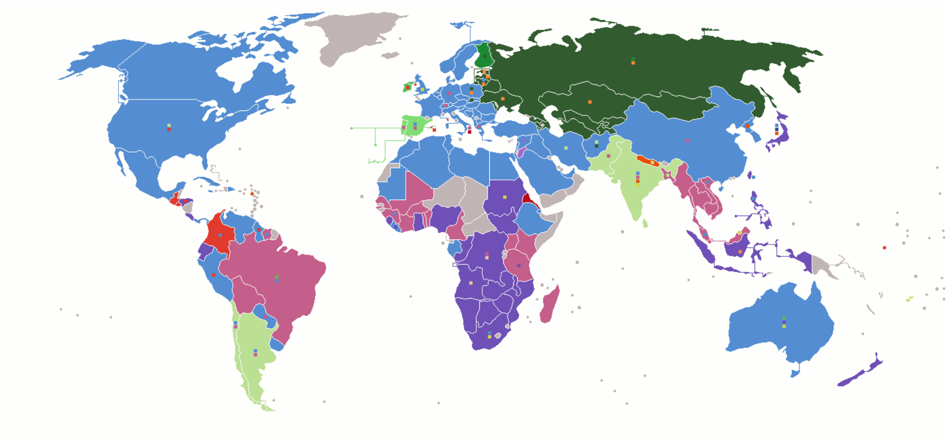


Figure 7 A category 9C road-rail excavator.

3.1.2 **Running gauge**

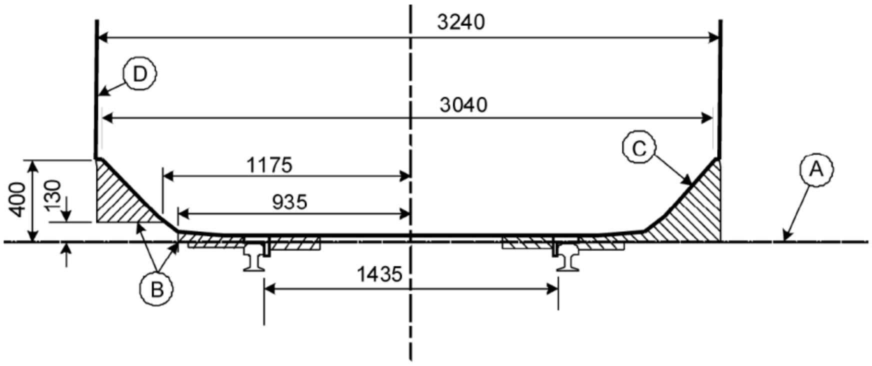
The study continued with studying geometrical restrictions from the standards and regulations. The running gauge is an important factor in specifying geometrical conditions for road-rail vehicles to verify compliance with the infrastructure and to avoid potential damage on both machine and rail. Due to different running gauges around the world, limiting a machine to using one track gauge will limit the potential global market, Figure 8.



mm	1676	1668	1600	1524	1520	1435	1372	1067	1050	1000	950	914	762	750	610	600
ft in	5'6"	5'5.67"	5'3"	5'	4'11.8"	4'8.5"	4'6"	3'6"	3'5.3"	3'3.4"	3'1.4"	3'	2'6"	2'5.5"	2'	1'11.6"

Figure 8 The most common track gauges around the world (Anon., 2023)

The most common track gauge in Europe and globally is 1435mm and the European standard EN15746-1 specifies allowed traveling gauge for road-rail machines. Road-rail machines are permitted to exceed the allowed gauge for rail vehicles and the allowance is dependent of the road-rail vehicle category. The permitted gauge can be seen in Figure 9.



Key

A = Rail level
B = Exceedance of gauge permitted for road-rail machines
C = Additional exceedance of gauge permitted for Category 9 C machines
D = Gauge according to EN 15273-2

Figure 9 Traveling gauge limits for 1435 nominal gauge according to EN 15746-1. Dimensions in mm.

Exceedance of gauge in zone B or C in Figure 9 is only allowed under the condition that the machine does not damage the infrastructure, such as railway switches and retarders. In addition to the regulations in Figure 9, special national conditions apply in some countries, see Table 1.

Table 1 Countries with special national conditions for track gauge limits according to EN 15746-1

Country	Specific regulations for track gauge
Netherlands	Cat. 9C machines are forbidden with some exceptions, e.g. specific tire sizes required etc.
France, Sweden	No exceedance of gauge into zone B or C allowed.
Finland	1524mm nominal track gauge. For cat 9B, maximum rail wheel width: 140mm.
Germany	Specific tire sizes and tire pressures are required for cat. 9C machinery. Wheels with tires are to be lifted 100mm.
Great Britain	Separate gauge limits restricted by British ‘plant gauge’.

Trafikverket are responsible for the Swedish railway and have therefore specified a national allowed traveling gauge in the TDOK 2015:0143. The Swedish national requirements are similar to the EN 15746-1 requirements but are more detailed in some respects, e.g. the area closest to the rail, see Figure 10

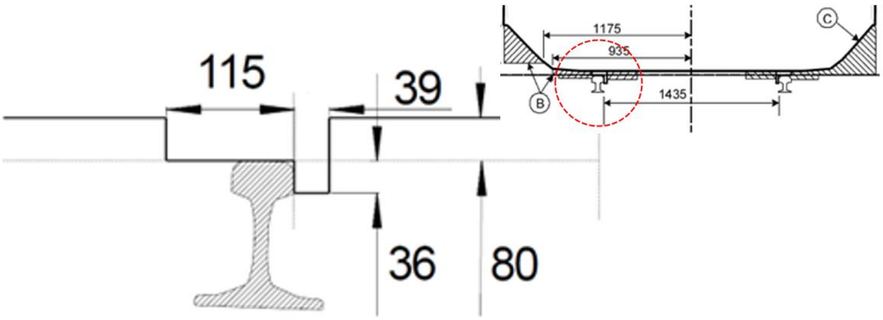


Figure 10 Allowed traveling gauge closest to the rail for working machines according to TDOK 2015:0143. Note: Minimum required ground clearance increases from 80 to 130mm when passing an active railroad retarder.

3.2 Technical standards

To simplify the CE-marking process and to fulfil the related EU directives, large amounts of regional and international technical standards are available to aid in the product development and manufacturing. The standards act as a guideline during the design process but aren't mandatory to fulfil. Since electrical construction equipment and their electric traction system is a new technical area, no suitable standard could be found for this application. However, does the EN-15746 cover general road-rail vehicles and was therefore used as a baseline and the reference list of the EN 15746 was used to find other applicable standards often with a more in-depth description for certain areas of technology. For example, the EN-50153 was found in the reference list, giving an in-depth explanation of provisions related to electrical hazards and the section about equipotential bonding was applicable for this type of project. In addition to standards referred to in EN 15746 others, mostly electrical standards, were studied after suggestions from CEES. These additional standards were only briefly studied and did not affect the further product development but can be seen as potential future relevant standards if the product development continues with, for example, the development of electrical motors. The result of the standard study can be found in Appendix B.

3.3 Patent search

The main target of the patent search was investigating prior art to get inspiration for the concept generation. The search was conducted by using a classification search at the European patent office, Espacenet. For the search, the cooperative patent classification (CPC) B60F 1/043F was used for searching patents for road-rail vehicles comprising own propelling units with separate road and rail axles. The classification was used in combination with search terms such as excavator and construction machinery. Below are two of the patents found that acted as inspiration to the project.

3.3.1 Vehicle rail -engaging device

A Chinese patent for a road-rail vehicle was found (Zhang & Yang, 2018). The patent describes a device for railway rail travel for non-rail vehicles such as construction machinery. The patent has multiple similarities with CEES´s previous products such as one rigid and one pendulum wheel axle. A pendulum axle, where the wheel axle is allowed to turn around the longitudinal axle of the vehicle ensures ground contact for all wheels. The device has only one driven axle, driven with two hub motors. The pendulum axle incorporates a dampening mechanism to reduce vibration and to keep the driven wheels in close contact with the rail. This patent was selected due to the different pendulum axle with a squared cross section compared to CEES´s previous solutions. The axle also looked spacious in the area close to the hub motors, possibly offering enough space for multiple different drivetrains. The device can be seen in Figure 11.

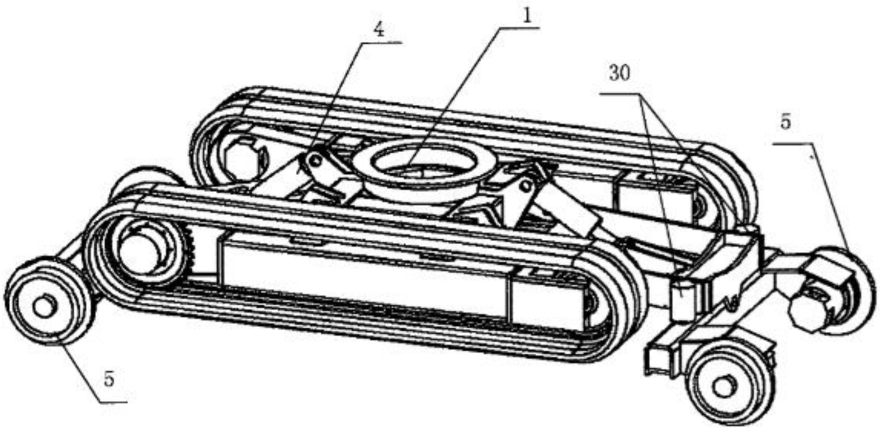


Figure 11 Device with a damped pendulum driven axle.

3.3.2 **Width adjustable railway axle**

A French patent granted by the World Intellectual Property Organization (WIPO). The patent describes a driven railway axle comprised of one or two motors connected to the wheel hubs which has a cylindrical rim with two flanges on its outer face (Lambert & Marchetta, 2020). The rail wheels can be mounted to either of the flanges and by that multiple trackwidths can be achieved. The mounting can be combined with additional shims for finer adjustment and allowing more different trackwidths. This patent was selected due to the simple change between different trackwidths in a modular way with a single axle, offering a single product adapted for multiple regions The solution can be seen in Figure 12 and Figure 13.

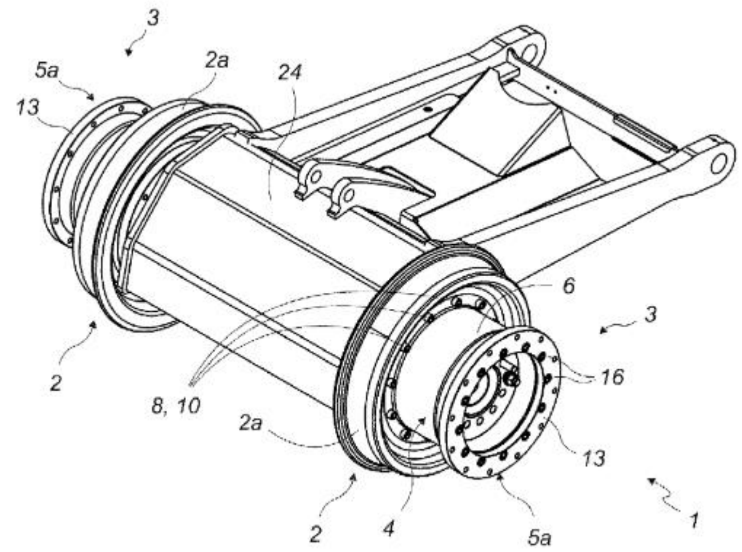


Figure 12 Width adjustable railway axle.

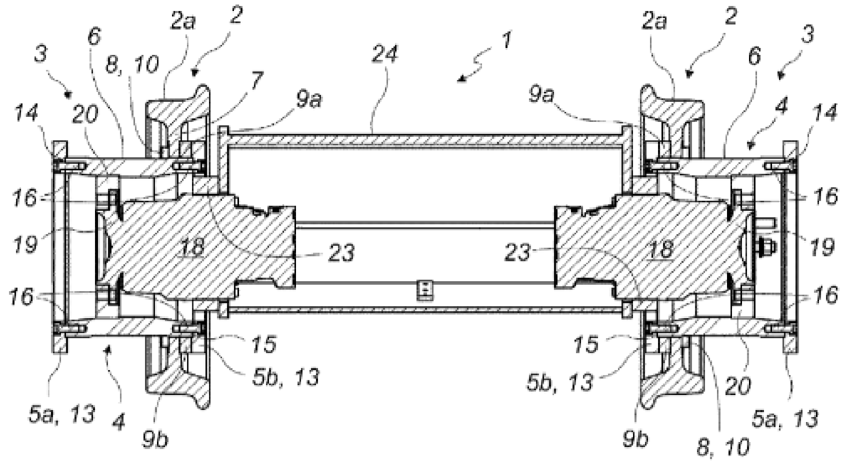


Figure 13 Width adjustable railway axle with two hub motors - cross section view.

3.4 Product brochures

Within the literature study, multiple product brochures, marketing materials and web pages from different manufacturers were studied to get an understanding of currently used solutions, customer needs and benchmarking data. The companies chosen for this part of the literature study were all companies found offering a complete road-rail excavator. Various other companies offer road-rail conversions kit to base machines of various brands but since the performance of these kits, e.g. power to weight ratio, will wary depending on the base machine, these types of solutions were excluded from the study.

In total, marketing material from five different manufacturers of road-rail excavators, excluding Volvo, were studied. The different manufacturers were Caterpillar, Liebherr, Kaiser, Mecalac and Hydrema. The marketing material highlighted in general performance of the base machine such as stability, efficiency, flexibility etc. but some features for railway usage were also covered and can be seen in Table 2.

Table 2 Example of features mentioned in different marketing materials.

Feature	Mentioned in:
Hydrostatic drive	Mecalac, Caterpillar
Possibility to change between track gauges	Liebherr
Dismountable rail units	Hydrema
"Large" rail wheels	Hydrema, Caterpillar, Liebherr, Mecalac
Pendulum axle	Caterpillar, Liebherr

Table 2 aimed at summary different features that the manufacturers choses to promote to get an understanding of customer needs. It does not cover all features for the different machines and there is a possibility that some manufacturers choses to not market a feature in their product brochures, even though the machine might have the feature.

3.5 Electrical braking

The study of technical standards highlighted a need for two separate braking systems, and the study of previously category 9A machines manufactured at CEES showed that the hydraulic motors had been used as one braking system. Because of this, the possibility to use electric motors for generating sufficient braking power by regenerative braking was briefly investigated.

The main advantage of regenerative braking in this application were deemed to be the possibility to completely replace the frictional brake for service braking to simplify packaging, reduce mechanical wear and risk the of failure. Therefore, the targets for regenerative braking were set to comply with the braking specifications from EN15746-2, Table 3.

Table 3 Maximum allowed stopping distance according to EN 15746-2.

Machine speed [km/h]	**Maximum stopping distance on level track of machine and any permitted (by the manufacturer) unbraked trailing load. [m]**
8	6
10	9
16	18
20	27
24	36
30	55
32	60
40	90
50	155
60	230
70	300
80	400
90	500
100	620

The required braking power was calculated using a constant deceleration for a fully loaded machine and maximum allowed stopping distance shown in Table 3. The average required braking power for 20 and 30 km/h was calculated to 48 and 79 kW respectively. The calculated braking power was compared to maximum charging power for a 90-kWh battery used in Volvo compact excavators, Table 4. The braking calculations can be seen in Appendix C.
.

Table 4 Maximum charging capacity for 90kWh Samsung battery designed for compact excavators, data supplied by CEES.

	Max charging power [kW]													
T\SOC	5%	7%	10%	15%	20%	30%	40%	50%	60%	70%	80%	85%	90%	95%
-20														
-10														
0	22	22	22	22	22	20	19	17	10	7				
10	52	52	52	52	51	52	50	41	34	26	16	9	5	
20	92	90	90	82	71	64	59	46	36	32	24	20	15	11
25	107	106	106	95	82	72	66	50	39	36	28	23	18	12
30	118	117	117	109	100	94	87	66	54	48	34	28	22	15
45	95	98	100	103	106	109	102	87	81	67	38	35	29	15

As seen in Table 4, maximum possible charging power is dependent of both initial temperature (T) as well as initial state of charge (SOC) and the required braking power, highlighted in yellow and green, cannot be fulfilled for all battery states. Even though the base machine might not be using a battery with the same specifications as listed in Table 4, the table shows that battery performance can limit the possibility to use regenerative braking and thereby making regenerative braking insufficient to fulfil the requirements. The possibility of using supercapacitors or rheostatic braking for higher power capacity was briefly investigated but discarded due to increased complexity and uncertainties regarding braking precision.

Replacing frictional brakes with regenerative braking was found to be impossible with current requirements and battery technology and therefore must the electric traction system incorporate two braking systems beyond the braking power that the traction motors can provide. Regenerative braking can instead be used in combination with frictional brakes. Electric vehicles are in general equipped with a regenerative-hydraulic hybrid braking system where the hydraulic braking is applied whenever the regenerative braking is insufficient. The hybrid braking system enable maximum energy recovery while keeping the same braking performance as conventional vehicles (Chau, 2014).

4 Establish target specifications

This section describes the process of establishing target specifications based on the previous literature study as well as experience at CEES from previous products. The target specifications were divided into four different groups listed in no particular order:

- *Customer needs – Specifications derived from customer needs.*
- *Regulated specifications – Specifications from various of technical standards.*
- *Integration to base machine – Specifications needed for possible integration of the concept to the base machine.*
- *Company perspective – Specifications that doesn't fulfil any customer need but opens for new markets globally due to different national standards and regulations.*

The target specifications concretize the different demands on the product that were found during the previous sections of this report. Where applicable, the specifications were set with both a marginal value as well as an ideal value. Ranking and weighting of different specifications where done in a later stage, see section 6 Selection of concept and subsection 6.3 Concept scoring. The full list of target specifications can be seen in Appendix D.

4.1 Customer needs

The customer needs were gathered by looking at product brochures and marketing material, discussions with employees at CEES as well as looking at previous investigations in CEES archive. Four customer needs were translated to six measurable metrics, see Table 5.

Table 5 Table of customer needs with corresponding metrics and target values.

Need:	Metric:	Unit:	Marginal value:	Ideal Value
High pulling force	Drawbar pull	[kN]	>40	>60
Fast during transport	Maximum speed	[km/h]	>20	30
Good precision in brake	Subj			
Good traction and stability	One pendulum axle	[Yes/No]	Yes	Yes
	Large rail wheel diameter	[mm]	>500	>600
	Differential	[Yes/No]	No	Yes
Doesn't break when derailing	All parts protected against damage from derailing	[Yes/No]	Yes	Yes

For the process of setting marginal and ideal values, a benchmarking table was made for different machines, four earlier models from CEES as well as competitor products. Due to difficulties finding data for all models, the benchmarking table wasn't complete but worked sufficient for setting the metric values. The benchmarking table is presented in Table 6.

Targets and benchmarking products

		Targets	Road mode		Models for comparison								
Input data	unit	EWR 150/170	EWR150e (On road)	EWR170e (On Road)	Volvo ECR 145 CL (Actuall values)	Volvo ECR 145 CL (Targets)	Volvo ECR 88 (Old model)	Vovlo ECR 88D (New model)	Cat M323F (On rail)	Cat M323F (On road)	Hydrema MX20G Rail	Hydrema MX16G Rail	Mecalac 216M Rail
Machine weight	kg	22000	17900	19200	18210	18210	10000	10000	23900	2.3900	20550	19,45	20350
Wheel diameter	m	0,6	1,07	1,07	0,7	0,7	0,4	0,5	0,632	1	0,65	0,65	0,63
Drawbar pull	N	40000	88000	111000	49000	35000	11342	21943	45000	104000			
Maximum speed	km/h	30	35	35	20	20	17,8	9,2	20	20			30
Gross power	kW		105	115	109	109	43,8	43	117	117			100
Nbr of driven wheels		4	4	4	4	4	2	4	4	4	4	4	4
Ground clearance (lowest point to rail)	mm	130			165		75	128	195	360	100	100	
Results:													
Pullin ratio (F/m)		0,19	0,5	0,59	0,2	0,2	0,12	0,22	0,19	0,44			
Wheel speed	RPM	265	174	174	152	152	236	98	168	106			253

Table 6 Benchmarking table for different machines from CEES as well as competitor products.

The highlighted values were used for comparison for setting marginal and ideal value.

Using the benchmark table, the following marginal and ideal values were set:

- **Drawbar pull –** By using the pulling ratio (ratio between drawbar pull and machine weight) a marginal value of 40kN was set, giving the concept same performance as the benchmarked competitor. An ideal value of 60kN was set, giving the concept the highest performance of the benchmarked products with a pulling ratio of 0,28.
- **Maximum speed –** Maximum speed for road-rail vehicles is in most cases regulated in different regional and national standards. The ideal value of 30 km/h was set to achieve a high traveling speed in most European countries while avoiding several regulations only applicable for vehicles traveling faster than 30km/h. The marginal value of 20km/h was set since it's the maximum allowed traveling speed in Sweden for road-rail vehicles.
- **Good precision in brake –** From customer reviews of previous road-rail machines at CEES, a need of good precision when braking has been addressed. Braking with good precision is crucial when mounting and dismounting heavy railway components in low speed. Braking only by reducing the speed of the hydraulic motor has previously been deemed insufficient.
- **Good traction and stability –** In order to ensure traction on all wheels while driving and at the same time have maximum stability while working at standstill, manufacturers generally design road-rail vehicles with one automatically lockable pendulum axle.

  Maximum allowed load on the rail wheel is dependent on rail wheel diameter. Throughout the years, CEES have tried multiple different diameters and different manufacturers offers different sizes. Previous experience within CEES is that larger wheel reduces the risk of derailment and increases traveling comfort while being more expensive.

  The last metric from "good traction and stability" was the use of differential. Trains and railway wagons typically have a wheelset with two wheels rigidly mounted to an axle and the rail wheel profile compensate for different travel lengths in curves. Two wheelsets are then typically mounted with a wheelbase of 2-3m on a bogie. CEES have previously used solutions with independent wheels and a wheelbase of about 4m. Due to uncertainties regarding slip and wheel wear with rigid axle and longer wheelbase, the use of differential was set as ideal.
- **Doesn't break when derailing –** Customer reports state that derailing is relatively common for road-rail construction equipment. Therefore, the axle must be designed in such a way that no components, such as brakes etc, breaks when the machine derails and the rail interfere with the axle instead of the wheels.

4.2 Regulated specifications

The study of technical standards resulted in several requirements primarily related to safety. Eight requirements from different standards were translated to ten measurable metrics. Some of the metrics were directly stated in the respective standard such as minimum ground clearance, the use of two independent brakes and railhead clearing device. Others were needed to be calculated, such as minimum braking torque where the torque was derived from maximum braking distance, friction between rail and wheels, dimensions of the machine and total weight of the machine. The regulated specifications can be seen in Table 7.

Table 7 Table of regulated specifications with corresponding metric, marginal value, and ideal value.

Regulation:	Metric:	Unit:	Marginal value:	Ideal value:
Max. Braking distance 27/55m[a]	Min. braking torque	[Nm]	>1500	>1660
A minimum of two independent brake-systems				
Min. Ground clearance	Specified in TDOK 2015:0143	[mm]	>80[b]	>130
Railhead clearing device	Must be equipped with a device for railhead clearing	Binary		
Equipotential bonding	Maximum impedance from highest point of machine to running rail[c]	[Ω]	<0,05	
Easy maintenance	"Machines shall preferably permit lubrication from the ground"			
	"Components which require frequent maintenance shall be easily accessible			
Failure recovery	Machines must have towing device at both ends			
Electrical safety	Live parts shall be located inside enclosures of at least IP2X			

[a] The minimum required braking distance is 27 or 55m for vehicles traveling 20 or 30km/h respectively according to EN 15746-1.A braking torque of 1500Nm and 1657Nm corresponds to a stopping distance of 27m and 55m respectively assuming a total weight of 30 tonnes and a wheel diameter of 0,7m.

[b] A minimum of 80mm required. A minimum of 130mm required for passing an active railway retarder.

[c] The maximum allowed impedance measured from highest point of the machine to the rail.

4.3 Integration to base machine

Specifications were also set to ensure possible integration to the base machine. The specifications involved electrical specifications regarding voltage levels, communication protocols and maximum current, all listed in the full product specification list in Product specificationsAppendix D. The specifications also covered mechanical integration and check for interference. Due to difficulties in quantifying possible sources of interference with the base machine, this was done in 3D-CAD.

By using a 3-D model of a base machine, the different solutions could be continuously modelled and modified to verify no interference between parts under any possible movement and verifying possibilities to mount components such as inverters etc. This could also be done with different types of equipment mounted to the base machine such as different tire sizes, to ensure possible integration to base machine.

4.4 Specifications from the company

The last type of specifications was targeting at needs mainly from CEES. It covered three needs that weren’t directly requested by customers but were requested from a business perspective:

- **Possible to use on different track gauges –** The standard gauge of 1435mm covers 55% of world railroad and by enabling multiple gauges the possible market increases. As a marginal value, 1435mm gauge with the option of 1520mm gauge (Used in Finland and most of former Soviet Union) was set, covering in total 72% of world railroad. As an ideal value, a set of track gauges ranging from 981mm to 1676mm was set covering almost all of world railroad except for some smaller railways used in mountain regions and some local tram lines.
- **Manufactured of standard components –** The road-rail products are a segment with relatively few products sold per year, manufactured in small series. This creates the need of the product being manufactured of “off the shelf” components as far as possible to keep the lead times lower, simplify spare parts distribution and lowering the final cost of the product. Since CEES is part of the Volvo group, components would preferably be chosen from different Volvo machines to simplify spare parts distribution even further.
- **Designed of proven technical solutions** – Product development involve lots of uncertainties and to reduce these uncertainties, one specification was aimed at using proven technical solutions. By using working principles already used within heavy duty machinery and construction equipment, possibly combined, or varied in a new way, would the chances of a successful prototype increase.

5 Concept generation

This section describes the process of generating new concepts. It also briefly describes the generated concepts.

5.1 Concept generation process

For concept generation, Ulrich and Eppinger recommend a five-step method to reduce the risk of some common dysfunctions during concept generation. The method involves five activities presented in Figure 14.

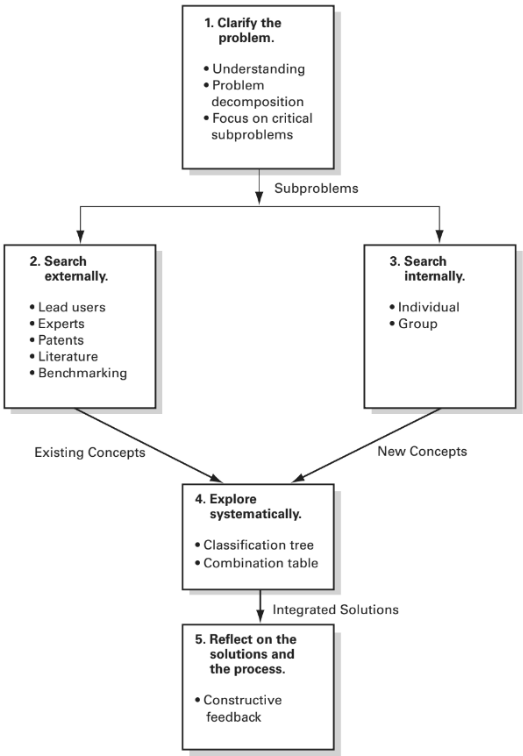


Figure 14 The five-step concept generation method according to U&E.

5.1.1 **Clarify the problem**

The main target of creating an electric drivetrain for road-rail vehicles was divided into two critical subproblems:

- Speed reduction – Electric motors generally have a nominal and maximum speed significantly higher than the 270 rpm which was calculated from the target specification.
- Braking possibility – To fulfil specifications, two independent braking systems is needed of which one must be a negative braking system.[1]

5.1.2 **Internal and external search**

Both an external and internal search was conducted to find solutions to the critical subproblems. The external search, using the previously studied literature, technical representatives at different companies and supervisors resulted in four different technical solutions:

- Gearbox with two output shafts.
- Wheel drive unit, commonly used for wheeled or crawler excavators.
- Complete wheel axle from another vehicle e.g. A truck.
- Integrated parking brakes within gearbox or wheel drive unit.

The internal search was conducted mainly by individual sketches of concepts. Some of these rough sketches, sometimes only showing a working principle, was refined after discussions with employees at CEES where they in some cases could suggest a product or machine using this working principle. This was the case for the idea with the portal axle. The individual search resulted in solutions with the transmission partly integrated into the wheel as well as some simpler transmissions using chain drive. During the internal search different braking solutions were also discussed, both previously used solutions within CEES as well as different mounting alternatives for the brakes. The different ideas generated were:

- Chain drive motor – wheel/wheel axle
- Portal axle
- Motor with outer rotor
- “Robson drive”

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[1]A braking system possible to perform braking without any power source, e.g. during a failure or a shut down machine is called a negative braking system.

The generated ideas formed different layouts and a possibility to combine and modify each idea to a complete solution to fulfil the target specifications. This was done in section 5.1.3, Explore systematically

5.1.3 Explore systematically

Mainly due to limited space with narrow track width and limitation in height to not disturb line of sight for the operator, some combinations of the different ideas were deemed impossible to integrate and were therefore excluded. Other ideas such as regenerative braking didn't require any physical components and could therefore be integrated with any combination. The different combinations that were deemed possible to integrate to the base machine are for overview categorized based on transmission layout in Figure 15 and for future reference listed with abbreviation in Table 8. The following pages describes each combination more in detail.

Table 8 Different combinations with corresponding abbreviations.

Combination	Abbreviation
Centre motor + Gearbox	A
Centre motor + Chain drive	B
Centre motor + Bought truck axle	C
Centre motor + Portal axle	D
Hub motor + Planetary gear (Inward mounted)	E
Hub motor + Planetary gear (Outward mounted)	F
"Robson Drive"	G
Wheel motor + Ring gear in wheel	H
Wheel motor + Chain drive	I

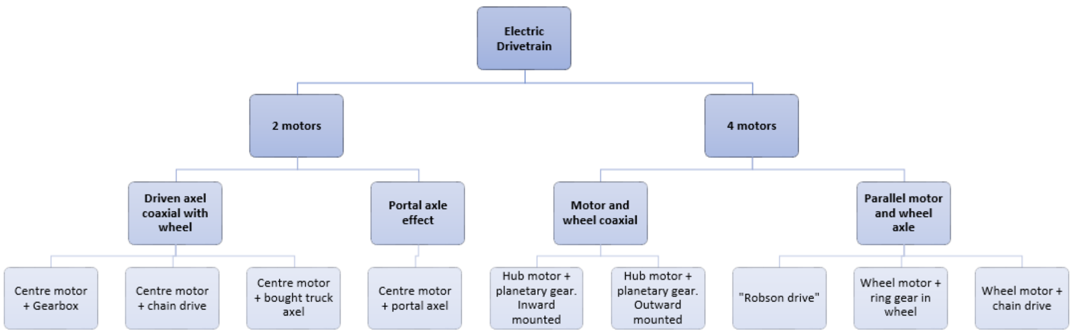

Figure 15 The generated combinations categorized based on transmission layout.

5.1.3.1 Centre motor and gearbox, A

A concept with a centre mounted motor with a gearbox for speed reduction. One motor and gearbox per axle and the two output shafts from the gearbox are directly coupled to the rail wheels. The rail axle is a simple tube-design. Braking can be solved with either drum or disc brake inside rail wheel or with disc brake on the driven axle, see Figure 16.

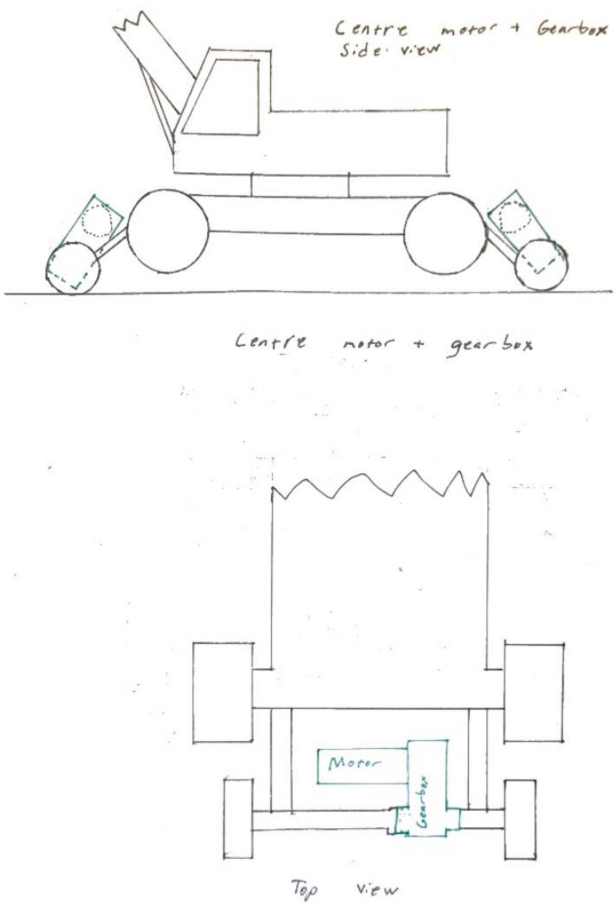

Figure 16 Concept "Centre motor and gearbox".

5.1.3.2 Centre motor and chain drive, B

A concept like “Centre motor and gearbox” but with chain drive between motor and wheel axle. The wheel axle is solid which makes it a simple construction but with no possibility to use a differential, see Figure 17.

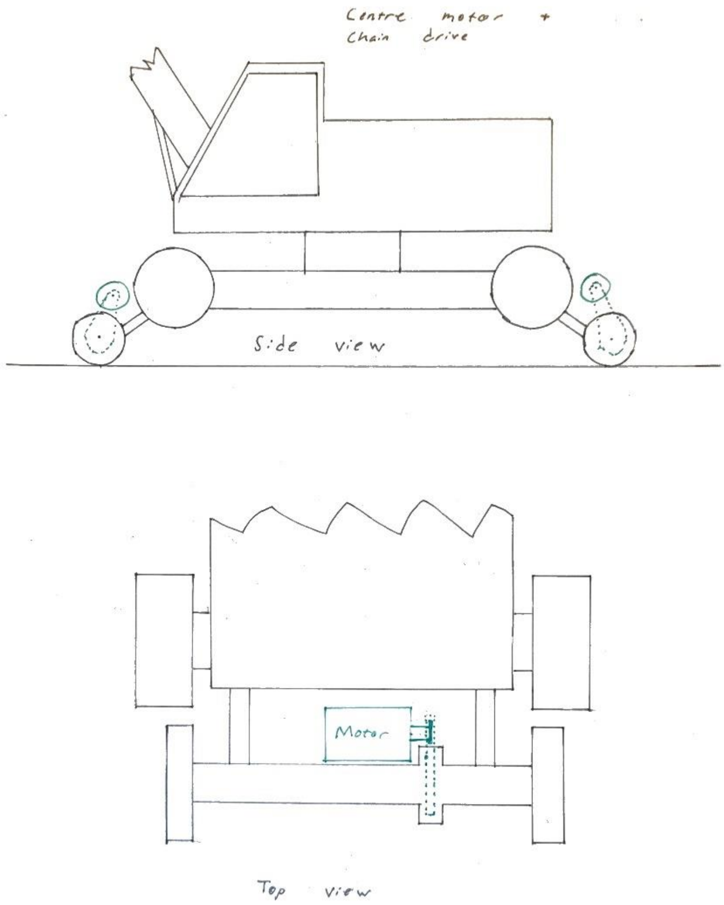

Figure 17 Concept “Centre motor and chain drive”.

5.1.3.3 Centre motor and truck axle, C

A concept with a centre motor coupled to a wheel axle from a heavy vehicle e.g., a truck or a wheel loader, Figure 18. A complete bought axle makes the concept easier to manufacture and it can be ordered with brakes and differential. The wheel axle will most probably be more expensive than the simpler ones in “Centre motor and gearbox” and “Centre motor and chain drive”.

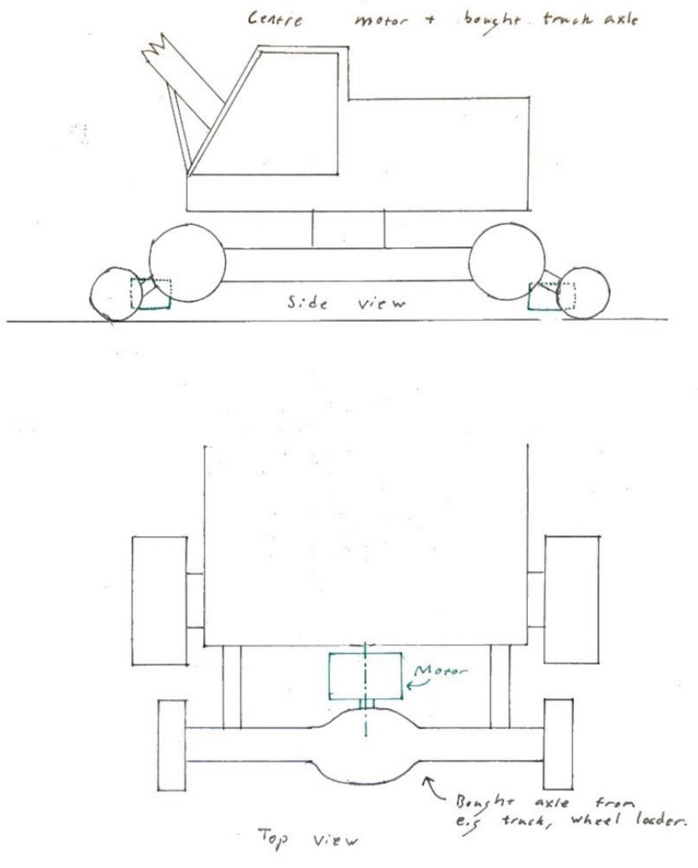

Figure 18 Concept “Centre motor and truck axle”.

5.1.3.4 Centre motor and portal axle, D

A concept like "Centre motor and truck axle" but with a portal axle for increased ground clearance. Braking can be solved with either brakes inside rail wheel or possibly with brakes on the motor axle, Figure 19.

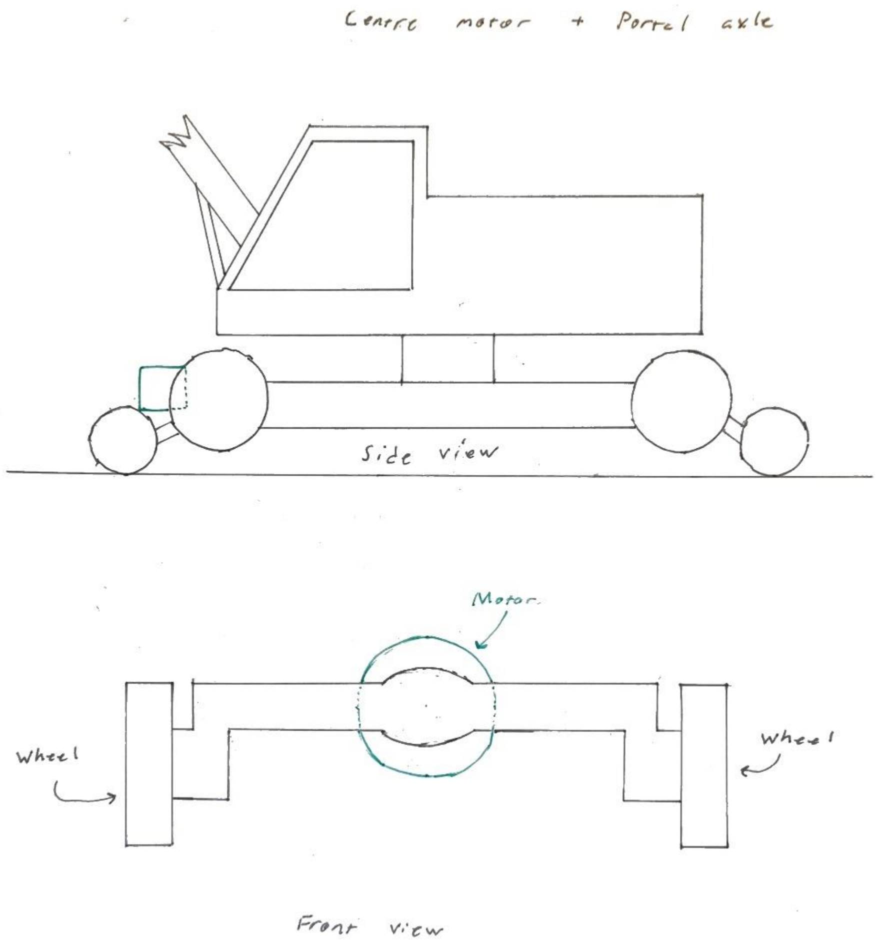

Figure 19 Concept "Centre motor and portal axle".

5.1.3.5 Hub motor, inward mounted, E

A motor and planetary gear where the gearbox forms the wheel hub. The motor is placed towards the centerline of the vehicle, preferably inside some sort of rigid structure for protection during derailment. This solution requires internal brakes inside the wheel hub, Figure 20.

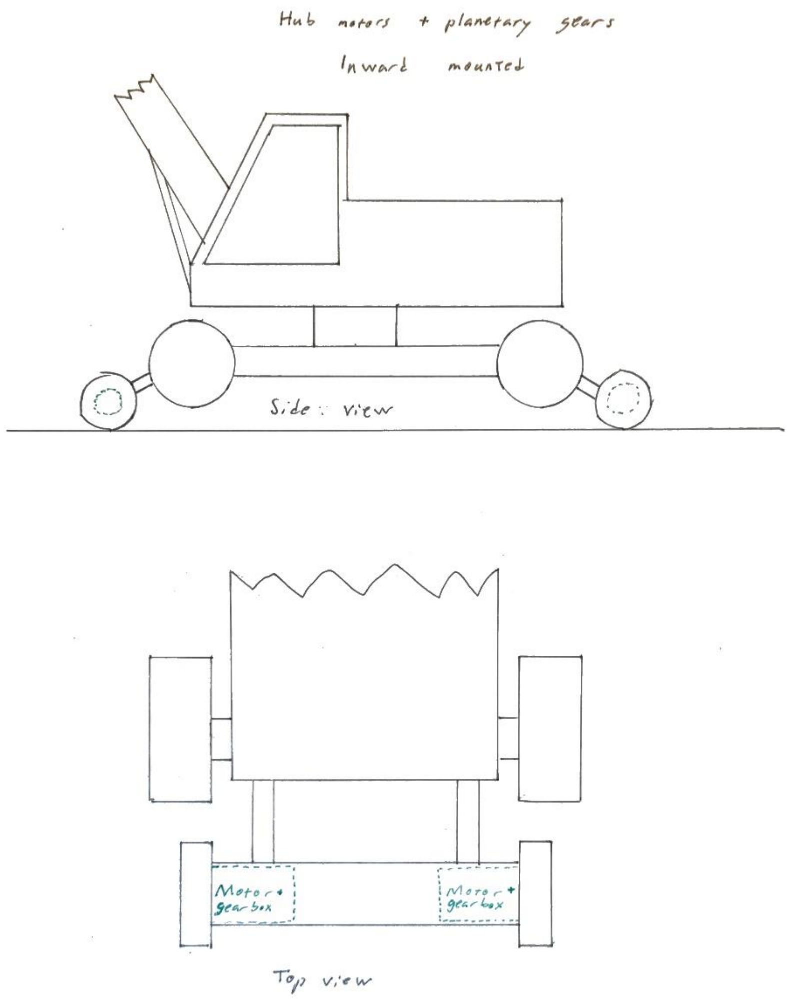

Figure 20 Concept "Hub motor inward mounted".

5.1.3.6 Hub motor, outward mounted, F

A motor with an outer rotor, possibly combined with a planetary gear, Figure 21. The outward mounting simplifies packaging and makes the axle easier to implement on other vehicles. Brakes can preferably be placed inside the rail wheel.

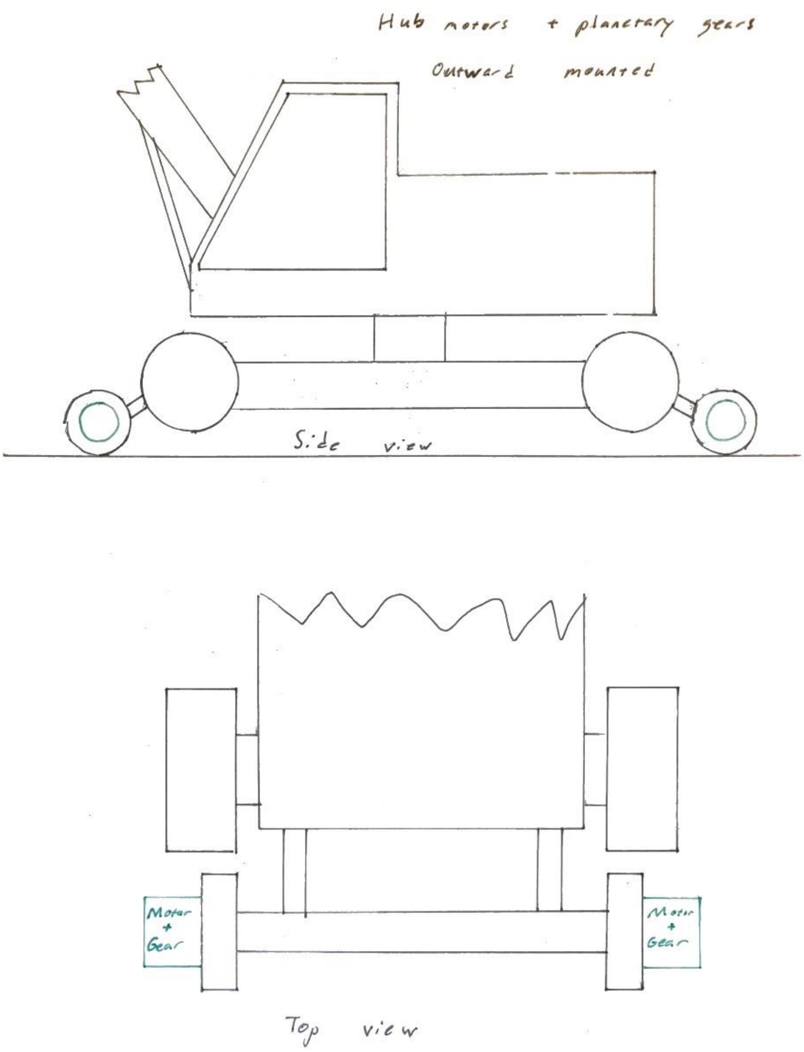

Figure 21 Concept "Hub motor outward mounted".

5.1.3.7 "Robson drive", G

A concept inspired by "Robson drive" where a large wheel is driven by an outer smaller wheel pressed against the perimeter of the wheel. In this concept, the motor is attached to a rubber wheel via a set of gears. The rubber wheel is in turn pressed against the rail wheel, see Figure 22. Since this motor is acting on the perimeter of the wheel, brakes could preferably be placed inside the wheel.

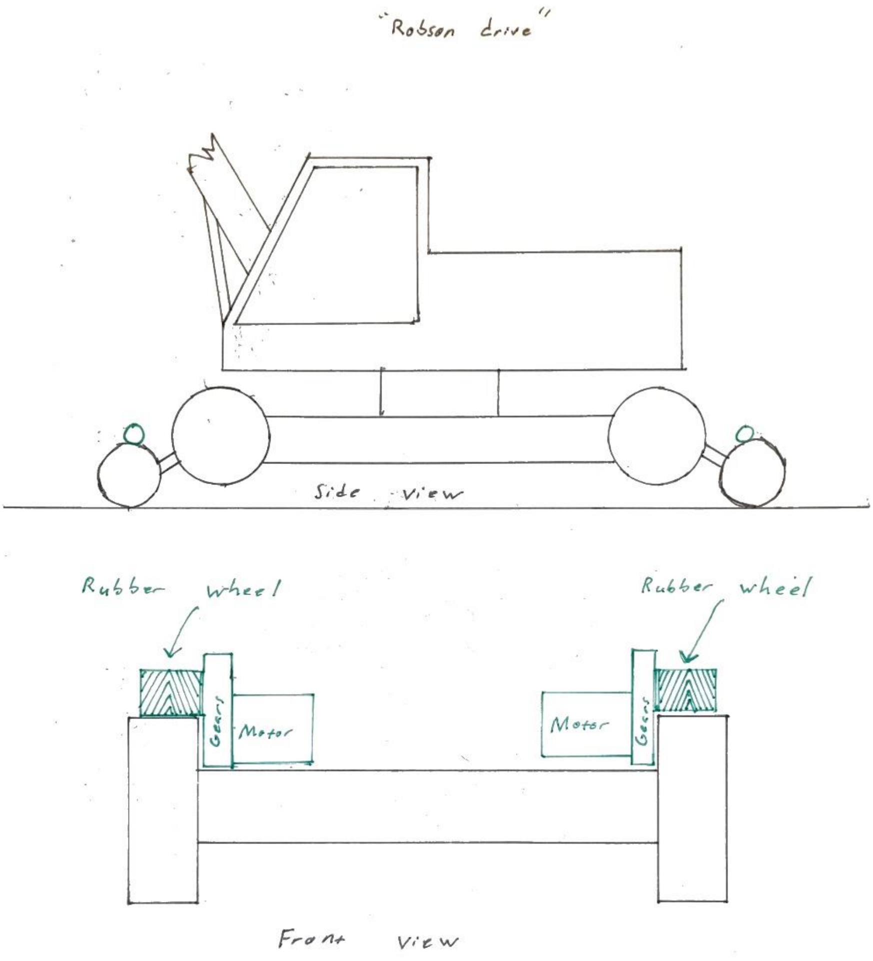

Figure 22 Concept "Robson drive".

5.1.3.8 Wheel motor and ring gear in wheel, H

A concept where a ring gear inside the rail wheel forms the last reduction in a set of gears. The motor is mounted in a similar position as the "Robson drive" concept, see Figure 23. The internal ring gear makes it harder to mount the brakes, but they can possibly be placed at any of the rotating axles of the gear set.

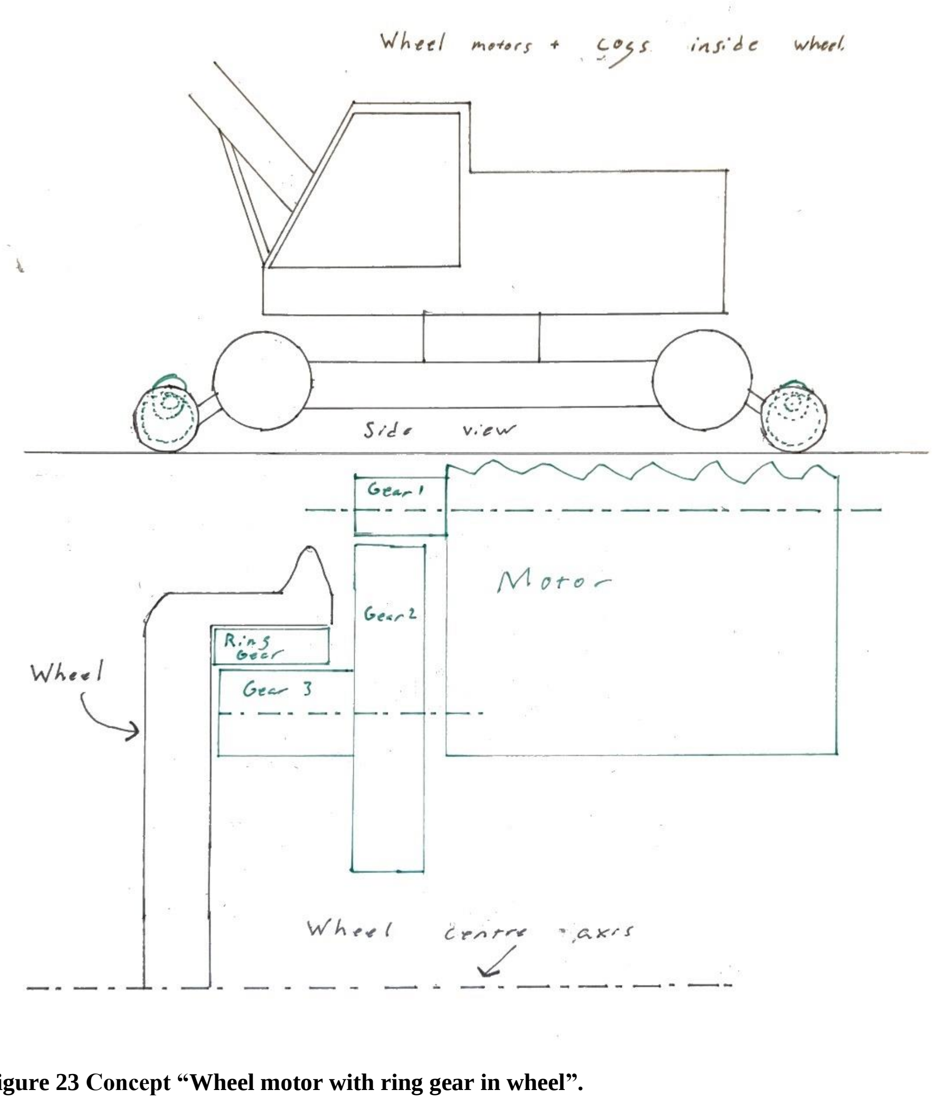

Figure 23 Concept "Wheel motor with ring gear in wheel".

5.1.3.9 Wheel motor and chain drive, I

A concept where torque is transmitted from the motor to the wheel via a chain and a sprocket mounted directly to the wheel. This gives the concept some freedom in the placement of motor as well as speed reduction, see Figure 24. Brakes can possibly be mounted inside of wheels.

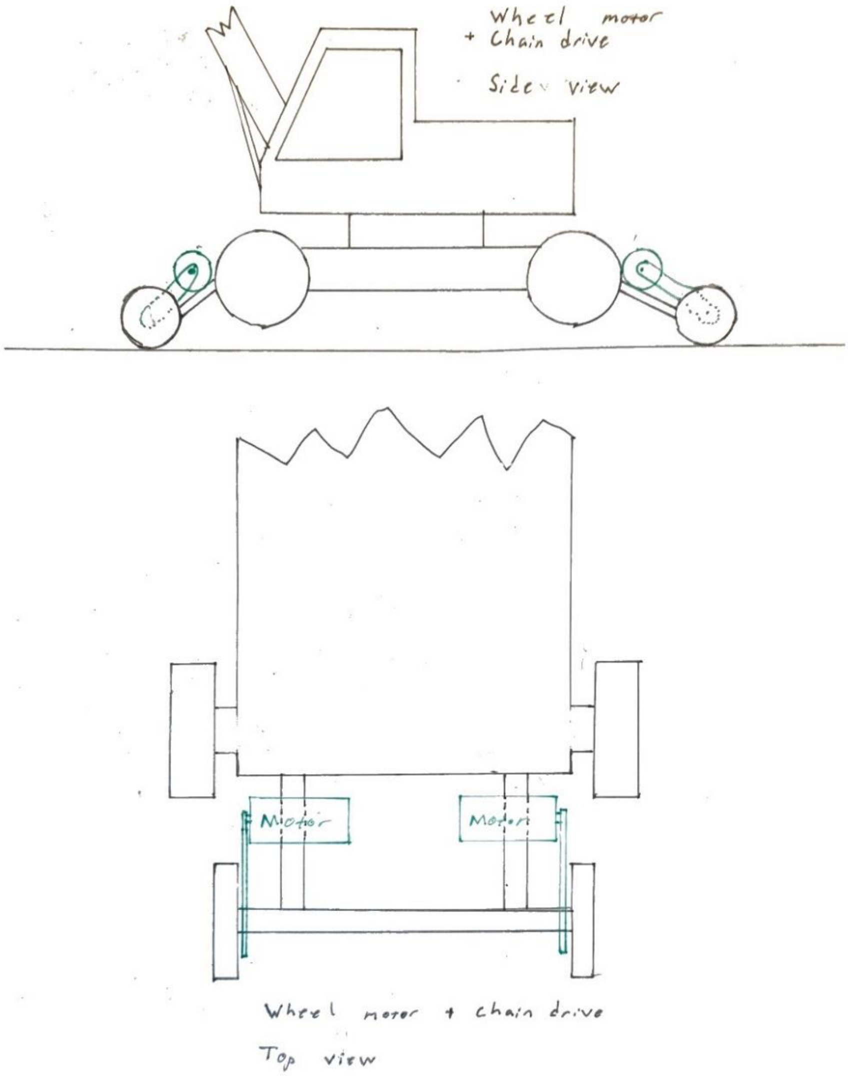

Figure 24 Concept "Wheel motor and chain drive".

6 Selection of concept

This section describes the process of reducing the number of concepts from nine to one using concept screening and concept scoring.

6.1 Concept screening

To quickly reducing the number of concepts to prioritise the more promising ones, Ulrich and Eppinger suggest concept screening. Concept screening is based on the *Pugh concept selection* and the idea is to compare the concepts relative to a reference concept for several criterions. The concepts are only ranked *better, worse* or *same as,* as the reference concept. Since the criterions aren't weighted, it's important to only chose the more important criterions for evaluation to get adequate results (Ulrich & Eppinger, 2012, pp. 150-153) .

For this thesis, five different criterions were derived from the product specifications and was chosen for the concept screening:

- **Protected during derailment**
  The customer need "doesn't break when derailing" was derived to a criterion to avoid designs with components mounted in risk full areas with small possibilities to provide sufficient protection. This was for example the case for concept A and B where transmission components with larger diameter than the axle could potentially be hard to protect while still fulfiling the required ground clearance.
- **Differential**
  The criterion "Differential" was also derived from the customer needs. Many of the concepts would be impossible to integrate with a differential and the influence on traction with a ridged axle was unknow. The concepts possible to integrate with a differential could also be modified to use a ridged axle, making these concepts more versatile.
- **Integration of two brakes**
  Integration of two brakes was a requirement from the technical standards, that were deemed to be hard to comply with while fulfilling other

specifications such as protected during derailment and minimum ground clearance and was therefore chosen as one criterion.

- **Use of “off the shelf” parts**
  Using “off the shelf” and standard parts could be an important economic factor as well as simplifying future spare part distribution which was a need from CEES. It also had an important role in the possibility to potentially manufacture a prototype within the project and was therefore chosen as a criterion.
- **Proven Solution**
  As previously describes was the use of proven technical solutions and working principles chosen as a specification to reduce overall risk within the project. It was therefore also selected as one of the criterions for the screening.

The concept screening can be seen in Table 9.

Table 9 Concept screening. + For “better than reference”, - For “worse than reference”.

	Concepts								
Selection criteria	A	B	C	D	E	F	G	H	I
Protected during derailment	-	-	0	0	0	-	0	0	-
Differential	-	-	+	+	0	0	0	0	0
Integration of two brakes	0	+	0	0	0	+	0	-	-
Use of “Off the shelf parts”	+	+	-	-	0	-	-	-	+
Proven solution	0	0	0	-	0	0	-	-	0
Sum	-1	0	0	-1	0	-1	-2	-3	-2
Rank	2	1	1	2	1	2	3	4	3
Continue?	No	Yes	Yes	No	Yes	No	No	No	No

Concept	**Abbreviation**
Centre motor + Gearbox	A
Centre motor + Chain drive	B
Centre motor + Bought truck axle	C
Centre motor + Portal axle	D
Hub motor + Planetary gear (Inward mounted)	E
Hub motor + Planetary gear (Outward mounted)	F
“Robson Drive”	G
Wheel motor + Ring gear in wheel	H
Wheel motor + Chain drive	I

6.2 Refinement of concepts

For concept B, C and E, the process continued with search for suitable components and 3-D modelling. The search for components were mainly done using product catalogues and contacts with different manufacturers and distributers. The manufacturers and distributors chosen where chosen from both local and global companies, both suppliers to CEES as well as other companies. The search involved the following companies:

- Bonfiglioli – Manufacturer of gears, travel drives for heavy duty machinery and hydraulic- and electrical motors.
- Parker Hannifin – Manufacturer of Electrical motors and inverters among other things.
- Bosch Rexroth – Manufacturer of both electric- and hydraulic drivetrains including motors and different types of gears.
- Neotec – Manufacturer of various rail vehicles, also offers complete hydraulic- and electrical axles for road-rail vehicles.
- Carraro – Manufacturer of transmission systems for tractors and off-highway vehicles.
- Bengtssons Maskin – Distributor of transmission components situated in Malmö. Offers components from various manufacturers.
- Reggiana Riduttori – Manufacturer of planetary gears and travel drives.

The companies were chosen to both get a more or less complete solution from one supplier as well as solutions developed by combining products from multiple suppliers. The selection was also based on technical information available online, where other companies with less technical information publicly available was dismissed due to time limitation in the project.

6.2.1 “Hub motor + planetary gear, inward mounted” concept E

For the concept “Hub motor + planetary gear, inward mounted” solutions with a wheel drive unit combined with an electric motor was found at both Bonfiglioli and Bosch Rexroth. A similar solution could be made by combining components from Parker Hannifin and Reggiana Riduttori. Wheel and track drive units are commonly used in off-road vehicles, often combined with a hydraulic motor. The drive unit consist of planetary gears in series with the wheel mounted directly to the unit and can often be combined with internal negative static brake, see Figure 25.

Figure 25 Wheel drive unit combined with an electric motor from Bonfiglioli.

The higher output speed from electrical motors compared to hydraulic motors requires the drive unit to have a higher reduction ratio while still withstanding the heat generated inside the gear with increased speed.

For this concept, two different approaches were investigated. Both approaches involved an electric motor and drive unit with similar motor diameter but different torque/speed characteristics and power levels. The two approaches were chosen to investigate possibilities and challenges with the use of high- versus low-speed motor. A size comparison can be seen in Figure 26.

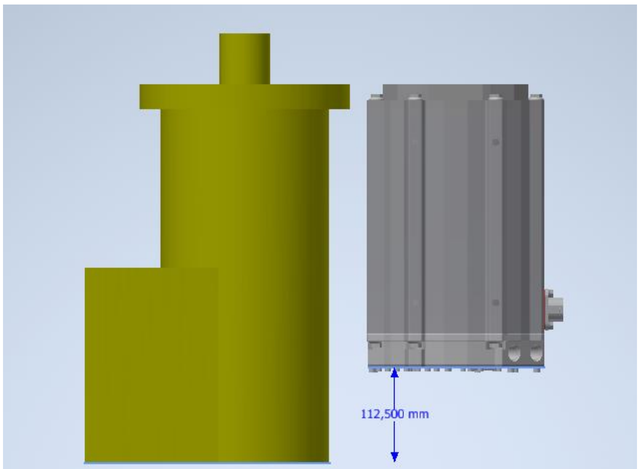

Figure 26 Size comparison between high-speed motor (left) and low-speed motor (right).

To fulfil the product specification of at least two different track gauges (1435 and 1520mm) the solutions were optionally equipped with 42,5mm spacers between rail wheel and drive unit flange. The increased axial distance between the drive unit and rail wheel increases the radial load on the internal bearings of the drive unit and thereby lowering the permitted radial load of the drive unit. An example of a load diagram for a drive unit can be seen in Figure 27.

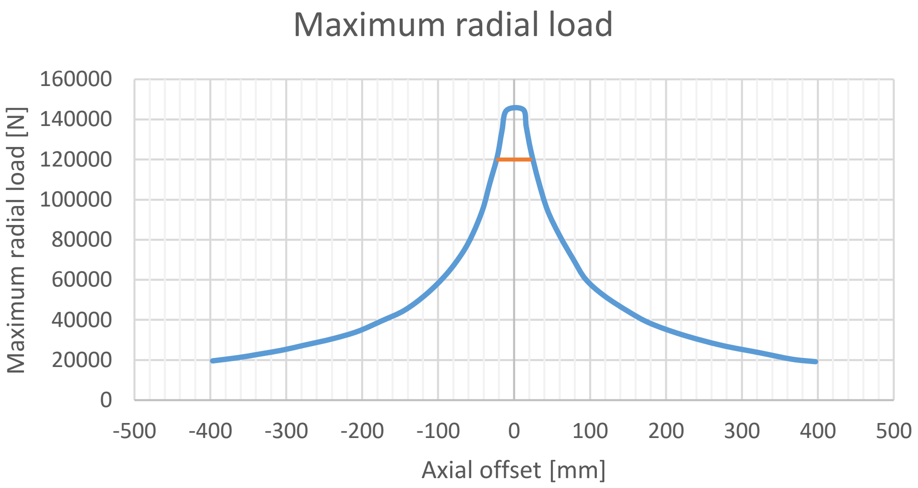

Figure 27 Maximum permitted radial load depends on the axial offset between drive unit and wheel contact patch. The orange line illustrates an offset of -21,25 – 21,25mm which gives a maximum radial load at approximately 120 kN if the axial offset is perfectly adapted for a 42,5mm spacer. Graph derived from Rexroth's data.

Due to the lower radial load capacity when using spacers, this approach might require a larger drive unit compared to using a solution without spacers for different track width. The approach also limits the solution from allowing large differences in trackwidth.

6.2.1.1 "Hub motor + planetary gear, inward mounted" solution E1, high speed

The high-speed solution was a complete solution provided from Bosch Rexroth's *eLion.* product range. The solution was chosen to be an example of a solution with a motor with all transmission components adapted performance-wise to each other. By selecting both motor and transmission from the same product range, the full performance of both motor and transmission could potentially be utilized without differing speed and torque limitations for motor and transmission. The product range includes motor, drive unit and inverter and thereby offering simple drive unit-motor-inverter. The *eLion* product range was presented to the public in September 2021 and includes electric motors, inverters, and gearboxes in different sizes in a modular design (Anon., 2021). The motor chosen for this application had a maximum speed of 12000 rpm and the drive unit a maximum input speed of 14000 rpm with a reduction rate of 58,634.

Based on simulation data from Henrik Jarl at Bosch Rexroth, diagrams of traction force and power for the machine could be made. A wheel diameter of 0,7m was used and the nominal traction force, without any limitations from the base machine can be seen in Figure 28. According to the specifications for integration to base machine, a maximum current of 200A could be supplied to the undercarriage of the machine. A current of 200A at 600VDC gives a maximum available power of 120kW and is illustrated in Figure 28 by the blue line. As seen, the traction force of the machine will from 7km/h be limited by the maximum power through the base machine swivel and it will be impossible to use the full potential of the motor at higher speeds. Note: The blue line represents 120kW without any mechanical or electrical losses.

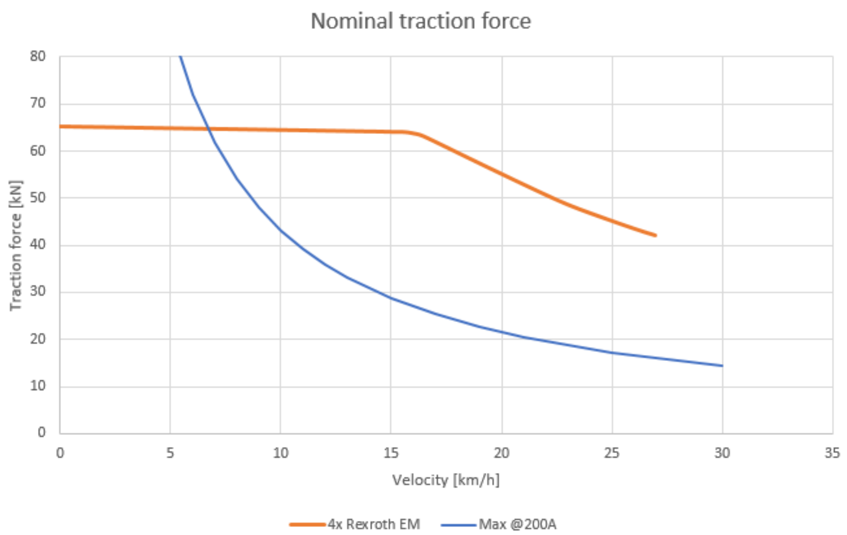

Figure 28 Nominal traction force for the E1 solution. The solution offers relatively high traction force compared to the margin value of 60kN. The maximum speed reaches 27km/h.

The total nominal traction power of the machine can be seen in Figure 29.

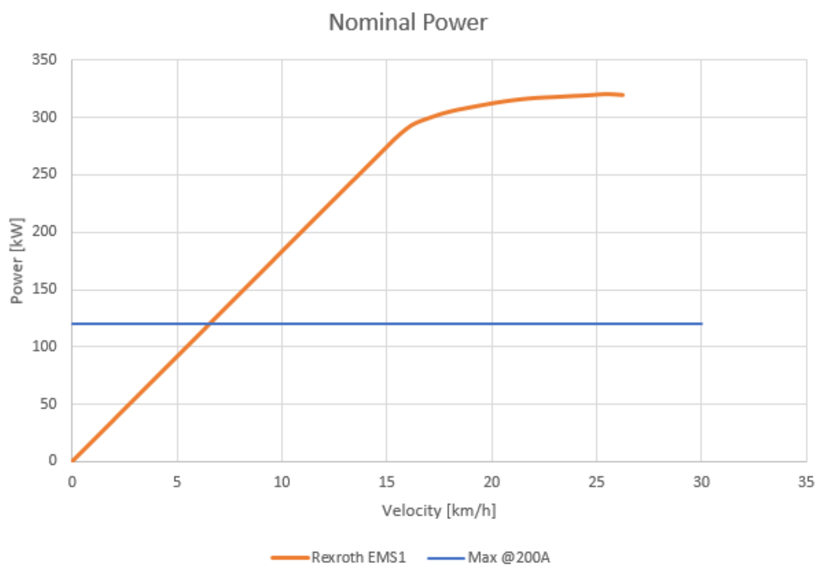

Figure 29 Nominal power of the E1 solution. For velocities above 7km/h the current through the swivel at 200A is the limiting factor.

The solution from Bosch Rexroth only had an integrated parking brake and the large diameter of the wheel drive unit made integration of service brake inside rail wheel impossible. A rough model was made with the brakes mounted on the axle connecting the motor with the wheel drive and can be seen in Figure 30.

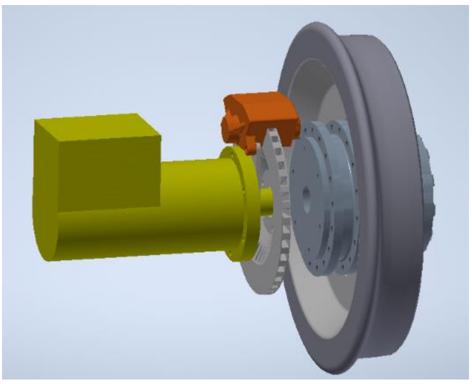

Figure 30 Rough model with Rexroth motor and wheel drive unit. A disc brake is placed between the motor and drive unit.

Due to the brakes acting on the input side of the drive unit, rotational speeds of up to 12000 rpm can be expected but only low braking torque is required. For reference, the rotational speed of a brake disk for a car equipped with tires with 2 m rolling circumference traveling at 200km/h can be calculated to 1700 rpm. Further research is needed to verify breaking functionality for the high speed.

The motor, drive unit and inverter were then mounted in a complete axle assembly, Figure 31. The assembly also had a sturdy bent steel sheet placed on the lowest point of the assembly to protect motor and brakes during derailment.

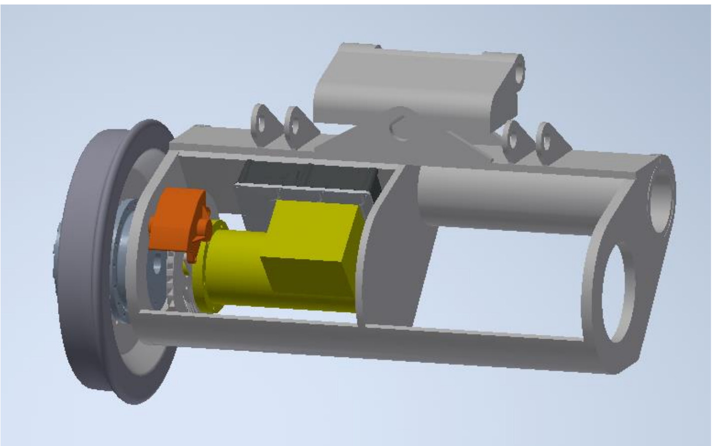

Figure 31 Motor and drive unit mounted to an axle. The inverter can be seen behind the motor.

6.2.1.2 "Hub motor + planetary gear, inward mounted" solution E2, low speed

The second solution that was found had a similar layout as the Bosch Rexroth solution but was assembled from a motor and inverter used in other Volvo equipment. The torque/speed characteristics of the motor can be seen in Figure 32.

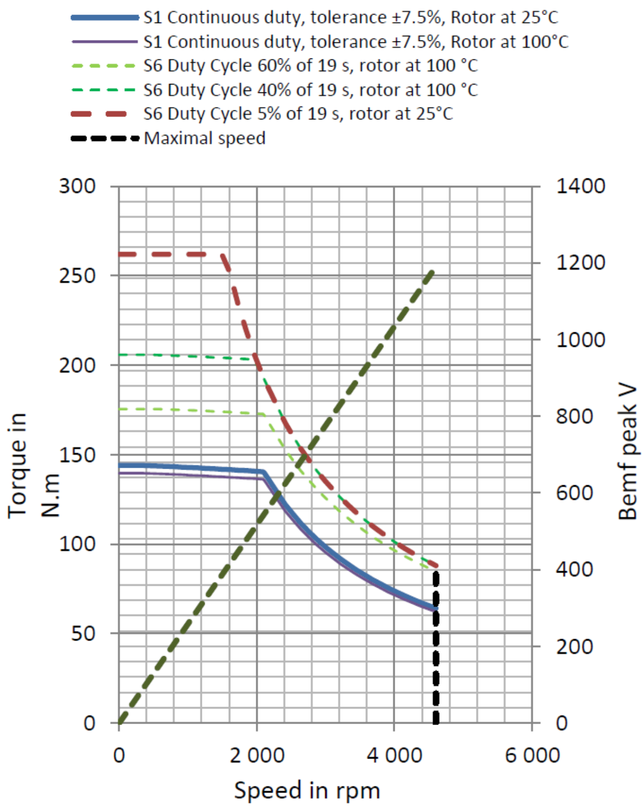

Figure 32 Torque and speed characteristic for the electric motor in the E2 solution. Data from Parker Hannifin.

The motor characteristic was combined with the gear ratio of 24,41 for the wheel drive unit and a wheel diameter of 0,7m to calculate traction force and maximum speed, Figure 33. The solution was chosen as an example of a solution with different performance and limitations in speed and torque due to different manufacturers of motor and drive unit. The maximum input speed of the wheel drive of 4000 RPM restricted the solution to not fully utilize the motor, restricting the maximum vehicle velocity to 21 km/h compared to 25 km/h at a theoretical motor speed of 4500 RPM.

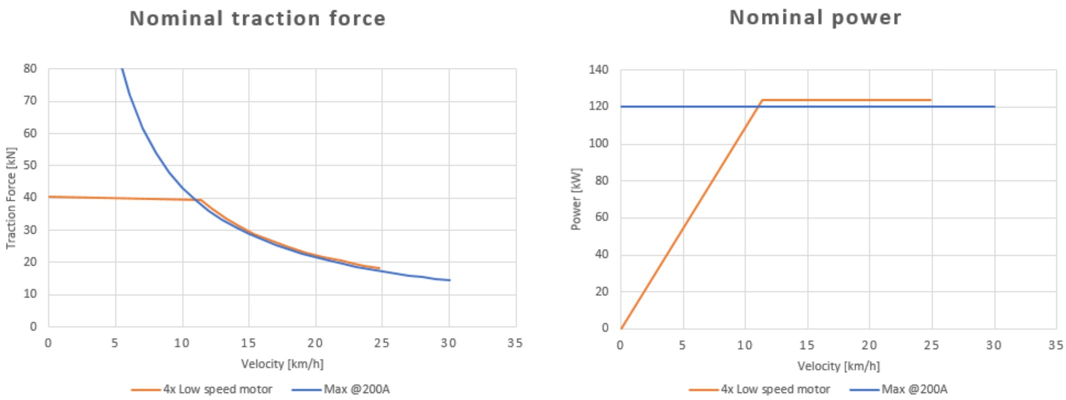

Figure 33 Nominal traction force and power for the second solution derived from torque characteristics in Figure 32, gear ratio of 24,41 and wheel diameter of 0,7m. As seen, both traction force and power are significantly lower than for E1. Note: The maximum input speed of the wheel drive unit at 4000 rpm restricts the maximum velocity to 21 km/h.

Motor, drive unit and brake were assembled in similar way as solution E1 in a rough model, Figure 34.

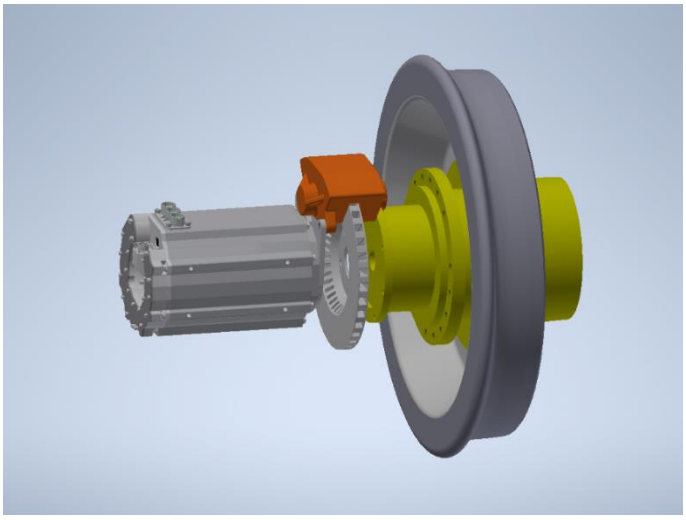

Figure 34 The E2 solution with motor and drive unit. Same integration of brakes and the motor is both smaller and less powerful compared to E1.

The smaller motor flange of the low-speed motor increased the possible ground clearance for the E2 solution compared to the E1 when mounted in a similar axle arrangement, Figure 35

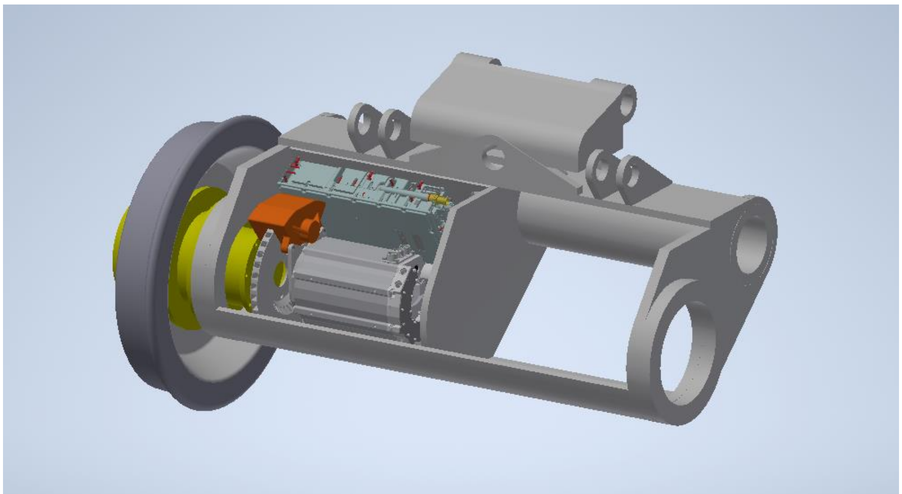

Figure 35 The E2 solution mounted in a similar way as E1.

6.2.2 **“Centre motor + chain drive” concept B**

For the concept “Centre motor + chain drive” the idea was to mount the motor parallel to the wheel axis. A chain drive would then transmit the torque from the motor to the rigid driven axis and having a suitable speed ratio for the motor. However, during the project no suitable low speed, high torque motor could be found for traction applications. All motors found had a nominal speed of 3000-8000 rpm and to fulfil the specification for traction force with a reasonable large motor, a speed reduction ratio in the range of 20-30 would be needed.

According to Khurmi and Gupta (Khurmi & Gupta, 2005, p. 760) chain drives permit ratios of 8 to 10 in one step and the maximum permissible speed of the smaller sprocket with different types of chains can be seen in Table 10.

Table 10 Maximum allowed speed for chains in rpm according to Khurmi and Gupta (Khurmi & Gupta, 2005, p. 770).

Type of chain	**Number of teeth on the smaller sprocket**	**Chain pitch in mm**				
		12	15	20	25	30
Roller chain	15	2300	1900	1350	1150	1100
	19	2400	2000	1450	1200	1050
	23	2500	2100	1500	1250	1100
	27	2550	2150	1550	1300	1100
	30	2600	2200	1550	1300	1100
Silent chain	17-35	3300	2650	2200	1650	1300

Due to the high reduction ratio combined with the high speed of the motor, a chain drive transmission was deemed insufficient for this type of application. A set of gears with a single reduction step was briefly considered to replace the chain drive but according to Childs, the useful gear ratio for spur gears is 1:1-6:1 and for double helical gears 1:1-15:1 (Childs, 2019).

The concept was instead reevaluated and an idea of combining the “Centre motor + chain drive” and “Centre motor + gearbox” formed. The combination with a reduction gear mounted to the motor in combination with a following chain drive or spur/helical gear would give the advantages of the chain drive concept with easy integration of two brakes. The reduction gear could preferably be of an industry standard model making the combined concept mainly built of “off the shelf” components.

6.2.2.1 Combination concept AB

The first combination evaluated was the use of industry reduction gearboxes. During a meeting with Anders Cronbring, key account manager at Bengtssons-maskin, different gearbox alternatives were discussed. The discussions mainly focused on planetary gearboxes combined with a chain or gear drive, Figure 36.

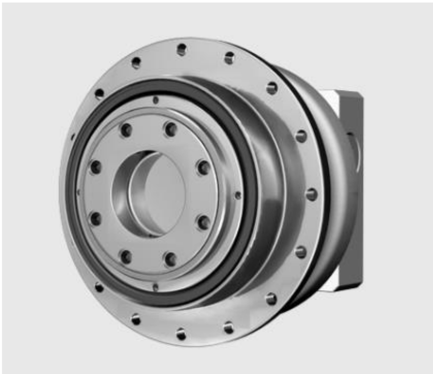

Figure 36 High precision planetary gearbox from Apex Dynamics. Available in multiple ratios and output torques.

Gearboxes with either parallel shafts or bevel gearboxes were also discussed to be mounted similar as the “Centre motor + gearbox” concept, Figure 37.

Figure 37 Gearbox with motor axle parallel to output axle from Rexnord. The output axle is hollow for mounting directly to wheel axel.

Starting point of the discussions was the use of a permanent magnet synchronous motor from Parker with the specifications listed in Table 11. The motor was chosen since it’s a motor designed for traction of both on and offroad vehicles. The chosen model was the smallest model to both fulfil the criteria for 40kN nominal traction force at standstill and a maximum speed of 30km/h and this could be accomplished with a gear reduction of 20,55:1-20,75:1 and a rail wheel diameter of 0,6m.

Table 11 Specifications for the electric motor used as a baseline during gearbox discussions. Data from Parker Hannifin

Motor: Parker xxx-yyy	
Rated power [kW]	104
Rated torque [Nm]	292
Rated speed [rpm]	3390
Peak power [kW]	170
Peak torque [Nm]	700
Maximum speed [rpm]	5500
Motor diameter [mm]	310
Motor length [mm] (shaft excluded)	315

No suitable industrial gearbox could be found with a reduction ratio around 20:1 that could handle the specified maximum input speed with the required output torque during low speed.

After further discussions with Anders Cronbring, the only viable option for using an industry standard gearbox was deemed to use a motor with sufficiently large nominal torque to reduce the input speed and ratio of the gearbox while still fulfilling the traction force criteria. The discussions formed a concept using a helical gear reducer combined with a larger motor. Due to the larger motor, the gear ratio could be reduced to 11,8 and thereby could also the gearbox input speed be reduced. A first 3D-model was made with most parts of the drivetrain placed inside a box shaped steel structure to be protected during derailment, Figure 38.

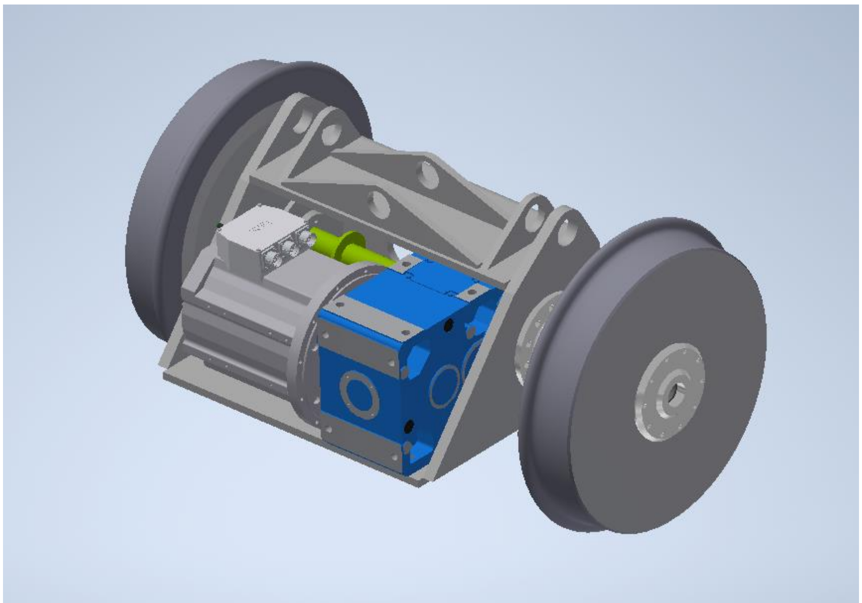

Figure 38 Early CAD model of the AB combination concept. Motor, gearbox(blue) and drive axle(green) are placed in a box shaped steel structure. A disc brake can be mounted at the flange of the driven axle and the steel structure protects all parts during derailment.

Even though the AB concept ended up like the previously dismissed concept “A - Centre motor and gearbox”, the project still proceeded with the concept. This is because the steel structure was deemed to give sufficient protection during derailment, adding an extra point to the concept A in the concept screening.

Based on the motor-gearbox combination for the AB-concept with a wheel diameter of 700mm for sufficient ground clearance, a traction force and power graph were derived and can be seen in Figure 39. The maximum input speed of the gearbox limits the maximum velocity to 28km/h

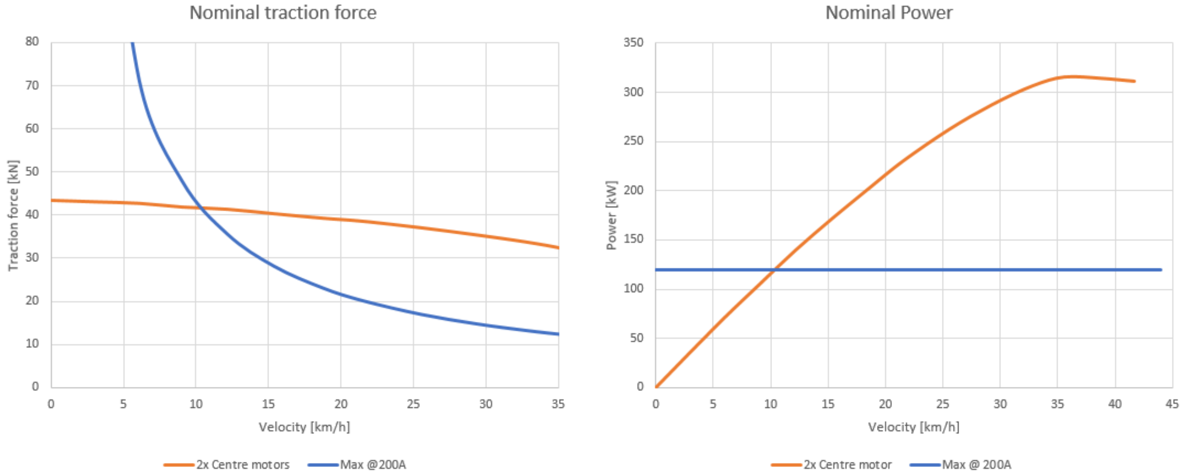

Figure 39 Nominal traction force and power for the AB-concept. Note: The maximum input speed at 2500rpm for the gearbox limits the maximum velocity to 28km/h.

The concept was further developed with a modular design for different track gauges making the concept a viable option for all track gauges in the product specification, Figure 40.

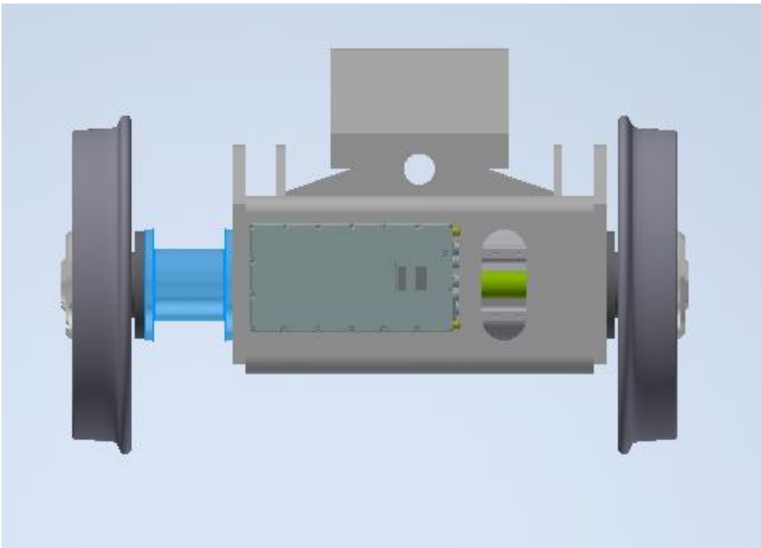

Figure 40 The AB-concept seen from the front. Identical wheel hubs on left and right side. The righthand wheel hub is mounted directly to the central steel structure and with an identical setup on the left side giving a trackwidth of 981mm. The lefthand wheel is mounted to an extra steel tube (marked in blue) which, with an identical setup on the right side, gives a trackwidth of 1435mm. Multiple trackwidths can be obtained by using different lengths of the steel tube and corresponding lengths of driveshafts.

6.2.3 “Centre motor + bought axle” concept C

For the concept “Centre motor + bought axle” research was done in applications with similar axle loads and track width. The concept initially focused on finding a wheel axle from a conventional transmission system for either trucks, construction equipment or material handling vehicles. Several axles could be supplied with multiple different gear ratios to be adapted to the required speed and traction force. The limiting factor was the relatively high axle loads for a 30 tonnes machine combined with the narrow track width of 1435mm. This led to two alternatives, one complete axle made for railway applications supplied by Neotec, Figure 41.

Figure 41 Electric railway axle from Neotec. Image from Neotec.

Neotec offers both road-rail vehicles as well as separate railway axles for such vehicles. Unfortunately, neither Neotec nor the Swedish distributor answered the requests on the electric axle and therefore is most of the info of the axle unknown. CEES have previously received quotations on other products from Neotec for road-rail use, but the products were deemed too expensive for that application. Due to those two reasons, no further research was done on the Neotec alternative.

For the second alternative, the possibility to buy a complete axle was discussed with the company Carraro. Carraro manufacture transmission system for tractors and off highway use. No off the shelf solution could be found but an axle for counterbalance trucks and airport tractors could be adapted for the application. The axle had an inbuilt motor with dual planetary gearboxes and could be delivered with integrated service and parking brake, Figure 42.

Figure 42 EC 30 axle from Carraro with an integrated center motor and dual planetary gearboxes.

For this project, the axle needed the following three modifications.

- Lengthen of the axle to increase the flange distance from 1075mm to 1435mm.
- Rewinding of motor to adapt for the 600V DC-bus instead of the original 48V.
- Modification of gear ratio to fulfil the specifications for drawbar pull and maximum speed.

Modifications of axle length and gear ratios were deemed possible by Carraro assuming sufficient market volume. Regarding the motor voltage, no clear answers could be obtained during the project and the concept of using a completely bought electric axle was left partly unanalyzed.

The three refined concepts were put together in a table to highlight differences in characteristics, Table 12.

Table 12 Main characteristics of the concepts

Characteristics	Unit	E1, High speed	E2, Low speed	AB1, Centre motor
No. of motors+inverters	-	4	4	2
No. of brakes	-	4	4	2
Differential	Yes/No	Yes (Traction control possible via software)	Yes (Traction control possible via software)	No
Wheel diameter	mm	700	700	700
Traction force at standstill	kN	66	40	43
Maximum speed	km/h	27	21	28
Motor max rotational speed	rpm	12000	4400	4000
Gearbox max rotational speed	rpm	14000	4000	2500
Nominal speed	km/h	16	12	30
Traction force at 20km/h	kN	22	22	22
Traction force at standstill (peak)	kN	127	73	70
Pulling ratio[1]	-	0,3	0,19	0,2
Total nominal power motors	kW	268	124	350
Total peak power motors	kW	Approx. 500	169	540
Ground clearance	mm	130	160	130
Possible rail gauges	mm	1435-1520	1435-1520	981-1675
Possible problems	-	- Braking at 12000 rpm - Small margins for axial clearance	- Braking at 4000 rpm -Wheel drive unit available but not a standard component	-Using an industry standard gearbox in mobile application
Price estimation	Relative	1	0,7	0,5

[1] Based on traction force at standstill.

6.3 Concept scoring

To better differentiate among the remaining concepts, U&E recommends a method called concept scoring. Concept scoring involves a more refined comparison between the concepts with respect to each criterion. The criteria are weighted to get a comprehensive ranking of the concepts (Ulrich & Eppinger, 2012, p. 154). Due to the weighting of the criterion, less important criterions can be evaluated during concept scoring unlike the previous concept screening.

The parameters chosen for concept scoring were scored 1-5 and divided into six different categories. The parameters were a mix of parameters based on the product specifications as well as more general parameters that were deemed important for all products from CEES. The parameters added from CEES´s product development process are marked in Table 13.

Scoring of the more subjective parameters and deciding of weight factor was done together with CEES during a design review where all concepts were described thoroughly. The different categories and the reasoning behind the parameter scoring can be seen below:

- **Technical performance**
  Technical performance included factors based on customer needs such as traction force, travel speed and differential. The ranking of technical performance was carried out by setting the mid ranked concept to score 3. The two other concepts were then scored 1-5 based on the mid ranked performance equaling three. For example: The three concepts E1, E2, AB had a traction force of 66kN, 40kN and 43kN respectively. That makes the AB concept mid ranked and thereby scored three. Based on 43kN equaling score three, the other two concepts can be ranked 1-5. A finer scoring e.g., 1-9 would better differentiate the concepts technical performance but was dismissed due to a more difficult ranking of the more subjective parameters.

- **Aftermarket**
  The category aftermarket was used to target non-technical parameters during daily usage. This included serviceability, reliability, and derailment robustness.
    - Serviceability was based on the regulated specification of easy maintenance and ranked the concepts mainly with respect to easy access of components and easy replacement of possible wear parts. Concept E1 and E2, having similar layout with easy access to all parts from the front were both scored three. The AB-concept

was scored one due to the steel structure making maintenance such as oil check/change and brake inspection/replacement more difficult.

- Reliability was intended to show difference in reliability during daily use and was based on the number of complex components were regularly service and maintenance would be expected. Since the AB-concept had half the number of brakes, gearboxes, motors etc. it was deemed to be more reliable. Reliability was one of the parameters added from CEES´s general product development process. Derailment robustness was instead focusing on reliability after a derailing and was based on the product specification on not braking during derailment. The steel construction for E1 and E2 was deemed to better resist potential damage during derailment and were therefore ranked better than AB.

- **Manufacturing**
  The category manufacturing was aimed at ranking parameters that would make manufacturing more difficult or other factors that would increase the cost of the product and was added from CEES´s general product development process.
  - Easy manufacturing– A ranking based on estimated difficulties to mount the different components as well as difficulties to manufacture the components such as multiple parts with different fittings and machined parts. Concept AB was ranked lower due to multiple machined parts and that motor and reduction gear had to be perfectly aligned with both wheels unlike the other two solutions where each wheel and drive unit could be aligned separately.
  - Electrical installations complexity – A ranking where AB was scored highest due to half the amount motors and inverters which should reduce the electrical complexity.

- **Development**
  The category development aimed at ranking two parameters that wouldn't directly affect the performance of the product but could risk the project in other ways and was added from CEES´s general product development process.
  - Risk design – A parameter to rank possible technical difficulties that may not be solved or could possibly lead to reduced durability and costumer complaints. The main technical risks are listed under *"Possible problems"* in Table 12.
  - Complexity – Parameter for factors such as multiple suppliers to coordinate or different kinds of peripheral equipment needed. E1

was ranked lowest due to the need of separate oil cooling for each drive unit. AB was ranked highest due to a motor and inverter from Volvo combined with a reduction gear from a nearby supplier to CEES.

- **Cost**
  Cost of components, product availability and the need for modification of bought components was addressed in the cost category.
  - Price of components – Estimated price of motor, gears, and inverter. The possibility to get some components in concept E2 and E1 supplied via Volvo and take advantage of their higher volumes lowered the price significantly.
  - Off the shelf components – E1 was ranked highest due to all products being available without any modifications in Rexroth's modular concept. AB was ranked lower due to different types of modifications needed for the solution. E2 was ranked lowest due to the modified drive unit that had never been manufactured.

- **Product features**
  The possibility to use the solution for different rail gauges was addressed in the product features category.
  - Multiple rail gauges – The axial length of the motor and drive unit restricted to E1 and E2 solution to rail gauges of minimum 1435mm. The AB solution offered a minimum rail gauge of 981mmm combined and the possibility to use wider gauges with the modular design.

As seen in the concept scoring matrix, Table 13, concept AB was scored highest and was further developed.

Table 13 Concept scoring matrix.

Parameters		E1, High Speed		E2, Low Speed		AB, Centre motor	
	Weight Factor	Score	Weighted score	Score	Weighted score	Score	Weighted score
Technical Performance:							
Traction force	4	5	20	3	12	3	12
Maximum Speed	2	3	6	1	2	3	6
Differential	1	3	3	3	3	1	1
Aftermarket							
Serviceability	4	3	12	3	12	1	4
*Reliability – robustness**	4	3	12	3	12	4	16
Derailment – robustness	5	3	15	3	15	2	10
Cost							
Price of components	4	2	8	3	12	4	16
Off the shelf components	2	5	10	3	6	4	8
Manufacturing:							
*Easy manufacturing**	3	3	9	3	9	2	6
*Electrical installations – complexity**	2	3	6	3	6	5	10
Development							
*Risk design**	4	2	8	3	12	4	16
*Complexity**	3	2	6	3	9	4	12
Product features							
Multiple rail gauges	4	3	12	3	12	5	20
Sum	42	40	127	37	122	42	137

*) Parameters added from CEES´s general product development process.

7 Detail design of final concept

The chosen concept was further developed in four different areas.

7.1 Different rail gauges

As previously shown in Figure 40, the chosen concept had the possibility of using multiple different rail gauges by using steel tubes and corresponding driveshafts of different lengths in a modular way. During the detail design this was further refined. By using a hub with increased axial length combined with increased spline length, smaller variations in rail gauge (such as 1435-1520mm) can be obtained by using shims, reducing workload and parts needed for changing of gauge, Figure 43

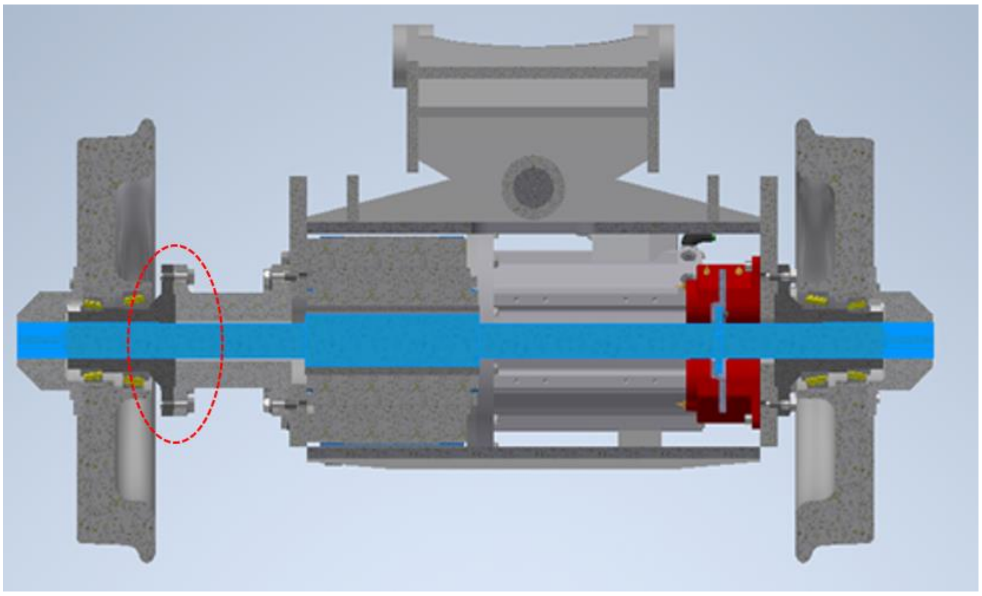

Figure 43 Cross section of the axle seen from the front. The drive axle (highlighted in blue) has extra-long splines at the hubs for enabling small changes in rail gauge with the same axle. The rail gauge can thereby be changed slightly by only adding or removing shims between wheel and steel tube (marked by red oval).

7.2 General dimensions

To get a rough estimation of feasibility, hand calculations for dimensioning were conducted for an area that was deemed critical.

7.2.1 **Bending of steel tube due to wheel load**

The modular steel tube for different rail gauges was deemed to be a critical area due to the bending stresses with a wide rail gauge and high wheel loads, Figure 44.

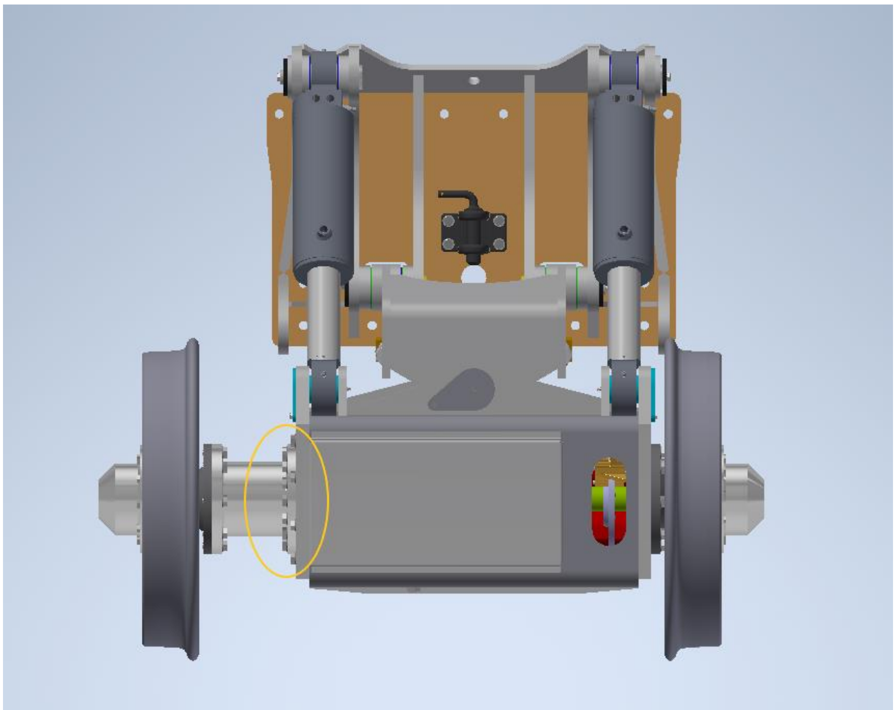

Figure 44 Front view of the axle with the critical area marked in yellow.

An assumption was mad that the hydraulic cylinders would make the welded center structure act completely rigid, Figure 45.

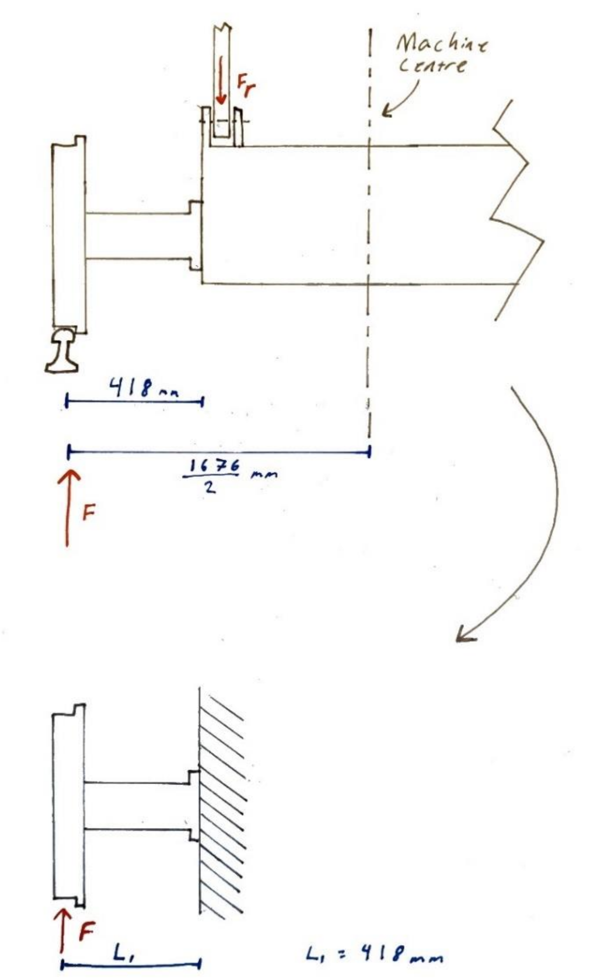

Figure 45 The welded center structure was assumed to be rigid due to the hydraulic cylinder on the side.

7.2.1.1 Bending of steel tubes – calculations

For the calculations, the load F was set to 346kN representing the full machine weight plus maximum breakout force of the excavator on one wheel in combination with maximum possible rail gauge according to product specifications. This being the maximum possible static load on one wheel.

The tube had an outer diameter, D, of 165mm. The inner diameter, d, was by the driveshaft limited to about 70mm. The total bending resistance W_b was calculated:

$$W_b = \frac{\pi}{4} * \left(\frac{D^3}{8} - \frac{d^3}{8}\right)$$

$$W_b = 407 * 10^3\ mm^3$$

The total bending torque, M_b was calculated to:

$$M_b = F * L_1$$

$$M_b = 134248\ kNmm$$

This resulting in a nominal bending stress $\sigma_{\mathrm{b\_nominal}}$ of:

$$\sigma_{\mathrm{b\_nominal}} = \frac{M_b}{W_b}$$

$$\sigma_{b\_nominal} = 330Mpa$$

Due to the flange on the tube, stress concentration can be expected at the interface between flange and tube. The stress concentration factor K_t for a circular axle subjected to bending is described in Figure 46 (Odqvist, et al., 2018, p. 371)**.**

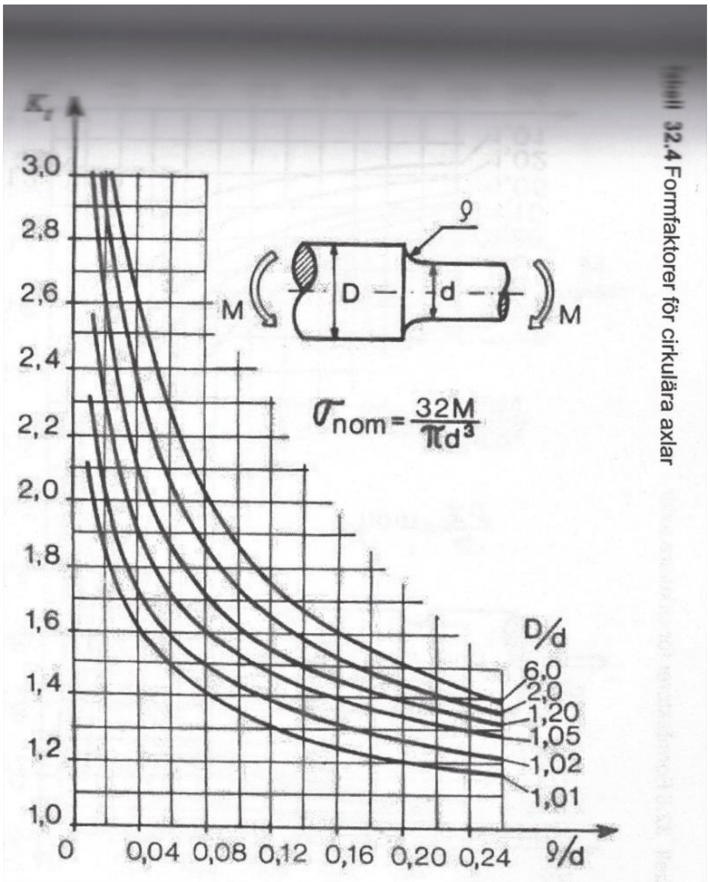

Figure 46 Stress concentration factor k_t for circular axles.

With an outer flange diameter *D*, of 220mm and an inner fillet of 7mm was the stress concentration factor, *Kt* estimated using Figure 46 to *Kt*=2. The stress concentration factor results in a higher maximal stress:

$$\sigma_{b\_max} = \sigma_{b\_nominal} * K_t$$

$$\sigma_{b\_max} = 660Mpa$$

The calculated maximum stress at 660Mpa doesn't account for fatigue nor other factors than pure bending and therefore a complementary analysis is needed such as FEA. However, the calculations show reasonable stresses, and the solution can be deemed viable, provided that high strength steel is used and some changes in dimensions e.g. larger outer diameter and larger fillets.

7.3 Tolerances and fittings

The design with a rigid axle through a gearbox connecting the two wheels sets higher demands on tolerances and fittings to avoid axial and radial displacement between parts. This is especially true for the welded centre structure connecting motor, gearbox, wheel hubs and brake callipers in one part. To reduce the amount of machining needed for tolerances, the centre structure was redesigned by adding extra material to be welded at the attachment points for motor and gearbox. Only this extra material could then be machined to fulfil the required tolerances, Figure 47.

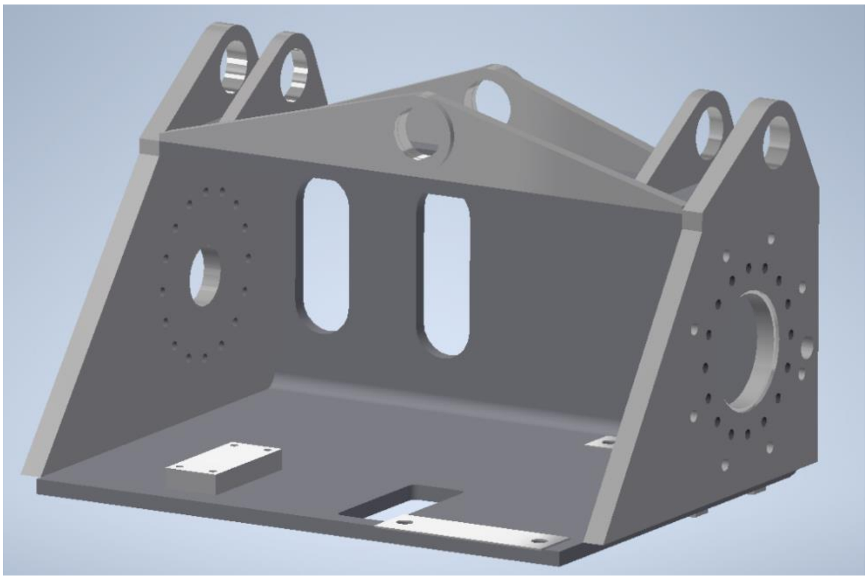

Figure 47 The welded center structure, machined only at the attachments for motor and gearbox. The interface to the tubes at the wheel is also machined for alignment of the two wheels and the driveshaft.

7.4 Brakes

Using a rigid axle connecting the two wheels gave the opportunity to only use one set of brakes per axle. For the chosen solution, the centre structure allowed for placement of the brake inside the structure and thereby being protected during derailment. The size of the centre structure limited the maximum diameter of the brake disc well as the axial space for the calliper.

The size restriction allowed using several different brake discs used within the automotive industry but limited the maximum possible brake torque. This was solved by using multiple callipers acting on one disc, Figure 48.

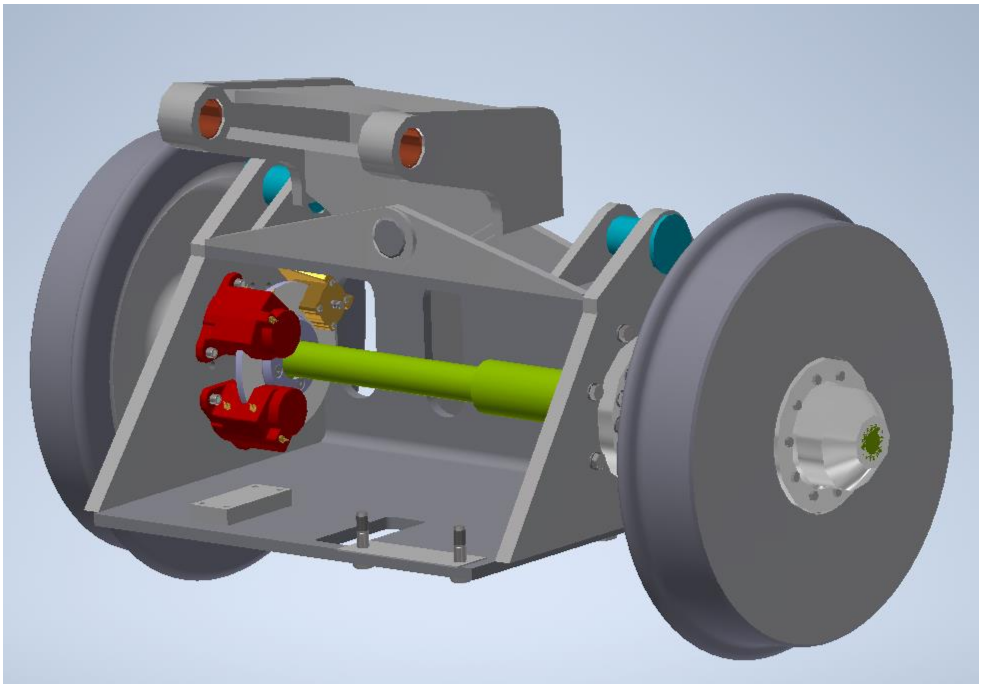

Figure 48 Dynamic brake calipers (red) and static brake caliper (brass colored) all acting on the same brake disc mounted to the drive axle (green).

8 Results

8.1 Mechanical design

The final solution consists of a single motor combined with an industry standard two stage helical gear reducer. A rigid axle through the hollow output shaft of the gear reducer connects the two wheels to the rest of the transmission. Two parallel brake callipers acting on a single brake disc mounted to the rigid axle offers sufficient brake torque to fulfil the specifications. The drivetrain can be seen in Figure 49.

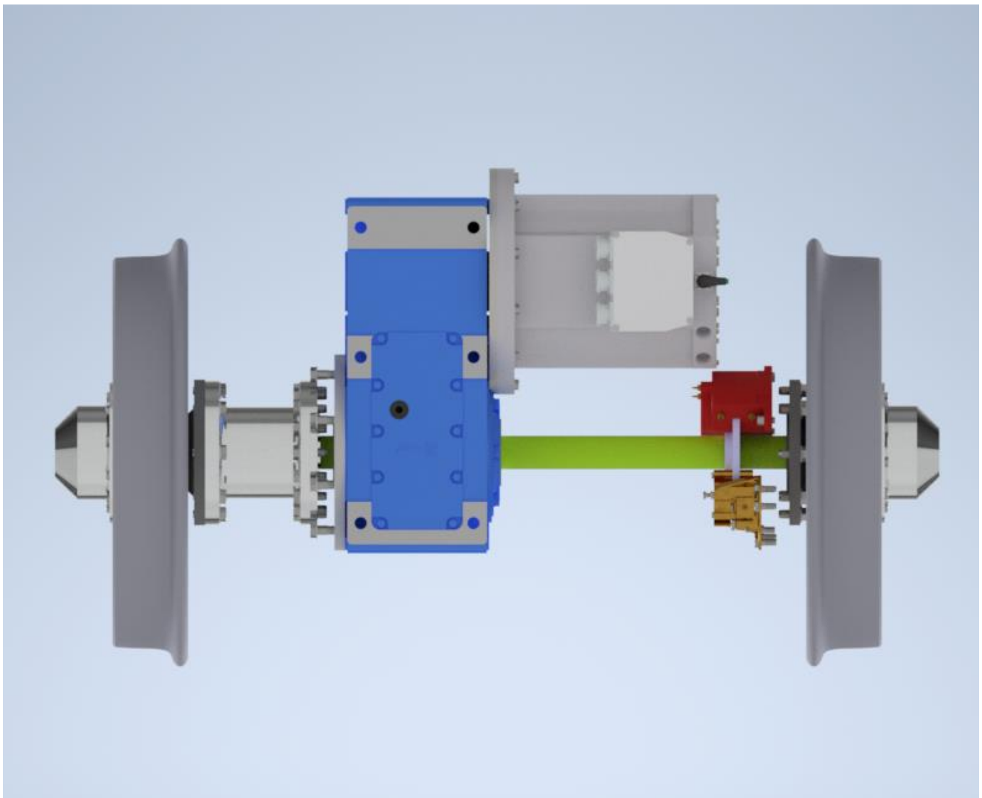

Figure 49 The drivetrain with motor, gear reducer, brake and wheels.

The solution has a welded central steel structure enclosing the transmission components and aligning the moving part relative to each other, Figure 50. Since no calculations or FEA was made for the central structure, further research is needed to verify the structural strength of this part during traveling and derailing.

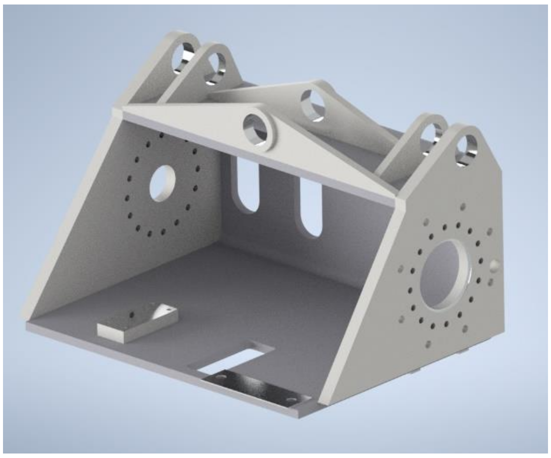

Figure 50 The welded central steel structure with machined surfaces for alignment of the various transmission components.

Various rail gauges varying between 981 and 1676mm can be achieved by using steel tube spacers of different lengths in a modular design together with drive shafts of different lengths, Figure 51.

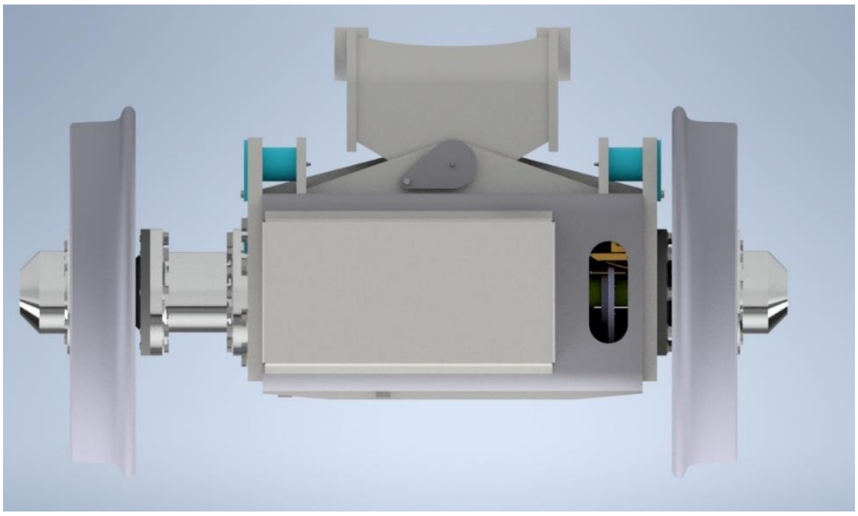

Figure 51 Different rail gauges with 1435mm for the left wheel and 981mm for the right wheel.

The inverter is placed at the front under a protective hood. DC-power cables, control signals and cooling hoses are routed from the inverter to the base machine. The interface to the base machine is kept identical as previous solutions from CEES except for new lengths of the hydraulic cylinders. The part connecting the base machine to the axel is also modified due to larger wheel diameter and larger axel overall, Figure 52.

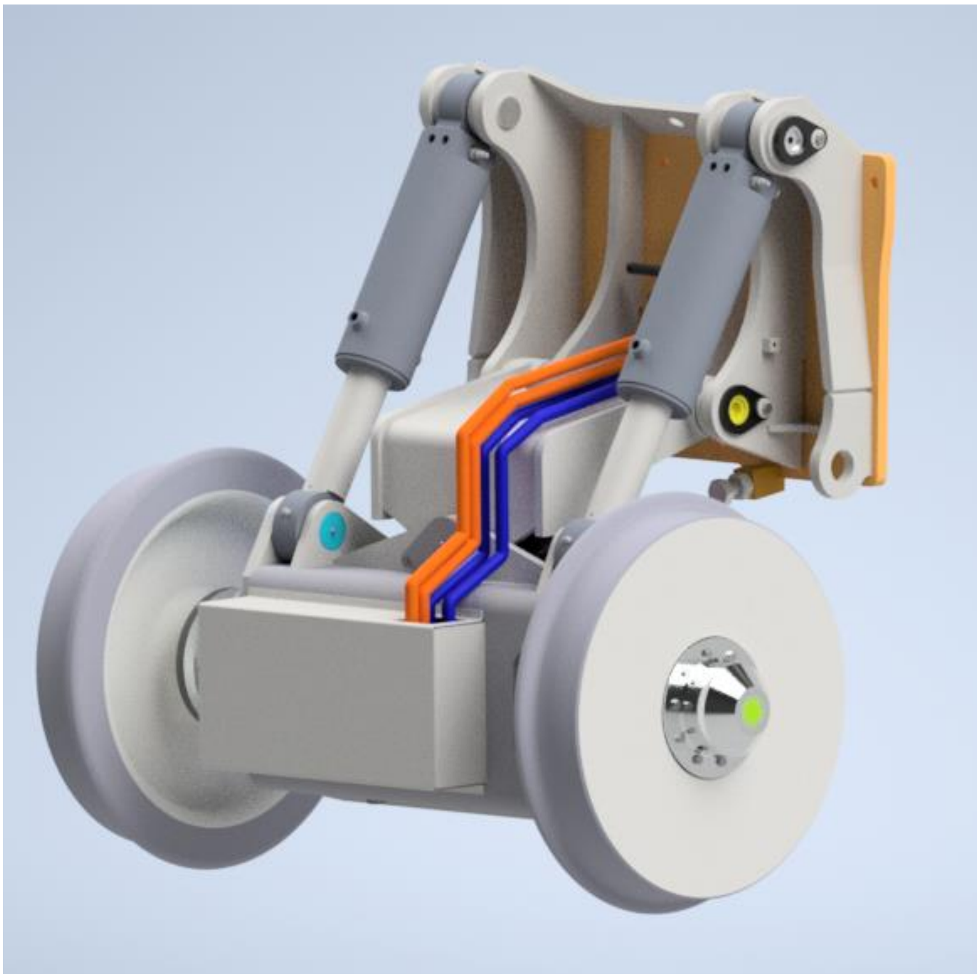

Figure 52 Cabling from the inverter and the interface to the base machine.

8.2 Results compared to the product specifications

The performance of the final solution compared to the quantified marginal and ideal values from the product specifications, set in section *4*, *Establish target specifications*, can be seen in Table 14. The solution mounted to the base machine can be seen in Figure 53.

Table 14 Comparison between result and product specifications

Metric	Margin value	Ideal value	Result
Drawbar pull	>40kN	>60kN	43kN
Maximum transportation speed	>20km/h	>30km/h	28km/h
Rail wheel diameter			700mm
Braking torque	>1500Nm	1700-2000Nm	
Minimum braking power	>50kW	>80kW	
Minimum ground clearance	>80mm	>130mm	120mm
Maximum impedance from highest point to rail	<0,05Ω		Unknown
Nominal voltage	600V		600V
Nominal DC current	<200A		580A[1]
Different rail gauges	1435mm 1520mm	981mm-1676mm	981mm-1676mm

[1] The nominal current will be restricted to 200A by the base machine as described in section 6.2.2.1.

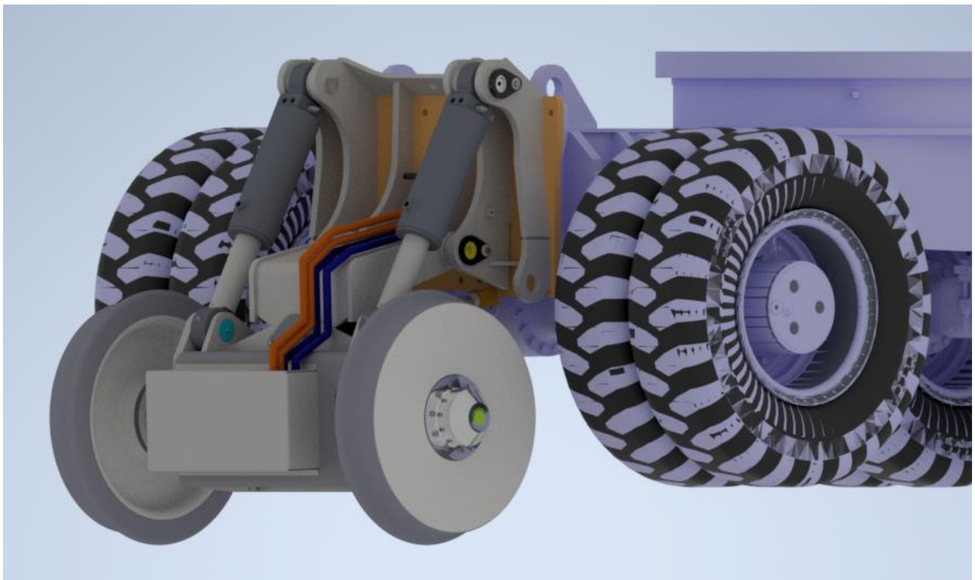

Figure 53 The solution mounted to the base machine.

9 Discussion

9.1 The solution

The chosen solution was selected after multiple iterations and refinements described in this report. The basis of the selection was U&E's method for concept generation, concept screening and concept scoring. The solution was primarily compared to the other two solutions in the concept scoring process where five main advantages could be addressed:

- Multiple rail gauges possible
- Price of components
- Risk design
- Reliability - robustness
- Complexity

The developed concepts and solutions developed throughout the project were all based on components available today and with the rapid development within electrification today, is it possible that the evaluation soon will be partly outdated due to new components changing the prerequisites. However, some of the main advantages of the chosen solution can be expected to remain static such as the ability to use the narrowest rail gauges, where solutions with different types of wheel drives are limited by the axial length of the motors. The reliability advantage of the solution can also be expected to remain static since it was mainly based on the comparison between using one or two motors per axle.

During concept scoring were parameters added from CEES´s general product development process. This were parameters that hadn't been or poorly addressed in the product specifications. The product specification of using proven technical solutions aimed at reducing risk and complexity within the project but was vague. More precise product specifications such as the parameters added from CEES´s general product development process with risk design, complexity and manufacturing would have been more useful and could have reduced the risk of criterions changing throughout the different evaluation steps due to slight changes in interpretation of the specifications.

The specification of using multiple rail gauges was listed early in the project and was given a relatively heigh weight in the concept scoring. However, the scoring did not account for the 1435 and 1520mm gauge being the most common. A more refined specification to better differentiate among the different gauges in respect to commonness would have been preferable.

The price estimation of the three solutions was based on a combination of list prices from suppliers, cost prices from Volvo and estimations of prices based on prices of components with similar performance. The concepts using components from Volvo (concept E2 and AB) assumed advantage could be taken from Volvo's higher volumes resulting in a price of components close to Volvo's cost prices and this resulted in both concept E2 and AB being cheaper than E1. The approach of using the exact same components as Volvo gives the possibility to take advantage of knowledge within Volvo and simplifies spare parts distribution but also heavily restricts the available components when designing. The Volvo components are ordered with specific configurations of motor sizes, windings, output axles etc to fit the standard Volvo products and these components might be suboptimal when used in other applications in respect to size and performance. By using non-Volvo components, suppliers can often modify both gears and motors to fit the application with the drawback that the smaller quantities results in a significant higher price.

In this project, rheostatic- and other types of non-regenerative electric braking was only briefly investigated and were in an early stage deemed too complex since the technology could mainly be found on trains with cooling fans to dissipate the generated heat. Later, the braking with conventional disc brakes turned out to be a problem with the drive unit concepts mainly due to the high rotational speeds. The brakes also increased the minimum possible axial length of the drivetrain even further. More research is needed into rheostatic braking since it can remove the need of frictional brakes completely, simplifying packaging thanks to a more flexible placement of brake resistors.

The motors investigated in this project typically have an efficiency varying with different torque and rotational speed and thereby the total driveline efficiency could be optimised for different work cycles by calculating an optimal gear ratio. This was not done. The electric traction system is the first electric, non-hydraulic traction system at CEES and can be seen as a proof of concept rather than a complete product. Optimizing efficiency was therefore seen as a future step in the development.

9.2 The project

At the beginning of the project, two alternative project plans were set. The plan was to focus the first half of the project on product specifications, working principle and to make a rough design of the solution. The second half of the project was divided into two different paths with either prototyping and test of prototype or if deemed impossible, further refinement with FEA, 2D-drawings, and a verification to the system specifications. The purpose of the two project plans was to have some flexibility in the project due to the high uncertainties involved in the development of new products.

As seen in the actual time plan, the first phase of project turned out to be more complex than expected and this delayed the first phase multiple weeks. The original plan of building a full-size prototype in the second phase of the project restricted the “refinement of concept” activity to only focus on concepts that could be implemented with components available today in small order quantities. The fact that the development was a master thesis and not a real development project also lowered the interest among manufacturers of different components. The difficulties in finding suitable components resulted in the “refinement of concept”-phase taking significantly longer time and that the initially three different concepts from the concept screening were reduced to only two different working principles. Knowing the outcome of not building a prototype within the project, a better approach might have been focusing more on different working principles such as in the concept generation and keep the solutions in a more conceptual way. This would have resulted in a solution further away from a “manufacture-ready” solution, but the results might have been more persistent, being more relevant in the future when the solution might be implemented with components available at that time.

For this project, U&E´s method for a generic product development was used. The method describes a structured way of concept generation and concept selection and worked well for this project. One important advantage was the concept scoring where the components and not only the working principle of each concept affected the results. The concept scoring matrix makes the decision clearer to the reader and gives the possibility to see how components with similar working principles, but other performance can affect the scoring.

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Appendix A Time plan

A.1 Planned and actual time plan

The actual time plan followed the planned time plan well until the refinement of concepts. The refinement of concepts and 3d-modeling took significantly longer time than expected due to difficulties finding suitable components. This delayed the rest of the activities throughout the project. The FEA and 2D-drawings were originally planned within phase two but were disregarded due to the delay of the project and that those parts would be outdated if the project in the future were implemented with similar but different components. The planned and actual time plan can be seen in Figure 54.

Figure 54 Planned and actual time plan.

Appendix B Studied standards

An overview of the standards studied as well as what part(s) that were deemed applicable.

Standards for electrical drivetrain Road-Rail EWR150/170

Color scheme:

Railway applications:
EMC Standards:
Control systems
Electrical drivetrain:
Volvo Standard
Misc:

Standards:	Applicable?	Applicable parts:	Available at CEES?	In 15746 Reference list?	Comment
SS-EN-15746-1:2020 - Railway applications - Track - Road-Rail machines and associated equipment Part 1: Technical requirements for traveling and working	YES	5.2 Rolling stock gauge 5.5 Safety against derailment 5.6 Stability and prevention of overturning 5.9 Running gear 5.17 Electromagnetic compability 5.19 Failure recovery conditions Annex A: Special national conditions	YES	YES	General dimensions and working limits Tests against derailment Calculations or tests to prove stability against overturning Wheel load, wheel profiles, wheel arrangements No EMC test needed if all additional electrical components fulfill EN 50121-3-2 Towing device and possibility to ratracts items within stated gauge National requirements, eg. max 20km/h in Sweden
SS-EN-15746-2:2020 - Railway applications - Track - Road-Rail machines and associated equipment Part 2: General safety requirements	YES	5.10 Prevention of derailment 5.14 Operators' controls and indicators 5.16 Electrical system 5.24 Braking systems Annex A: List of significant hazards Annex ZA: Relationship between MD 2006/42/EC and	YES	YES	Part 2 is harmonized with the European machinery directive 2006/42/EC. Railhead clearing device Warning signals and performance levels according to EN 13849-1:2015 Electrical equipment shall be in accordance with EN 474-1. Wiring, cables, and conductors according to 60204-1. Minimum distance to overhead cables. Equipotential bonding No. of braking systems. Maximum stopping distance List of significant hazards for the risk analysis
SS-EN-15746-3:2020 - Railway applications - Track - Road-Rail machines and associated equipment Part 3: Technical requirements for running	NO		YES	YES	Not needed since machine is not "...designed and intended to operate signalling and control systems...". Therefore is the machine not a running machine, only a working/traveling.
SS-EN-15746-4:2020- Railway applications - Track - Road-Rail machines and associated equipment Part 4: Technical requirements for running, traveling and working on urban rail	Possibly		YES	YES	The machine is not intended for urban use in this early stage. Future versions might be adapted for urban use
EN 14033-1:2017 Railway applications - Track - Railbound construction and maintenance machines - Part 1: Technical requirements for running.	NO		YES	YES	Only applicable for machines exclusively running/working on rail
EN 14033-2:2017 Railway applications - Track - Railbound construction and maintenance machines - Part 2: Technical requirements for traveling and working.	NO		YES	YES	Only applicable for machines exclusively running/working on rail
EN 14033-3:2017 Railway applications - Track - Railbound construction and maintenance machines - Part 3: General safety requirements.	NO		NO	YES	Only applicable for machines exclusively running/working on rail
EN 50153:2014, Railway applications - Rolling stock - Protective provisions relating to electrical hazards	YES	6.2 Protective bonding	YES	YES	Specifications of equipotential bonding
SS-EN 13849:2016 Safety of machinery - Safety related parts of control systems	YES	All	YES	YES	Safety requirments for control systems. Mostly software but might require some hardware
SS-EN-50121-3-2 Railway applications - Electromagnetic compability - Part 3-2:Rolling stock - Apparatus	YES	All	YES	YES	Limits and test methods for immunity and emmision. Harmonized with EMC 2014/30/EC Need to verify that all additional electrical component fullfill this standard.
EN 13309 - Construction machinery - Electromagnetic compability of machines with internal power supply	Possibly		NO	YES	Test methods and acceptans criteria for evaluation of EMC for construction machinery. From EN 15746: "Except where a host vehicle is already stated to be compliant with European Automotive EMC directive 95/54/EC, machines shall meet the requirements of EN 13309 or EN 50121-3-1:2017, Clause
SS-EN-50121-3-1 Railway applications - Electromagnetic compability - Part 3-1:Rolling stock - Train and complete vehicles	Possibly		NO	YES	Emission and immunity requirements for all types of rolling stock From EN 15746: "Except where a host vehicle is already stated to be compliant with European Automotive EMC directive 95/54/EC, machines shall meet the requirements of EN 13309 or EN 50121-3-1:2017, Clause
ISO 13766-1 - Earth-moving and building construction machinery - Electromagnetic compability of machines with internal electrical po	NO		YES	NO	Not needed since EN 15746-1 states that if host vehicle is compliant with EMC directive 95/54/EC, additional
ISO 60204-1 Safety of machinery - Electrical equipment of machines -Part 1: General requirements	YES	6.2 Protection against direct contact 6.3 Protection against indirect contact 7.2 Overcurrent protection 7.3 Protection of motors against overheating 8.2 Protective bonding 13 Wiring practice	YES	YES	General standard for electrical machines. Enclosures, insulation and protection against residual voltages. Discharge times Protection against electrical shocks from metal parts due to poor insulation. Done by insulation, separation or automatic disconnection Conductors to be disconnected, placement, devices and ratings Either: Overload protection, over-temperature protection och current-limiting protection What parts to be bonded, type of cable Connections, colour of cables, bending radius, routing etc
IEC 61800 - Adjustable speed electrical power drive systems.	NO		NO	NO	Part 1 for DC motors. Part 2 for AC motors but not applicable for vehicles. Part 3 for EMC. Part 4 voltages over 1000V. Part 5-1 safety requirements Electrical etc. Part 5-2 Safety requirements functional Not needed since there will most probably be an AC motor
ISO 14990-1:2016 Earth-moving machinery - Electrical safety of machines utilizing electric drives and related components and systems	Possibly		NO	NO	ISO standard. The project mailny focus on European market
ISO 20474-1:2017 Earth-moving machinery - Safety Part 1: General requirements	Possibly		NO	NO	ISO standard version of EN 474
SS-EN 60349 Electric Traction - Rotating electrical machines for rail and road vehicles -Part 1 Machines other than electronic converter-fed alternating current motors	NO		NO	NO	Only one part of 60349 is needed. Standard for performance evaluation
SS-EN 60349 Electric Traction - Rotating electrical machines for rail and road vehicles -Part 2 Electronic converter-fed alternating current motors	NO		NO	NO	Only one part of 60349 is needed. Standard for performance evaluation
SS-EN 60349 Electric Traction - Rotating electrical machines for rail and road vehicles -Part 4 Permanent magnet synchronous electrical machines connected to an electronic converter	NO		NO	NO	Only one part of 60349 is needed. Standard for performance evaluation
Volvo STD 515-0003 Electro-magnetic compability EMC	NO		YES	NO	Test methods and critereas related to EMC
Volvo STD 871-0003 Safe operation of electrical installations in electric products	NO		YES	NO	Instructions for working on electrical parts.
EN 474-1:2006+A6:2019, Earth moving machinery - Safety - Part 1: General requirements	YES	5.17 Electrical and electronic systems 5.22 Maintance Annex A: List of significant hazards	YES	YES	Some general informmation regarding electrical systems and wiring. Minimum IP55, overcurrent protection Design for safe maintance and easily accessible Hazards for risk analysis
EN ISO 12100:2010 Safety of machinery - General principles for design - Risk assessment and risk reduction.	YES	All	YES	YES	Standard for risk analysis
IEC 61373:2010 Railway applications - Rolling stock equipment - Shock and vibration tests	Possibly		NO	NO	Requirements for vibration and shock testing of electrical and pneumatic components for railway vehicles

Appendix C Braking calculations

Based on the braking requirements in EN 15746-2, the minimum required braking torque was calculated

C.1 Prerequisites and assumptions

Base for the braking calculations was the braking requirement of maximum 27- and 55-meter braking distance from 20 and 30 km/h respectively according to EN15746-2. The braking was assumed to be at constant brake pressure. A simple model of the machine was sketched, Figure 55.

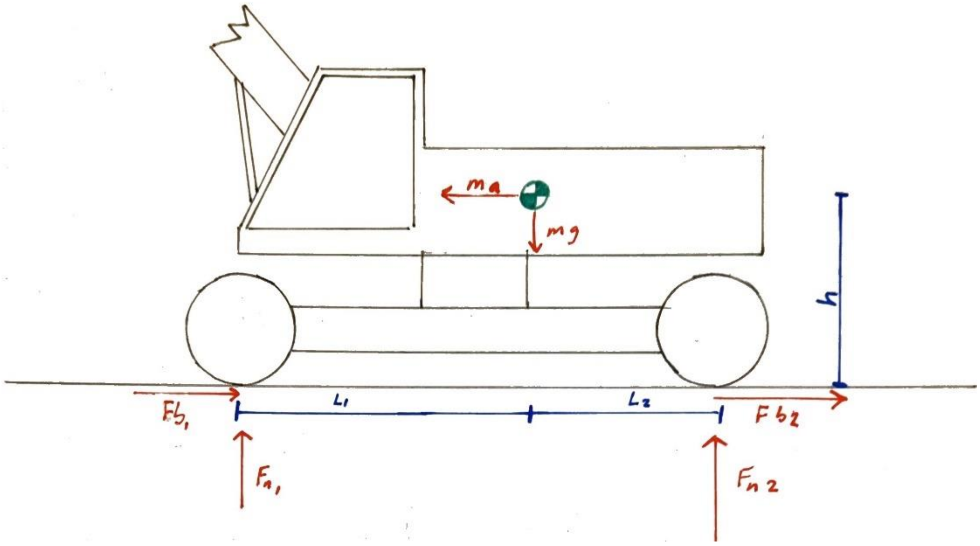

Figure 55 Sketched model of the dimensions and forces acting on the machine at braking.

Parameter	**Unit**	**Value**
Machine weight [m]	Kg	30 000
Maximum speed [v]	m/s	5,56/8,33
Maximum allowed stopping distance [L_{stop}]	m	27/55
Friction steel-steel static [μ_{static}]	-	0,25
Friction steel-steel sliding [$\mu_{sliding}$]	-	0,15
Wheel diameter [d]	m	0,7
Dimension h in Figure 55	m	1,5
Dimension L_1 in Figure 55	m	2
Dimension L_2 in Figure 55	m	2

C.1.1 Minimum required brake torque

The machine traveling at 20 respectively 30 km/h have a kinetic energy, W_k, of:

$$w_k = \frac{mv^2}{2}$$

The minimum required braking force, F_{Brake}, for stopping at the specified braking distance:

$$F_{Brake} = \frac{W_k}{L_{stop}}$$

Required average braking torque, $F_{TorqueWheel}$, per wheel was calculated using wheel diameter and total required braking force:

$$F_{TorqueWheel} = \frac{F_{Brake} * d}{8}$$

The average braking torque $F_{TorqueWheel}$, was calculated to 1500 and 1657 Nm respectively. To verify the possibility to have the same brake torque for both front and rear axle and thereby lowering the maximum required brake torque, the deacceleration and weight distribution was calculated during braking.

Minimum required deacceleration, *a*, for stopping at the specified braking distance:

$$a = \frac{F_{Brake}}{L_{stop}}$$

The required deacceleration, *a*, was calculated to 0,57 and 0,63 m/s^2 respectively.

The deacceleration resulted in the following normal forces action on the wheels:

$$F_{n1} = \frac{m * g * L_2 + m * a * h}{L_1 + L_2}$$

$$F_{n2} = m * g - F_{n1}$$

$$F_{n1} = 153730N$$

$$F_{n2} = 140870N$$

Maximum possible braking torque at sliding, $F_{MaxTorque}$, for the lowest loaded wheel:

$$F_{MaxTorque} = F_{n2} * \mu_{\text{sliding}} * \frac{d}{2}$$

$$F_{MaxTorque} = 7396Nm$$

Since the average braking torque per wheel was lower than the maximum possible braking torque of the lowest loaded wheel, the product specification for braking torque was set equal to the average braking torque per wheel.

C.1.2 Minimum required braking power

For calculating braking power for regenerative braking, a constant deacceleration was assumed. A maximum deacceleration time, s, was calculated using the previously calculated deacceleration:

$$S = \frac{v}{a}$$

The average power generated during braking, P_{avg}, was then calculated using the total kinetic energy:

$$P_{avg} = \frac{w_k}{s}$$

$$P_{avg20km/h} = 48kW$$

$$p_{avg30km/h} = 79kW$$

The momentary brake power was also numerically calculated using the kinetic energy, w_k, to calculate the momentary velocity:

$$V = \sqrt{\frac{w_k * 2}{m}}$$

The momentary velocity was then used to calculate the momentary brake work, W_{Brake}, and brake power, P_{Brake}:

$$W_{Brake} = F_{Brake} * V * dt$$

$$P_{Brake} = \frac{W_{Brake}}{dt}$$

The momentary brake work resulted in a new kinetic energy, W_{k2} and thereby a new velocity:

$$W_{k2} = W_{k1} - W_{Brake}$$

The results were illustrated in four graphs that can be seen in Figure 56.

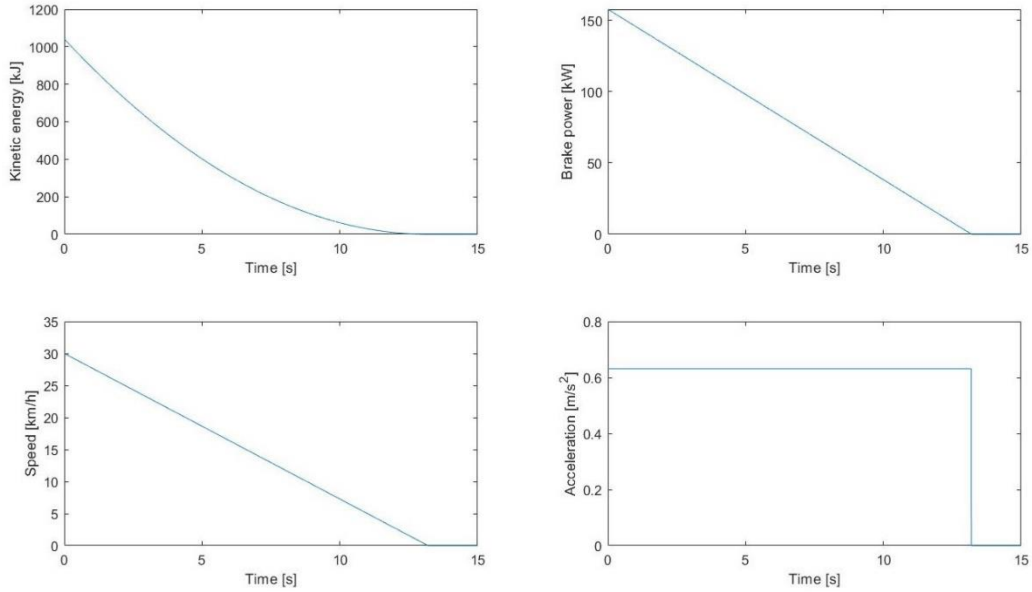

Figure 56 Constant deacceleration from 30km/h. As seen the momentary brake power is linear between 158 and 0 kW with an average value of 79kw.

Appendix D Product specifications

A full list of the product specifications including customer need, metric, marginal and ideal value.

Table 15 A full list of product specifications.

Need:	Metric:	Unit	Marginal value	Ideal value
Customer needs:				
High pulling force	Drawbar pull	kN	>40	>60
Fast during transport	Maximum speed	km/h	>20	>30
Good "precision" in brake	subj			
Good traction and stability	One pendulum axis			
	Large rail wheel diameter	mm	>500	>600
	Differential	Yes/No	No	Yes
Doesn't break when derailing	All parts protected against damage from derailing	Yes/No	Yes	Yes
Regulations:				
Max. breaking distance 27/55m	Min. breaking torque	Nm	>1500	1700-2000
	Min. breaking power	kW	>50	>80
Minimum two independent brakes, of which one is failsafe	Specified in EN 15746-2			
Minimum allowed ground clearance according to Trafikverket	Min. ground clearance specified in TDOK 2015:0143	mm	>80	>=130
Railhead clearing device	Must be equipped with a device for railhead clearing	Binary		
Equipotential bonding	Maximum impedance from highest point of machine to running rail	Ohm	<0,05	

Easy maintenance	"Machines shall preferably permit lubrication from the ground"			
	"Components which require frequent maintenance shall be easily accessible"			
Failure recovery	Machine must have towing device at both ends			
Electrical safety	Live parts shall be located inside enclosures of at least IP2X			
Integration to base machine:				
Nominal voltage	Nominal voltage for inverter and motors	V	600	
Nominal DC current	Nominal DC current to motors	A	<200	
No interference between base machine and product during any possible movement				
Specifications from the company				
Possible to use on different track gauges	Rail gauge	mm	1435 1520	981 1000 1067 1435 1520 1600 1668 1676
Manufactured of standard components	Keep lead time low and spare parts availability high by using of the shelf parts as much as possible			
Designed of proven technical solutions	Use mainly working principles from heavy duty machinery or construction equipment.			