Complementary Hydraulic Drive with Four Motors

- Improving Speed and Startability

Master’s thesis in Product Development

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Gothenburg, Sweden 2016

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Abstract

A general timber truck encounters all from dry asphalt roads to muddy or ice forest roads depending on the season. Having high accessability is therefore cruicial in order to retrieve the timber from the woods, but it is also important to have an efficient solution for the long journey back to the final destination of the timber.

Several truck manufacturers have the past years announced work with C.H.FWD (complementary hydraulic front wheel drive) where the C.H.FWD solution, compared to the conventional M.FWD (mechanical front wheel drive), is compatible with more truck combinations at the same time as a weight reduction of atleast 400 kg is done. A large restriction with the C.H.FWD however is the low top velocity that can be achieved if an acceptable torque is desired. This velocity problem can be solved by adding two additional motors on the otherwise non driven pusher or tag axles, i.e. a hydraulic gear can be achieved. This hydraulic gear can if correctly configured result in higher top velocities and unchanged torque without changing the volumetric flow or pressure within the system. By engaging only two motors, one axle, lower total volumetric motor displacement is obtained and a higher velocity can be achieved. When engaging all four motors, two axles, a larger volumetric displacement will be obtained and also a larger total torque with the consequence of lower top velocity.

Today Volvo’s trucks are not adapted to have C.H.D (complementary hydraulic drive) installed on any axle which results in less optimized solutions. None the less, in this thesis work it has been proven that both the RAPDD-GR (the pusher axle configuration in the report) and RADT-GR (the tag axle configuration in the report) configurations are compatible with the C.H.D. The RAPDD-GR combination offers with today’s wheel end layout more available space to fit the routing of the system compared to the RADT-GR. Virtual studies imply that only one single part will be affected by the implementation of hydraulic drive on the RAPDD-GR. This compared with the tag axle which would require several smaller and larger modifications in order to be compatible with the C.H.D.

Acknowledgements

This project, as many other projects has required the help from several different areas and would not have been possible to execute with a good result without the acquired help which has been obtained throughout the project.

Firstly I want to thank my supervisor at the company Emil Pettersson, and also Dennis Persson and Heléne Jarlsson which have been very helpful the entire project. Also my project leader Lena Larsson at Volvo requires an extra gratitude to the opportunity of doing this thesis work.

I also want to bring forth my gratuítude to my examiner Lars Lindkvist at Chalmers which also has been my supervisor along the project.

Special thanks also go to:

Jörgen Ahlberg from VCE and Michael Andersson from Specma have both been of great value regarding questions of routing of the system.

Many thanks also goes to Jan Hendriks from the suspension department, Richard Söder at the Vechile productivity & truck functions department and to all other Volvo employees who have aided in answering my questions and thoughts.

Terminology

AUXPark - Auxiliary parking brake

BOM – Bill of material

C.H.D – Complementary hydraulic drive

C.H.FWD – Complementary hydraulic front wheel drive

CHH – High – Chassis height high

CHH – Med – Chassis height medium

FAA20 – Front axle arrangement, two axles zero driven

GTA – Global transport application

M.FWD – Mechanical front wheel drive

M.RWD – Mechanical rear wheel drive

PTO – Power takeoff

RADT-GR – Rear axle arrangement, driven axle & tag axle – global rear axle air suspension

RAPDD-GR – Rear axle arrangement pusher axle, driven axle & driven axle – global rear axle air suspension

REPTO – Rear engine power take-off

VCE – Volvo Construction Equipment

FH-1825 – Test truck no. 1825 having FH cabin

Table of Contents

1.0 Introduction

The following introduction presents the background to why there is a need for a prestudy regarding C.H.D on other axles than the front axle, a purpose describing what is to be achieved, and the delimitations which serves as a frame of project. Based on these three areas a problem statement is presented, describing the questions which are to be answered throughout the project timeline.

1.1 Background

Complementary front wheel drive is today offered by several truck manufacturers, both as mechanical and hydraulic configuration. By having front wheel drive in addition to the rear driven shaft(s), accessibility can be improved when driving with an unloaded trailer or when driving without a trailer. This is because in these situations the main weight is located on the front axle, which also is subjected to the highest normal force.

A truck which transports timber serves 50-95 percentage of its lifetime driving on the top gear where increased number of driven axles results in reduced fuel efficiency (Hedman, 2016). By enabling the option to reduce the number of driven axels when desired fuel consumption can be reduced and increased transport efficiency can be achieved.

Mechanical drive today is achieved by connecting one end of a propeller shaft to the gear box outlet and to a set of wheels on the other. C.H.D is achieved by a pump which drives one or several motors with high pressure oil led via pipes and hoses. Using C.H.D compared to mechanical drive, reduction of the total weight of the vehicle can be expected. C.H.D also requires less packaging space and when disengaged results in almost no loss of fuel efficiency (Poclain Hydraulics, n.d.).

Today the C.H.FWD is theoretically limited to 26,5 km/h in unloaded condition and is electronically limited to 30 km/h. This means that for higher velocities the C.H.D cannot offer any support and the truck relies solely on the M.RWD (mechanical rear wheel drive). The maximum rotational speed of a hydraulic motor is dependent on its volumetric displacement, where reducing each motor’s hydraulic displacement and increasing the number of motors, higher speed can be achieved while maximum produced torque is kept unchanged.

1.2 Purpose

The aim with this thesis is to further develop the hydraulic drive system, C.H.FWD, which has been the subject of several thesis works prior to this, by performing a pre study of hydraulic drive on the pusher/tag axles as an addition to hydraulic front wheel drive. The purpose can be divided into four different parts:

Analyse and assess different hydraulic motor sizes and each motor sizes expected start ability and top velocity.

Dimensioning of hydraulic hoses, due to changes that will be performed in the system the volumetric flow can also change as a consequence to this. Therefore new calculations of the volumetric flow, as a verification whether the same pipes and hoses as are fitted onto FH-

1825 (Test truck no. 1825, having FH cabin) can be used or new dimensions, needs to be implemented.

The valve block which controls the oil flow on FH-1825 and previous test trucks, is designed to function using two motors and is not compatible with a four motor solution. A new solution of how to control the oil flow for four motors needs therefore to be derived where a hydraulic scheme also needs to be developed.

Finally routing of the hydraulic system for RAPDD-GR and RADT-GR needs to be done, where possibilities and obstacles needs to be identified and assessed.

1.3 Problem statement

Based on the purpose of the thesis, following problems which are to be answered have been stated:

- Will there be sufficient startability once desired velocity is achieved using the pump specified?
- Will there be a need of new hydraulic hoses and pipes, or can the same dimensions as for FH-1825 be used?
- How can the valve block be configured to suit intended purpose while simultaneously be relatively cheap to develop?
- Is there a viable packaging solution for both RAPDD-GR and RADT-GR which does not require massive changes on already existing parts?

1.4 Delimitation

Due to the large number of configurations being offered by Volvo, as standard and customized solutions regarding; axle arrangement, chassis height, etc. the following delimitations have been determined:

- The following rear axle arrangements will be the focus
  - RAPDD-GR & RADT-GR
- The following chassis heights will be the focus
  - CHH-MED, (chassis height medium)
- Only steerable tag axle 10 t capacity, and pusher axle specified for 7,5 t and 9 t will be considered and included.
- FAA20 (front axle arrangement, two axles, zero driven) will not be included or considered in the thesis.
- Already existing hubs and knuckles shall not be modified in any way.
- Interposed pulley on RADT-GR will not be taken in consideration.

2.0 Methodology

Product development contains several different phases, which all can be iterated until a satisfactory result can be achieved. Between each of these phases are so-called “tollgates”, which needs to be passed, these gates works as go or no go occasions. Where if given a go, passage to next gate is granted, if given a no-go rework can be needed to pass or the project can be terminated. The different stages are illustrated in Figure 1, where the reader should keep in mind that this is just one of all different ways of defining a development process and was deemed suitable for this project.

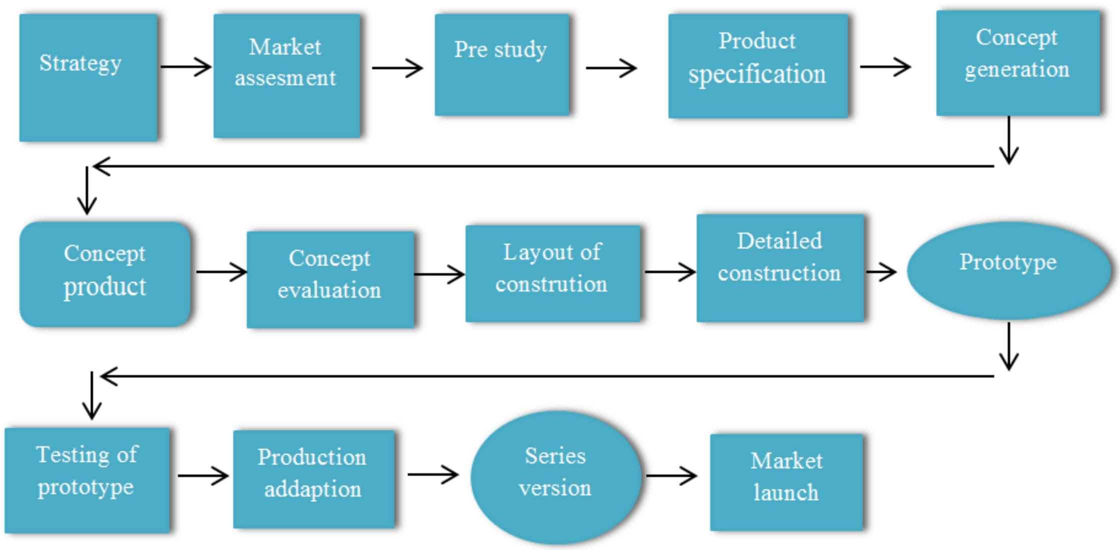

Figure 1 Method

This thesis project only involves the stages until detailed construction, the stages from prototype to market launch will not be involved. This is a theoretical study and should analyse whether a C.H.4WD system as specified in this report is possible to realise and what could be expected of the system in that case.

Competitor assessment was done on four truck manufacturers which have announced their work with C.H.FWD and three aftermarket solutions, also Volvos current C.H.FWD has been assessed. Research for truck manufacturers developing C.H.D for pusher and tag axle have been conducted without any results. The Market assessment was primary based on a previous thesis works which includes interviews regarding C.H.FWD and gathered knowledge within the project participants. In this thesis work only one phone interview has been conducted and was with Bertil Andersson who said that they have experienced 100 % more efficiency since the implementation of M.FWD.

The pre study and knowledge gathering regarding trucks and hydraulics have mostly been based on previous work regarding C.H.FWD at Volvo and literature provided by the hydraulics supplier. The knowledge gathering was divided into two parts; literature based and study visits. The literature based knowledge contains both self-studies and a one day education at the hydraulics supplier. The study visits have been conducted at three different places, VCE, Volvo Truck’s production line and at one of Volvo Truck’s workshops the GB workshop. The first study visit at VCE in Braås, involved general knowledge regarding hydraulics and fittings and worked as an introduction to the subject. The second visit was at Volvo truck manufacturing facilities at Tuve where the production was discussed and seen to enable some manufacturing considerations in the development process.

The product specification contains the requirement list, where all requirements are listed. A separate list with only requirements which are to be met by the hydraulic supplier has been compiled. This list has however not been included in this report due to confidentiality.

During the concept generation a half day workshop was conducted, where five participants with different amount of knowledge regarding the project were involved. During the workshop the generation method used was brainstorming which resulted in a numerous numbers of concepts where similar/equal concepts were grouped together. The developed concepts were part solutions to a complete solution, the complete solutions were derived using a version of a morphological matrix.

The concept evaluation was conducted using a Kesselring matrix with the added criteria "perceived feeling". This was added to assess the participants overall feelings of the concepts since the weighting of each criteria is not derived based on facts and therefore might lead to that wrong concept wins.

When developing the final concepts detailed construction, the CAD software Creo Parametrics was used throughout the entire step since this was deemed most suitable due to the problems complexity. The parts used are standard parts as far as possible in order to reduce the total cost and reduce the complexity. Adaption to production has been of consideration throughout the entire process and mainly in this step, this in order to have a solution which requires less changes to suit the production which hopefully will increase chances of series production (Johannesson, et al., 2004).

3.0 Technical references

In this chapter relevant and necessary theoretical knowledge regarding the area is presented and discussed. This in order to offer the reader larger possibilities to quickly learn needed base knowledge and terms in order to understand the discussed topics throughout the report.

3.1 Previous work at Volvo

There have already been many years of work in the area regarding C.H.D, where the area of interest has been implementing the C.H.D on the front axle. The first field test vehicle was in service from 2009 to 2014, where the hydraulic motors were driven by a 130 cc hydraulic pump from Poclain Hydraulics.

In 2014 the first field test vehicle with a hydraulic pump provided by the hydraulics supplier was realised.

What these two trucks have in common is that the hydraulic pump retrieves its torque from a PTO (power take off) via a prop shaft. This configuration results in increased losses and increased weight. By instead mounting the pump directly onto the PTO, the prop shaft can be removed which also removes the extra weight and losses that follows with the prop shaft. Numbers of tests with the configuration where the pump is mounted directly to the PTO is set to launch sometime in 2016 with the test truck FH-1825 (Bertilson & Östman, 2015).

3.2 Market assessment

Based on earlier conducted interviews it was evident that the customers of Volvo desires additional driving force during low speed and while turning or cornering in less good conditions. It was also brought up by a user of the system that it could be a necessity to have the possibility to reach higher speeds than 25 km/h (Jonsén, 2011).

C.H.FWD is aimed for the customers who drive where there are poor road conditions and tough terrain and with high demand on good traction e.g. timber trucks. This segment represents less than one percentage of Volvo’s production of trucks, but the volume is still significant enough to develop this type of solution (Bertilson & Östman, 2015).

In order to handle the customer segments regarding e.g. trucks ploughing snow or other applications where a lot of drive force is needed even at higher velocities, hydraulic drive specified for velocities up to 50 km/h is a requirement (Pettersson, 2016).

3.3 Competitor Assessment

In the following section, known competitors in the field of C.H.D are discussed, where system specification and solution of relevant routing is analysed.

There are today four other truck manufacturers whom are working with C.H.FWD but not a single known truck manufacturer that is developing C.H.D on other axles on the truck than the front axle(s). One patent from 1974 was found regarding hydraulic drive on trailer axle(s) to increase accessibility with trucks fitted with trailer (Greene, 1974). And one company which currently is working with fitting C.H.D on trailers, which will be reviewed in section 3.3.2.3.

Presented information regarding the different solutions can differ between the manufacturers since accessable public material varies a lot in informational quality.

3.3.1 Truck manufacturers solutions

The following section regards found truck companies whom have announced on-going work with C.H.FWD.

3.3.1.1 MAN - Hydrodrive

MAN was in 2005 the first truck manufacturers who could offer the C.H.FWD solution to the market as an option to the mechanical drive (MAN, 2015). The system can be used up to a velocity of 30 km/h where it automatically disengages itself; if the velocity is below 22 km/h the system automatically reengaged if turned on by the driver.

The hydraulic pump is mounted on a PTO located on the gear box, leading to that a gear has to be selected in order for the system to be engaged. Also, by having the pump mounted to the gear box the gearing ratio to the pump is not limited to one ratio as in the case when mounted to the REPTO (Rear engine power take-off).

The claimed advantages by MAN's Hydrodrive solution are:

- Lower total fuel consumption.
- Reduced weight of 400 kg compared to M.FWD.
- The vehicle does not has to be extra high, i.e. CHH-High
- Increased drivability and road safety
- Hydraulic brake, meaning the system can be used as a brake, sparing the regular brakes (MAN, n.d.).
- Maximum motor torque up to 14 580 Nm for the entire system
- Fitted with a differential lock to further increase hydrostatic drive further on loose ground (MAN, n.d.).

The only found picture of the routing which MAN used at least during 2006 can be seen in picture of MAN routing in, Appendix A.

3.3.1.2 Renault - Optitrack

Second company which launched C.H.FWD as an option was Renault Trucks in 2010, Renault Trucks later announced a new version of the "Optitrack" in 2014.

The pump in the Optitrack solution receives its power from the engine via a prop shaft which is connected to the engines PTO.

The two solutions are relatively similar where one of the new features of the 2014 version is an added hydraulic retarder. The reduced weight compared to having M.FWD is 400-490 kg and the system can be used up to velocities of 25-30 km/h with a maximum produced power of each wheel is 41 kW (Renault Trucks, 2010), (Trucks, n.d.) & (Renault Trucks, n.d.).

Found pictures of the routing of Renaults Optitrack solution can be seen in Appendix B.

3.3.1.3 Mercedes Benz – Hydraulic Auxiliary Drive
The third competitive truck manufacturer whom today offers complementary hydraulic front wheel drive to the market is Mercedes Benz. The system called Hydraulic Auxiliary Drive (HAD) was launched in the first half of 2015. The pump is mounted directly to the PTO of the flywheel housing, which enables the possibility of engaging the C.H.FWD without engaging a gear in the gearbox and also reduces the total weight of the truck.

System specifications specified by Mercedes Benz:

- Motor size - 934 cc, per motor
- Maximum torque output and effect, per motor – 6250 Nm and 40 kW
- Maximum pump effect – 112 kW
- Maximum system pressure – 450 bar
- Maximum volumetric flow – 350 l/min
- Maximum feeder pump pressure – 30 bar
- Top velocity when engaged – 25 km/h, automatically disengaged when exceeded and reengaged if dropped below while the system is turned on
- Weight savings – 350-500 kg

It is estimated by Mercedes that, between 15-25 km/h the system will only be engaged approximately up to five minutes at a time and below 15 km/h more continuously. The entire system is cooled by a cooling unit capable to produce a cooling effect of 20 kW.

The system, as the two mentioned solutions above, can be used as a brake which reduces wear on the brakes (Mercedes Benz, 2015).

Found pictures with describing text regarding routing of Mercedes C.H.FWD solution can be seen in Appendix C.

3.3.1.4 Ginaf
The Dutch truck manufacturer Ginaf has announced their hydraulic drive system, HydroAxle+ for trucks, but dates and availability has been hard to establish. The system produces a maximum power of, although questionably high compared to all other solutions, 200 kW and can be enabled up to velocities 20-26 km/h, while reducing weight by 500 kg (Ginaf, n.d.).

No pictures of the routing from the wheel end to chassis could be found regarding the HydroAxle+ solution.

3.3.2 Aftermarket solution
The following section presents the aftermarket solutions found regarding C.H.D for trucks and trailers.

3.3.2.1 Terra Drive Systems
Terra drive systems are located in the USA, specialised in steerable hydraulics on medium- and heavy trucks. Terra drives systems offers an aftermarket solution called EZ Trac which has the following specifications:

- Maximum effect – 110 hp or 82 kW
- Highest velocity which system can be engaged – 32 km/h
- Weight savings compared to conventional M.FWD – 362-453 kg
- Can be mounted onto several different types of trucks and models (EZTrac, n.d.), (EZTrac, n.d.).
- Pump and motor manufacturer – Poclain Hydraulics
- Maximum pressure of system – 414 bar (Startrucks, n.d.).

No pictures of the routing from the wheel end to chassis could be found.

3.3.2.2 Terberg Techniek

Terberg Techniek offers the possibility to fit a standard Volvo front axle with the X-Track solution, enabling a top velocity of 20 km/h with the hydraulic drive engaged and a weight saving of 600 kg compared conventional M.FWD (Terberg Techniek, n.d.). The earliest announcement found regarding the X-Track was in 2013 and is here viewed as the year it was launched (Terberg Techniek, 2013).

No pictures of the routing from the wheel end to chassis could be found.

3.3.2.3 SAF-Holland

SAF-Holland is a company which manufactures systems and components for both trucks and trailers, but also busses and recreational vehicles.

In 2011 SAF Holland released a C.H.D system called SAF Pendulum ZMP9-3015 which is fitted onto a swivel axle specified for 9 tonnes (SAF-Holland, 2011). System specifications:

- Weight of complete solution (axles and other parts included) – 440 kg
- Used motors – MFE 08, provided by Poclain hydraulics (SAF-Holland, n.d.)
- Produced maximum power of both motors:
    - 164 kW specified by SAF Holland (SAF-Holland, 2014)
    - 82 kW specified by Poclain Hydraulics (Poclain Hydraulics, 2012)

Illustrative pictures of the SAF Pendulum ZMP9-3015 can be seen in Figure 2.

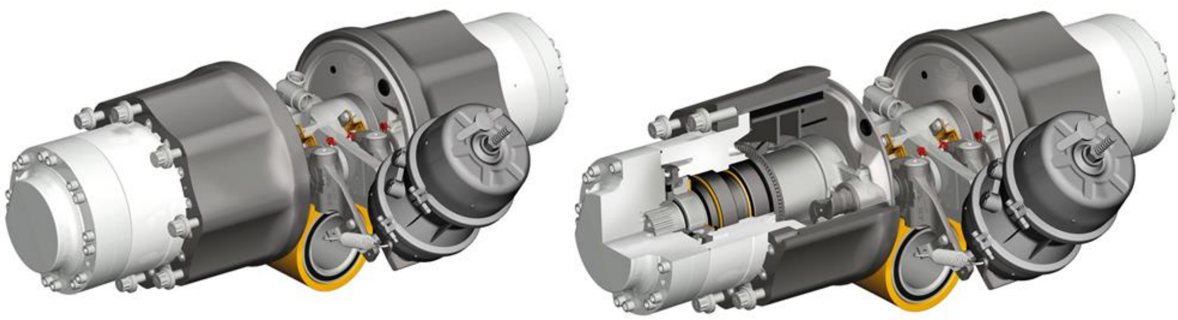

Figure 2 SAF Pendulum ZMP9-3015

SAF-Holland later announced that a new C.H.D to trailers was set to be released in 2015. The new solution is compatible with all rigid, disk-braked standard axles of the type BI9-19/22" using steel rims. Specifications of the new system:

- Weight of the complete solution – 200 kg
- Possible to have engaged up to velocities of 5 km/h where the system automatically is disengaged if not switched off.

Information regarding name of the solution, maximum torque or illustrating pictures has not been found (SAF-Holland, 2014).

During an exhibition in Germany during the spring of 2016 yet another solution was revealed, SAF Intra Trak, however no information regarding this solution can be found in SAF-Holland's web page. In Figure 3 is a picture of the solution provided, taken from the brochure handed out at the exhibition.

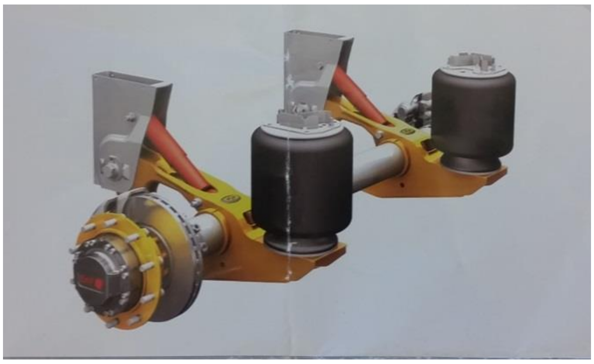

Figure 3 SAF Intra Trak

3.4 The hydraulic system in general and components for current front wheel drive

Hydraulic systems are used in several different areas and applications, where one area is to use the hydraulics as a transmission which drives wheel motors located in the wheel hubs. The two main benefits which Volvo obtains from implementing this system today are; weight reduction of a magnitude of at least 400 kg (Poclain Hydraulics, n.d.). The second advantage is that it is easier to package into the truck compared to a mechanical solution because it uses smaller parts and the hoses, which transfer the force, are flexible and can be bent.

A hydraulic system such as the one Volvo uses is a closed loop system and requires certain parts in order to work:

1. A pump is needed which is being driven by a rotational force, in Volvos case this pump is now mounted directly on the PTO where needed rotational force is transferred from the engine.
2. Motors, Volvo uses radial piston motors where the pistons push on a ring with highs and lows, causing a rotational motion. In Figure 4 below this is illustrated, where the enlarged part on the right shows how piston A pushes towards the ring due to the increased pressure inside the cylinder and forces the outer ring rotate. At the same

time piston B is contracted by reducing the pressure inside the cylinder in order to reduce losses.

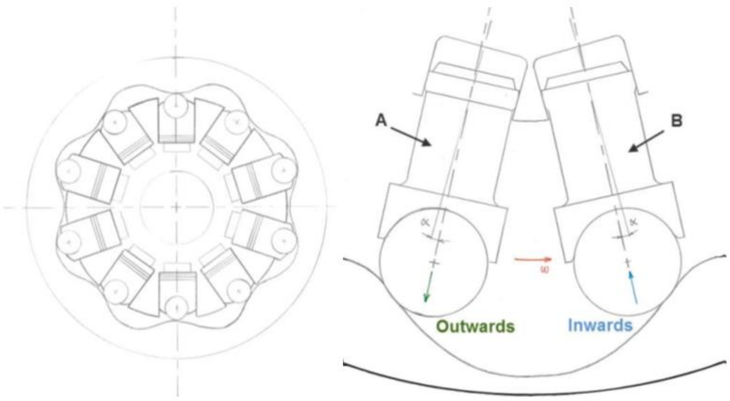

Figure 4 Hydraulic motor, source: (Karlsson & Persson, 2012)

3. Valve block, the valve block can be viewed as the heart of the system where all flows goes through, the purpose of the valve block is to distribute the flow over the system.
4. ECU, this is where the software and logic for the system is stored, all signals comes from the ECU, if the valve block can be viewed as the heart the ECU is the brain.
5. Hydraulic tank, since oil expands when it increases in heat a tank is of need. Also the hydraulic oil needs to aerate in order to avoid cavitation, where the conventional solution is to use a large reservoir where the oil can naturally de-aerate by slowly circulating.
6. In addition to these five components hydraulic hoses, pipes and fittings are also needed in order to complete the system. Further some sort of cooling might be of need if the temperatures risks of exceeding stated maximum temperature, see Table 4. Also, in order to obtain a clean particle free oil flow a system as such has to have at least one filter (König & Robertsson, 2015).

3.5 Hydraulic hoses/pipes and nipples

This thesis does not consider the final choice of hoses, pipes and nipples in the assembly, that is to covered by an expert in the area, but consideration regarding sizes and how the routing can be done will be performed. Therefore some pre knowledge regarding hoses, pipes and nipples is of need.

3.5.1 Hoses and pipes

A hydraulic hose consist of three layers, the inner tube which the flowing medium will be in contact with, the middle which is reinforcement and an outer cover which will be in contact with the ambient environment, see Figure 5.

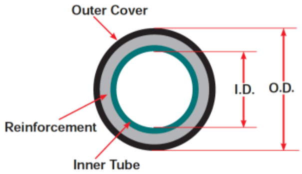

Figure 5 Cross section hydraulic hose (Parker Hannifin, n.d.)

When choosing a hose five different areas need to be addressed, which Parker Hannifin Corp defines as STAMP:

1. Size
2. Temperature
3. Application
4. Media
5. Pressure

1) The inner diameter of a hose of pipe must be of accurate sizing in order to obtain desired flow properties. Having to low flow results in reduced system performance, while having a to high flow velocity results in higher pressure drops, system damage and leakage.

2) There are two temperatures to consider when dimensioning the hose(s), the ambient temperature, meaning the outside temperature which the hose will be subjected to and the media temperature, referring to the temperature inside the hose. If not chosen correctly the hose lifetime can be severely reduced.

3) Depending on application different hoses/pipes are suitable, it is therefore important what kind of environment the equipment will be used in, if the routing will be confined, subjected to abrasion, etc.

4) Different materials are compatible with different fluids, which can result in reduced service life time if chosen incorrectly.

5) It is out of a safety and life time perspective the pressure should be considered when choosing and dimensioning the hydraulic hoses. A hose's/pipe's maximum pressure has to be equal or exceed the maximum system pressure, important to remember is that pikes which are larger than the working pressure can occur in hydraulic systems. Parker Hannifin Corp uses a safety factor of four as standard, meaning that the hoses can handle at least four times stated pressure (Parker Hannifin, n.d.).

When designing the routing layout there are a few factors which are important to consider in order to not have reduced lifetime of the hose:

- Enough length to enable possibility of having slack along the path.
- Remove possibilities of hose to rub against other parts, abrasive wear can otherwise occur.
- Not exceed set minimum radius by the manufacturer.

- A hose is rotational stiff, i.e. should only be subjected to bending motion and not twisting, a hose can/should therefore only be bent in one plane at a time (Parker Hannifin, n.d.).
- A hose should have at least two times the outer diameter as free length before forced into a bend (Andersson, 2016).

3.5.2 Nipples and fittings

There are a large number of different types of nipples and fittings; there are different standards on the threads, differently chamfered edges and different shapes and overall solutions. The fittings are categorised into two different groups depending on how the sealing is handled, either metal against metal or metal against an elastomeric ring, a so called O-ring. The metal sealing is done either with the threads that keep the male and female connected, a solution which has high risk of leaking when applying high pressure. Or the metal to metal sealing can be done with a flare edge, where a coned tube end is pressed towards a chamfered nipple edge (Pneumatics, 2012), this solution is widely used and can be implemented at very high pressures. However since the sealing area is metal surface against metal surface, high surface smoothness is required, making the sealing sensitive towards damage and small scratches can remove the sealing capability.

There are some “special” types of fittings, quick couplings and swivels are two types. Quick couplings have the advantage of being very easy to attach and detach, but are not very durable in extreme conditions. Swivel couplings are couplings which allow rotation and are often used in excavators and other applications where the hydraulic system has to perform rotational motion. If a swivel coupling is exposed to dirt, the lifetime will be reduced since the dirt will wear on the rotating parts.

When a hose is to be attached in both ends, it is common to use female swivel fittings or fittings with flanges and clamps. Using rigid male fittings in one end and female in the other can be done, but using rigid male fittings in both ends can be troublesome. The advantages of using female swivel fittings or fittings with a flange and a clamp is that the hose can be rotated as desired until it is completely tightened, something that cannot be performed if a rigid male fitting is used (Ahlberg, 2016).

Parker has defined a two-step method with several sub steps to systematically select correct fitting, the substeps will not be further reviewed in this thesis:

1. Clarify all design criteria for the system
2. Determine the “best solution” given the defined design criteria (Parker, 2011)

3.6 Hydraulic schematics

The schematics of a hydraulic system are the hydraulic drawings of the different parts in the system and consist of how these parts are connected to each other. In order to increase understanding among engineers a common way of defining the system has been developed by the international standard organization (ISO).

The area of interest regarding schematics in this report is the valve block, and in order to reduce the huge number of different types of symbols needed to be defined; only found symbols which might be of relevance for the control block will here be discussed.

Firstly the lines which the oil flows in are indicated by different colours depending of the type of flow and are drawn differently depending on if it is a pipe or flexible hose. Figure 6 illustrates the different colours and configurations different lines have in schematics for hydraulics. Note here that the pilot lines and drain lines have different length on the dashed lines, the pilot line have longer sections between every void compared to drain lines.

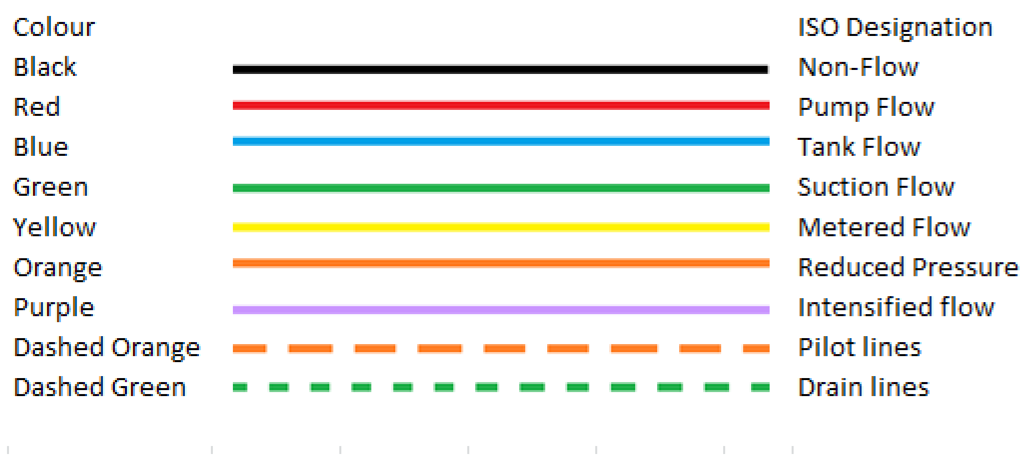

Figure 6 Colour coding

Whether a line represents a hose or pipe can be seen by if it is drawn as a straight line or with a curve, if there is a curvature along the path it is a hose, either the entire section is a hose or a part see Figure 7. Whether two lines are crossing each other or are connected with each other is indicated with a dot, if there is no dot the lines are crossing each other, if there is a dot the lines are connected see Figure 8. Indications of the direction of flow is done with a black triangle with one of the tips pointing towards the flow direction, see Figure 9 (Hydraulics & Pneumatics, 2006).

Figure 7 Hose, (Hydraulics & Pneumatics, 2006)

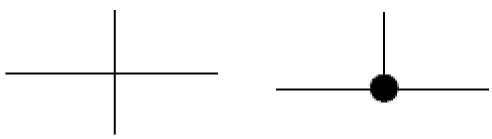

Figure 8 Crossed lines and connected lines, (Hydraulics & Pneumatics, 2006)

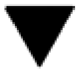

Figure 9 Direction of flow, (Hydraulics & Pneumatics, 2006)

The direction of the flow in the system is controlled by either the pump or so called valves. A valve can either be controlled by a solenoid, pneumatic or by hydraulic pressure. If a valve is pilot controlled, it means that the valve will react upon a change made somewhere else in the system (Arvidsson, 2016). It is common to attach a spring with desired spring constant in order to keep the valve in a certain position once the solenoid or pilot valve is not engaged. For symbols of solenoid, hydraulic controlled valve (pilot valve) and spring see Figure 10, Figure 11 and Figure 12.

Figure 10 Spring, (Pneumatics, 2012)

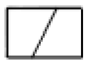

Figure 11 Solenoid, (Hydraulics & Pneumatics, 2006)

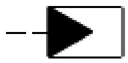

Figure 12 Pilot, (Hydraulics & Pneumatics, 2006)

A valve can perform a series of different effects on the flow of the system, change the direction, act as a stop, only allows passage in one direction, etc. Example of how types of flow controllers can be seen in Figure 13, where seen from the left are; cross flow, flow stop, flow in two directions.

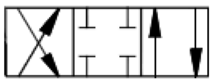

Figure 13 Flow valves, (Hydraulics & Pneumatics, 2006)

If a damping effect is desired to be obtained in the system a restriction can be inserted, either as non-variable or as variable, see Figure 14 which illustrates these two types.

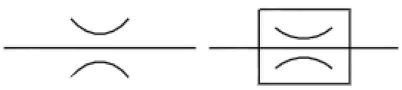

Figure 14 Non-variable & Variable, (Hydraulics & Pneumatics, 2006)

3.7 Calculations

In the following sections required formulas and theory regarding ratios for transmissions, hydraulics, flow and losses are presented.

3.7.1 Ratios for transmission

Almost all vehicles today have some sort of transmission mounted in between the engine and the driven wheels. The purpose of the transmission is to change the angular input speed and torque from the engine to the driven wheels, depending if low speed and high torque or higher speed and lower torque is wanted. There are several different configurations of transmissions, e.g. mechanical, hydraulic, electric, etc. The mechanical transmission is the most common type and can be found in e.g. normal every day cars; these can be either manual or automatic and have several different configurations of how to solve the gearing. A stepped manual transmission is a transmission where each gear is individually selected making the gearing occur in steps. This compared to a continuously variable transmission which can continuously change the gearing without any steps. An automated transmission removes the need of gear change from the driver and instead utilizes a computer which controls which gear is to be selected (Hillier & Coobes, 2004).

When calculating the gearing of a manual transmission something called ratio is used, ratio tells how the angular velocity and torque inserted relates to the output. The total ratio for one gear is as following if in- and output are located on the shame shaft:

$$i_g = \frac{\omega_{in}}{\omega_{out}} = \frac{T_{out}}{T_{in}} = i_{prim} * i_{sec} \tag{1}$$

Where i_{prim} is the ratio step over the primary gear pair and i_{sec} the ratio over the secondary gear pair which will determine which gear is selected.

These calculations only consider the ratio of the transmission and not the gearing in the differential, i_f, and from the wheel radius. The gearing in the differential and the wheels are calculated in the same way as for the gear box, equation (1), dividing the rotational input speed with the rotational output speed. The velocity of the vehicle can thereby be calculated with following formula, equation (2) (Hedman, 2016):

$$v_{vehicle} = \omega_{engine} * \frac{r_{wheels}}{(ig * if)} \tag{2}$$

3.7.2 Hydraulics

Calculating a hydraulic transmission is much similar to calculating a mechanical transmission, where the real difference is; instead of angular speed and torque, flow and pressure is calculated.

There are two components which are performing work in the system, the pump and the hydraulic motor. The following equations can be used for calculations of hydraulic pumps with concern to efficiency, see equations (3) – (7):

$$Q_p = \frac{Dp * n_p * \eta_{volp}}{1000} \tag{3}$$

Where Q_p is flow [l/min] for the pump, Dm is the geometric displacement of the pump [cm^3/rev], n_p the rotational speed of the pump shaft [rpm] and η_{volp} the volumetric efficiency.

$$P = \frac{p * Q_p}{600 * \eta_{totp}} \tag{4}$$

Where P is the power of the pump [kW], p the working pressure [bar] and η_{totp} the total efficiency of the pump which is calculated with following equation:

$$\eta_{totp} = \eta_{volp} * \eta_{mhp} \tag{5}$$

Where η_{mhp} is the hydro mechanical efficiency

The rotational ratio for a hydraulic pump can be calculated with the following formula, equation (6):

$$i_{rot} = \frac{n_p}{n_{p,max}} \tag{6}$$

Where n_p is the actual rotational speed of the pump in rpm and $n_{p,max}$ the maximum achievable rotational speed.

The torque which the shaft of the pump will be subjected to and transfer to PTO can be calculated via the following formula, equation (7):

$$T = \frac{D_p * \Delta p}{20 * \pi * \eta_{mhp}} \tag{7}$$

The following equations apply for the hydraulic motor, but are similar to those for the pump, see equations (8) – (11):

$$Q_m = \frac{D_m * n_m}{1000 * \eta_{volm}} \tag{8}$$

Where Q_m is the flow to each hydraulic motor [l/min], D_m is the geometric displacement [cm^3/rev], n_m the rotational speed [rpm], η_{volm} the volumetric efficiency.

$$n_m = \frac{Q_m * \eta_{volm} * 1000}{D_m} \tag{9}$$

$$T_m = \frac{\Delta p * D_m * \eta_{mhm}}{20 * \pi} \tag{10}$$

Where T_m is the torque provided by the hydraulic motor [Nm], η_{mhm} the mechanical hydraulic efficiency and Δp the differential pressure which is calculated as following:

$$\Delta p = p - p_{out} \tag{11}$$

Where p is the pressure out from the hydraulic motors [bar], (Arvidsson, 2016).

The torque needed to be produced in order to move a vehicle can be calculated by following formula, equation (12).

$$T_x = N * \mu * r \tag{12}$$

Where T_x is the vehicles driving force in driven direction [Nm], N is the normal force generated by the weight acting on the driven wheels [N], μ is the friction coefficient (Pettersson, 2012).

The maximum speed which the hydraulic system can be used for is also determined by its ability to keep up with the mechanically driven axle(s). This can be calculated via equation (13), where if the relation is higher than zero, > 0, the hydraulic output speed is higher than the mechanical and it can be used for that particular gear. But if the result is less than zero, < 0, the hydraulic motors cannot keep up with the mechanical axle speed and would act as a brake. Using a safety factor of at least five percent is recommended (Pettersson, 2016).

$$Relation = \left(\frac{D_p * \eta_{volp} * i_{PTO} * i_{tot}}{2 * Dm * \eta_{volm}} - 1\right) * 100 \tag{13}$$

The maximum angle which a vehicle can start in if the rolling resistance is neglected is determined by the driving force, the total mass to be moved and the experienced gravity (=9.81), see equation (14).

$$\alpha = \sin^{-1} \frac{F}{M_{tot} * g} \tag{14}$$

The limiting friction can be determined by following formula, equation (15):

$$\mu = \frac{F}{M_d * g * \cos\beta} \tag{15}$$

Where M_d is the mass on the driven axle and β is determined by 90-α (Pettersson, 2012).

3.7.3 Inner diameter and wall thickness

It is recommended that the flow velocity in pressure lines is kept below 8 m/s, higher velocities can result in reduced lifetime of the hoses/pipes leading to leaks and failure. Recommended maximum flow velocity for pressure lines is 5-6 m/s, return lines 3 m/s and suction lines 1 m/s (Parker Hannifin, 2008).

Reduced inner diameter with maintained volumetric flow increases the flow velocity, where increased pressure losses in the system can be expected (Andersson, 2016). Losses due to dimensioning and specification of the system are further discussed in section 3.7.4.

The flow velocity in a hose is directly connected to the hoses inner diameter and the expected volumetric flow. The hose inner diameter can also be calculated if the volumetric flow and velocity are known. The volumetric flow is here seen as the maximum flow of the system which is produced by the pump, see equation (3), for the velocity following formula is used:

$$c = \frac{Q_p * 4}{d^2 * \pi} \tag{16}$$

Where c is measured in [m/s] and d is the inner diameter of the hose in [m].

Further, numerous of simplified graphs have been found in order to easily determine analyse the system and find the systems volumetric flow rate, hose diameter or flow velocity. None of these graphs have in this report been used and is only mentioned with the purpose to notify the interested reader.

Using to small wall thickness of the pipes can result in failure; due to the high pressure in the system. Previously used thickness has been 2.5 mm which will in this project be kept unchanged (Andersson, 2016).

3.7.4 Losses due to routing

In a hydraulic system the hoses/pipes and fittings always have some losses, which results in reduced total efficiency. The losses in a pipe and individual coupling can be calculated with the formulas:

$$\Delta p_p = \frac{\lambda * L * \rho(T) * c^2}{d * 2} \tag{17}$$

$$\Delta p_c = \frac{\zeta * \rho(T) * c^2}{2} \tag{18}$$

Where Δp_p is the pressure loss in the pipes and Δp_c the pressure loss in the couplings, both measured in [Pa]. λ is the pipe friction factor, L the pipe length [m], $\rho(T)$ the density of the medium depending on temperature [kg/m^3], d the inner diameter [m], and ζ the individual pressure loss coefficient of a coupling.

The individual pressure loss coefficient for a coupling, ζ, is provided by the hydraulics supplier and is higher at very low volumetric flows. Depending on the type of coupling different level of losses are obtained. A straight fitting where the inlet has the same area as the outlet a coefficient interval of $0{,}01 \le \zeta \le 0{,}05$ can be expected. For a straight coupling where the outlet area is larger than the inlet area, equation (19) can be used:

$$\zeta = \left(\frac{A_{out}}{Ain} - 1\right)^2 \tag{19}$$

Where A_{in} is the inlet area and A_{out} is the outlet area of the coupling.

If the inlet area instead is larger than the outlet area, the following values can be expected, see Table 1.

Table 1 Straight large too small

Performance data				
A2/A1	0,8	0,6	0,4	0,2
ζ	0,15	0,25	0,35	0,42

As can be expected, when applying a bend within the system higher losses are also a resulting consequence. If a 90 degree elbow fitting is used, a loss coefficient of 1 can be expected, if instead a bent pipe is used the losses can be greatly reduced, see Table 2.

Table 2 Losses bent pipe

Performance data			
Bend radius / inside diameter	2,0	4,0	≥6,0
ζ	0,21	0,14	0,11

Banjo fittings are more compact than the regular nipples, but results in lower efficiency and an expected loss coefficient of 3 to 9. Banjo fittings are often used where space is limited and little interest to pressure drops is of need.

In a tee/cross fitting or in a manifold, flow can either be divided into two flows at a branching or two flows combined into one at a junction. Therefore in tee/cross crossings and manifolds there will be two locations which will affect the pressure drop and be different depending of flow direction. The different loss coefficients have been summarized into Table 3 where $\dot{V}$ is the total flow and the sum of $\dot{V}_A$ and $\dot{V}_B$ is the total flow.

Table 3 Losses tee and manifolds

Performance data				
Flow division	Pressure loss coefficients at branch		Pressure loss coefficients at junction	
$\dot{V}b/\dot{V}$	ζa	ζb	ζa	ζb
0,6	0,07	0,95	0,4	0,47
0,8	0,2	1,1	0,5	0,73
1	0,35	1,3	0,6	0,92

The valves within the system are parts which offer a relative large loss coefficient compared to the other couplings discussed. The pressure loss coefficient varies from four to 5.5 depending on the type (Parker, 2011).

3.8 Friction and rolling resistance

Without friction a vehicle cannot move if located on a horizontal surface without any external forces acting on the vehicle, e.g. wind, see equation (12). The higher the friction the higher force/torque can be applied to propel the vehicle forward without slipping. Slipping will occur once the produced force/torque is higher than the available friction force of the surface.

Rolling resistance is due to, among other, the resistance in the ground which prevents movement of the vehicle. The rolling resistance depends on the type of surface but also the inclination. The looser ground the higher rolling resistance since the vehicles wheels will sink deeper into the ground and have more material to overcome. Also the higher degree of inclination the higher rolling resistance, this is due to equation (15).

In order to avoid a three dimensional problem with friction, inclination and rolling resistance, the rolling resistance can be added together with the inclination if GTA (global transport application) parameters are being used, meaning if there is x % inclination and y % rolling resistance, the total force the truck has to overcome in order to move is x + y %, example of this can be seen in chapter 5.2 (Pettersson, 2016).

Based on previous testing done by Volvo, data for different surfaces friction and rolling resistance has been gathered and defined into four groups ranging from $0,3 \leq \mu \leq 0,8$. As can be seen in Appendix D, ice offers amongst other surfaces the lowest friction and dry asphalt and concrete offers the highest. Regarding the rolling resistance, it ranges from 0,4 % to 40 %, where Volvo has chosen to categorise this interval into five groups, from ice, asphalt, etc. to loose clay, dry and loose sand, etc. see Appendix E. These two appendices are a part of the so called GTA parameters.

4.0 Requirements

In the following section the final requirements which have been developed throughout the project is presented. To the right is a check column which indicates whether requirement or request have in the report been established as fulfilled (green + yes), needs more information (yellow + maybe) or not met (red + no).

4.1 All requirements

As can be seen in Table 4 seven requirements needs more information before these can be set as passed or not passed. Six of them are internal questions for the supplier of the hydraulic system and one is a Volvo issue. The one which concerns Volvo is the free distance from tires, which needs physical tests to further evaluate.

Calculated maximum theoretical torque for the pump would be 596 Nm which is relatively little above the maximum value, but above none the less and therefore set as not passed. The request of starting in 40 % rolling resistance has not either been fulfilled, but this seem difficult to achieve even with M.FWD.

Table 4 Requirements

Criteria	Requirement/Request	Target value	Comments	Check
Maximum working/peake pressure	Requirement	420 / 450 bar	Maximum working pressure is 425 bar with occasional peaks of 450 bar	Yes
Feeder pressure	Requirement	25 bar		Yes
Maximum weight on pusher axle	Requirement	9 t	Maximum specified wegiht of the pusher axle is 9 tonnes	Yes
Maximum weight on tag axle	Requirement	10 t	Maximum specified wegiht of the tag axle is 7 tonnes	Yes
Steerable / rigid pusher/tag axle	Requirement	Steerable	Both the pusher axle and the tag axle should be steerable	Yes
Used engine type	Requirement	16-liters	Maximum torque and rev. Stop differs between engine sizes	Yes
Used transmission	Requirement	ATO3512D	Different transmissions have different gearing	Yes
Used tyres on all wheels	Requirement	385/65R22.5	Different tyres have different diameter and deformation behaviour	Yes
Pump volume	Requirement	90 cc in 71 cc chassis	The pump should be a 90 cc pump in 71 cc chassis and should be fitted with through drive	Yes
Should not be needed to develop a new solution for the routing of the front wheels	Requirement	Yes		Yes
Minimum achiavable startability	Requirement	12 % on	Should be able to start in 12 % slopange on asphalt	Yes
Parts from the existing solution should be kept as they are if possible	Request	Yes	Parts from previous C.H.FWD system shold be used as carry overs to this solution if possible	Yes
Used high pressure hoses	Requirement	797 TC-10	Use the same hoses as are fitted onto FH-1825	Yes
Minimum bending radius high pressure hoses		R = 100 mm	Minimum allowed bending radius for the high pressure hoses 797 TC - 10	Yes
Used hoses for leakage and feederline to motors	Requirement	462 ST	Use the same hoses as are fitted onto FH-1825	Yes
Minimum bending radius leake/feeder pressure hoses	Requirement	R = 130 mm	Minimum allowed bending radius for the leake and feeder line hoses 462 ST	Yes
Surrounding max temperature	Requirement	+40 Celsius		Yes
Surrounding min temperature	Requirement	-40 Celsius		Yes
Lowest acceptable natural frequency	Requirement	≥ 30 Hz		Yes
Free distance with snow chains	Requirement	60 mm (50 mm +10 mm)	50 mm free distance to clear the snow chains and 10 mm from wear of tires	Maybe
Rear axle gearing	Requirement	3.61	rear axle gearing = ifinal	Yes
Switch between C.H.FWD C.H.4WD and no C.H.D	Requirement	Yes	Posibility to enable the hydraulic drive on the front wheels and on the front wheels and pusher/tag axle or disengaging it fully	Yes
Engage C.H.D on only pusher or tag axle	Request	Yes	Posibility to enable the hydraulic drive on only the pusher or tag axle	Yes
Max rotational speed when active	Requirement	>= 50 km/h = 248 RPM	Maximum rotation speed using a the wheel radius 0.4951 at 9 t	Maybe
Active distance	Requirement	140' km	Manage 140' km of usage which is 10 % of total driving distance 1400' km	Maybe
Active time	Requirement	7630.0 hours	Time in hours the system is expected to be activated	Maybe
Freewheeling distance	Requirement	1260' km	Manage 1260' km of passive rotation, 90 % of total driving distance 1400' km	Maybe
Maximum rotational speed when motors are disabled	Requirement	90 km/h	The motors should manage to passively rotate up to a linearly speed of 90 km/h	Maybe
Maximum total torque	Requirement	≥ 8700 Nm	Maximum total produced torque of all motors should exceed 8700 Nm	Yes
Torque on each motor	Requirement	2000-2500 Nm	Each motor should be dimensioned for 2000-2500 Nm	Yes
Compatible with the old spindle	Requirement	Yes	Same interface between motor and spindle as before	Yes
Compatible with the same type of wheels	Requirement	Yes	Same interference between wheels and motor	Yes
Number of expected activiations	Requirement	500' Times		Maybe
PTO ratio	Requirement	1.26	PTO ratio = ipto	Yes
Rolling resistance	Request	40%	Manage a rolling resistance of 40 %	No
Air suspension	Requirement	Yes	The trucks should have air suspension	Yes
Max torque on pump	Requirement	572 Nm	Maximum allowed torque for continous use at a differential pressure of 400 bar	No

5.0 Calculations and dimensioning of motors

In the following sections necessary data needed to perform calculations regarding velocities, torque and startability for the existing test truck FH-1825 and the upcoming four motor solutions are presented with tables and graphs.

5.1 Necessary data for calculations of hydraulics

It is of desire to have a hydraulic system which could provide higher top velocity than today's solution at the same time as the torque is maintained or increased. This could of course be solved by increasing the flow to the motors or making the motors larger, see equation (8) & (10). But as can be seen in the list of requirements the pump model and size has to remain the same, thus the maximum flow to the motors cannot be increased by increasing the pump size. The flow can also be increased by increasing the rotational speed of the pump, which either can be done via increasing the rotational speed of the engine or the gear ratio between engine and PTO. The engine is limited to maximum 2000 rpm and can therefore not be increased; in order to change the PTO ratio, entirely new flywheel housing would be needed, or some sort of gearing could be used.

The optimal PTO ratio can be calculated using equation (6), where optimal ratio is the quota of maximum allowed rotational speed for the pump and the maximum possible rotational speed of the engine see Appendix F. As can be seen a PTO ratio of 1,525 would be optimal for this set up, compared to today's ratio on FH-1825 which is 1,26. However changing the ratio value of the PTO is out of the scope for this thesis and the ratio 1,26 will further be viewed as a fixed constant.

The way which higher speed could be obtained while the torque is maintained or increased, studied in this thesis work, is decreasing the displacement of each motor and increasing the total number of motors fitted onto the truck, see equations (9) & (10).

5.1.2 The previous test truck FH-1825

As a reference calculations of the precious C.H.FWD solution on test truck FH-1825 has been conducted where relevant data has been gathered and specified in the sections below.

The two hydraulics motors have an individual displacement of 780 cc which is being provided high pressure oil by a 90 cc pump where the 90 cc pump is continuously being fed by a 19.6 cc feeder pump. The hydraulic system has been specified by the hydraulics supplier to a working pressure of 425 bar, with possible spikes of 450 bar and a differential pressure of 400 bar. The two motors are located in the front wheels of a 3 axle truck where two axles are driven, and the front wheels are viewed as non-driven, i.e. 6x4. Maximum weight which the front axle can be subjected to is nine tonnes and the rear bogy is specified to 26 tonnes, however according to legislations the bogie cannot exceed a weight of 19 tonnes for BK1 and the entire truck should not exceed 26 tonnes (Transportstyrelsen, 2014). The weight distribution of FH-1825 has been specified as following; also see Appendix G:

- Weight on front axle
    - Service weight - 6000 kg
    - Max gross weight - 7085 kg

- Weight on rear axles
    - Service/unladen weight – 6000 kg
    - Max gross weight – 18570 kg
- Total weight of equipage
    - Service/unladen weight – 28650 (trailers included)
    - Max gross weight – 73656 kg

The tyres of a vehicle are compressed according to the weight that is to be supported; increased weight leads to increased deformation, where different tyres deform differently. The non-driven wheels on FH-1825 are of models 385/65R22.5, which are compressed according to Appendix H.

The efficiency of the motors and pump are as stated in 3.0 measured in volumetric efficiency and mechanical-hydraulic efficiency, where the sum of the two is the total efficiency. The efficiency has been based on data provided by the hydraulics supplier which is confidential material at a differential pressure of 400 bar and a rotational ratio of approximately 0,8. The ratio 0,8 has been calculated using equation (6), see Appendix I.

The efficiency of the hydraulic motors are measured in the same way as the pump, both volumetric efficiency and mechanical-hydraulic efficiency. The volumetric efficiency has been approximated from the efficiency curve presented from confindential material. The line of interest is the black line which is representing the efficiency for a differential pressure of 400 bar. For used calculations the curve was approximated as a linearly increasing curve until 25 rpm, and then an average efficiency of 88 percentages was assumed for all remaining rotational speeds, this simplification was verified to be an accurate enough estimation by Jakob Arvidsson.

The truck is powered by Volvos 16-liters diesel engine which has a maximum rotational speed of 2000 rpm and produces a maximum torque of 3550 Nm (Pettersson, 2016). The transmission which transfers this power to the rear wheels is a twelve speed automated gear box, named ATO3512D. The different ratios for each gear within the gearbox can be viewed in Appendix J (Volvo Trucks, 2011).

The final ratio, which is obtained through the differential, is set to 3,61 and the total efficiency of the powertrain has been approximated to 93 % (Hedman, 2016).

A timber truck such as FH-1825 is during its lifetime subjected to both good and bad road conditions, such as dry asphalt but also steep hills and muddy and icy roads. A timber truck can encounter a rolling resistance up to 40 % if encountered with clay, sand or similar, occasions which also provides low friction, as low as less than 0,3. Further a truck should be able to start in a 12 % slope on dry asphalt (Pettersson, 2016).

5.1.3 Upcoming four motor solutions

By increasing the number of hydraulic motors, as the maximum displacement of each motor is proportionally decreased, the total system should in theory produce the same velocities and torques. That is if the rest of the system is maintained the same, see equations (9) & (10). If only half of the motors are engaged the top velocity should be doubled at the same time as the total produced torque should be halved.

The speed relation between the hydraulic drive and the mechanical drive also needs to be considered. The hydraulic system is a complementary system to the mechanical drive, meaning it will aid the mechanical drive and is not intended to be engaged individually. This leads to that the highest possible velocity the motors can be engaged, will also be dependent on the capability to keep up with the mechanical drive, see equation (13)

As stated in the delimitations the only axle configurations that will be of concern is RAPDD-GR, 8x4, and RADT-GR, 6x2. The performed calculations are based on the same specifications as the FH-1825, with the exceptions; hydraulic motor configuration, weight and axle arrangement.

For RAPDD-GR following weight distribution has been used:

- Weight on front axle
  - Service/unladen weight – 6962 kg
  - Max gross weight - 8832 kg
- Weight on pusher axle
  - Service/unladen weight – 0 kg
  - Max gross weight - 6720 kg
- Weight on driven axles
  - Service/unladen weight – 6075 kg
  - Max gross weight - 17280 kg
- Total weight on equipage, trailers included
  - Service/unladen weight – 24 400 kg
  - Max gross weight - 90000kg

When driving unloaded, service/unladen weight with trailers, pusher and the last driven wheel will be lifted in order to reduce rolling resistance and increase startability. For illustrating picture see Appendix K.

For the RADT-GR, the total weight is reduced due to the wheel configurations and is set as following:

- Weight on front axle
  - Service/unladen weight - 5347
  - Max gross weight – 9000 kg
- Weight on tag axle
  - Service/unladen weight - 0 kg
  - Max gross weight – 7475 kg
- Weight on driven axles
  - Service/unladen weight – 3774 kg
  - Max gross weight – 11213 kg
- Total weight of truck combination (trailers included)

- Service/unladen weight – 23300 kg
- Max gross weight – 74000 kg

Just as in the case for the RAPDD-GR, when driving unloaded, few wheels in contact with the surface is desired which is why the tag axle in this case is lifted to increase startability and rolling resistance. For illustrating picture see Appendix L.

The total weight and weight combinations have been specified accordingly to the Swedish Transport Agency (Transportstyrelsen, 2014), (Transportstyrelsen, 2012), (Transportstyrelsen, 2015).

5.2 Calculations of hydraulics

In the following sections calculations regarding the velocities, torques and startability which can be expected for the different motor sizes and configurations are presented.

5.2.1 Previous test truck FH-1825

The C.H.FWD is mainly of use when the truck is unloaded or is driven without trailer connection, these are the occasions when the front axle is subjected to the largest percentage of the total truck weight and offers the highest advantage. However engaging the C.H.FWD when the truck is in maximum gross weight, the total produced driving force is of course also improved.

The theoretically achievable velocities of the hydraulic system to FH-1825 is 26,5 km/h in unloaded condition and 24,9 km/h when the front axle is loaded at its maximum. The difference in velocity is due to the expected wheel radius reduction when driving in fully loaded condition. However due to the mechanical ratios within the powertrain, the theoretical maximum velocity is limited according to equation (13), see Table 5.

Table 5 FH-1825 ratio velocity

		Motor size		Engine rpm									
		Drive ratio		0,00	222,22	444,44	666,67	888,89	1111,11	1333,33	1555,56	1777,78	2000,00
Gear	1	177,68	Speed in km/h	0,00	1,06	2,12	3,18	4,24	5,30	6,36	7,42	8,48	9,54
	2	118,03		0,00	1,35	2,70	4,05	5,40	6,75	8,10	9,45	10,80	12,16
	3	67,84		0,00	1,75	3,51	5,26	7,02	8,77	10,53	12,28	14,04	15,79
	4	31,86		0,00	2,23	4,47	6,70	8,93	11,17	13,40	15,63	17,87	20,10
	5	2,98		0,00	2,86	5,72	8,58	11,44	14,30	17,16	20,02	22,88	25,74
	6	-19,28		0,00	3,65	7,30	10,94	14,59	18,24	21,89	25,53	29,18	32,83
	7	-36,08		0,00	4,61	9,21	13,82	18,43	23,03	27,64	32,25	36,86	41,46
	8	-49,81		0,00	5,87	11,73	17,60	23,47	29,34	35,20	41,07	46,94	52,81
	9	-61,41		0,00	7,63	15,26	22,89	30,52	38,16	45,79	53,42	61,05	68,68
	10	-69,70		0,00	9,72	19,44	29,15	38,87	48,59	58,31	68,02	77,74	87,46
	11	-76,33		0,00	12,44	24,88	37,32	49,75	62,19	74,63	87,07	99,51	111,95
	12	-81,54		0,00	15,95	31,89	47,84	63,79	79,74	95,68	111,63	127,58	143,52

The highest gear which the hydraulic system can be fully used on is the fifth gear. But the safety factor is relatively low, and there is risk that the hydraulic motors will not manage tight corners and start to break the equipage.

The torque will be dependent on the efficiency of the motors, in particular the mechanical-hydraulic efficiency for angular velocities in between 0-25 rpm. The produced torque for two motors based on ten different values in the interval from 68 – 88 % efficiency can be seen in Table 6.

Table 6 Torque vs. efficiency FH-1825

Torque vs efficiency FH-1825										
Efficiency	0,68	0,7	0,72	0,75	0,77	0,79	0,81	0,84	0,86	0,88
Torque	6753,26	6973,96	7194,65	7415,35	7636,04	7856,74	8077,43	8298,13	8518,82	8739,52

As expected, the produced torque from the system is greatly increased at higher velocities, a fact which is not in favour for the startability.

The hydraulics supplier has specified that the 780 cc motors are dimensioned for a torque of 5580 Nm, which could be achieved if a pressure of 450 bar is used. As can be seen in Table 6 the highest achievable value was 8940 Nm for two motors, i.e. 4470 Nm per motor which is considerably lower than the 5580 Nm specified by the hydraulics supplier. This can be explained by the difference in used max working pressure and also possible differences in used efficiency.

The torque which the pump will be subjected to will be equal on all studied combinations since the variables to calculate this is identical in all cases. The calculated max torque on the pump was 595 Nm, which is higher than the 572 Nm, see 4.0, specified by the hydraulics supplier. But since FH-1825 is being fitted with the hydraulic system with used specifications this value is considered acceptable.

Further during freewheeling the maximum rotational speed has been set to 600 rpm by the hydraulics supplier (Arvidsson, 2015), which if converted to linearly velocity is about 121 km/h, using specified tyre subjected to zero load. The maximum theoretical velocity achieved by mechanical drive is as can be seen in Table 5 144 km/h. But the highest allowed speed for heavy trucks in Sweden is 90 km/h and in Finland and Norway 80 km/h, which means 121 km/h should never be exceeded.

For calculations of how the maximum velocities and the different torques has been calculated see Appendix O.

The maximum achievable torque determines the maximum inclination which can be conquered from a standstill, which is among other dependant on the friction coefficient and rolling resistance. Also, depending on if the truck is loaded or unloaded the weight distribution on the axles and the total weight is different; see 5.1, which also results in different startability. Therefore it is of interest to analyse the startability in both conditions to fully understand the systems limitations.

In Appendix M and Appendix N, the startability for FH-1825 can be seen, where the purple line indicates the startability with only rear wheels drive, the cyan line when the C.H.FWD is also engaged and the doted red line with M.FWD instead of C.H.FWD. The area below each of the lines are occasions where the truck can start from a standstill, area above the lines are inaccessible, meaning the truck will not be able to start from a stand still. To the left of the graph the rolling resistance for different road conditions has been inserted and below the graph is the friction coefficient of these different road conditions, further also referred as GTA parameters which previously have been described in section 3.8. The graph can be read by either disregarding the GTA parameters and only consider inclination, only consider the GTA parameters or including both inclination and GTA parameters simultaneously. In the two first scenarios “alfa” on the y-axis can be assessed directly from the graph. In the third case where both the inclination and the GTA parameters are regarded at the same time these two should be added together to get the total sum which then should be used as the alfa value in the graph. E.g. let’s say that truck FH-1825 is driving in unloaded condition on a gravel road which imposes an approximate rolling resistance of 0-5 % and a friction coefficient between 0,55-0,75 and the truck is to climb a slope with 10 % angle. The total alfa will in this case be 10-15 % which can be seen in Appendix M would require C.H.FWD or mechanical front wheel drive to be able to start as long as the friction coefficient is below 0.7.

As can be seen, there are two cyan lines in the two appendices regarding startability for FH-1825, Appendix M and Appendix N. The lower of the two lines is when the wheels have zero rotational speed and also the lowest efficiency, 68%. The upper is when the motors have obtained their maximum efficiency of 88 % where it is assumed to be relative stable for the higher rotational speeds. It can also be seen that in both graphs the two cyan lines breaks of after some time and withholds the same inclination as when there is no aid of FWD. Up to this brake of point the hydraulic motors utilises the surface friction to its fullest. After the brake of point the maximum torque of the hydraulic motors is reached and the surface friction cannot be used to its fullest, resulting in a constant improvement compared to when not having FWD compared to the previous increasing improvement.

Further, as can be seen in both appendices, mechanical FWD, red dashed line, offers superior performance regarding startability compared to no FWD and C.H.FWD. But the mechanical drive has some large disadvantages; today it can only be offered on CHH-HIGH and not CHH-MED or CHH-LOW, also the mechanical drive can only be implemented on Volvo´s 13 litres engines and not the larger 16 litres (Pettersson, 2016). Further, the hydraulics supplier has estimated that by replacing the M.FWD with C.H.FWD the service weight of the truck can be reduced by at least 400 kg (Poclain Hydraulics, n.d.).

The calculations regarding the C.H.FWD can be seen in, Appendix O, Appendix P and Appendix Q.

5.2.2 Upcoming four wheel motor torque, startability and velocities

The aim velocity which is desired that the upcoming trucks can utilize the C.H.D. to, is 50 km/h compared to the approximate 23-26 km/h of FH-1825. When having no slip on any wheel the flow can be equally distributed to each motor making it possible to combine equation (3) and (8) to following equation (20):

$$D_m = \frac{D_p * n_e * n_{volp} * n_{volm} * R * 2 * pi * i_{pto}}{20 * v * 60} \tag{20}$$

Where the rotational velocity n = v/R, and v is the traveling velocity [m/s] and R the wheel radius.

If using the largest possible radius of the specified wheels the only variable in the equation is the engine speed which out of economic reasons is not desired to have maximised during driving over a longer period. Due to this ten different engine speeds were used in the calculations to get ten different hydraulic motor sizes which should theoretically, given the criteria in the equation, have a top velocity of 50 km/h. The motors will however, as in the precious case, be limited by the ratios in the transmission as stated in equation (13).

The result for the calculations regarding motor size and achievable velocity when engaging two or four motors can be seen in Table 7 and Table 8 below.

Table 7 Four motor velocity, two engaged

		Motor size cc									
	Drive ratio	0	45,94	91,87	137,81	183,74	229,68	275,61	321,55	367,49	413,42
Gear	1	Inf	4615,11	2257,55	1471,70	1078,78	843,02	685,85	573,59	489,39	423,90
	2	Inf	3602,15	1751,07	1134,05	825,54	640,43	517,02	428,88	362,77	311,35
	3	Inf	2749,97	1324,98	849,99	612,49	469,99	374,99	307,14	256,25	216,66
	4	Inf	2138,97	1019,49	646,32	459,74	347,79	273,16	219,85	179,87	148,77
	5	Inf	1648,57	774,29	482,86	337,14	249,71	191,43	149,80	118,57	94,29
	6	Inf	1270,72	585,36	356,91	242,68	174,14	128,45	95,82	71,34	52,30
	7	Inf	985,32	442,66	261,77	171,33	117,06	80,89	55,05	35,66	20,59
	8	Inf	752,18	326,09	184,06	113,04	70,44	42,03	21,74	6,52	-5,31
	9	Inf	555,21	227,61	118,40	63,80	31,04	9,20	-6,40	-18,10	-27,20
	10	Inf	414,52	157,26	71,51	28,63	2,90	-14,25	-26,50	-35,68	-42,83
	11	Inf	301,97	100,99	33,99	0,49	-19,61	-33,00	-42,58	-49,75	-55,34
	12	Inf	213,54	56,77	4,51	-21,62	-37,29	-47,74	-55,21	-60,81	-65,16
		0	222,22	444,44	666,67	888,89	1111,11	1333,33	1555,56	1777,78	2000,00
		Engine speed to obtain 50 km/h for each motor size									

Table 8 Four motor velocity, four engaged

		Motor size cc									
	Drive ratio	0,00	45,94	91,87	137,81	183,74	229,68	275,61	321,55	367,49	413,42
Gear	1	Inf	2257,55	1078,78	685,85	489,39	371,51	292,93	236,79	194,69	161,95
	2	Inf	1751,07	825,54	517,02	362,77	270,21	208,51	164,44	131,38	105,67
	3	Inf	1324,98	612,49	374,99	256,25	185,00	137,50	103,57	78,12	58,33
	4	Inf	1019,49	459,74	273,16	179,87	123,90	86,58	59,93	39,94	24,39
	5	Inf	774,29	337,14	191,43	118,57	74,86	45,71	24,90	9,29	-2,86
	6	Inf	585,36	242,68	128,45	71,34	37,07	14,23	-2,09	-14,33	-23,85
	7	Inf	442,66	171,33	80,89	35,66	8,53	-9,56	-22,48	-32,17	-39,70
	8	Inf	326,09	113,04	42,03	6,52	-14,78	-28,99	-39,13	-46,74	-52,66
	9	Inf	227,61	63,80	9,20	-18,10	-34,48	-45,40	-53,20	-59,05	-63,60
	10	Inf	157,26	28,63	-14,25	-35,68	-48,55	-57,12	-63,25	-67,84	-71,42
	11	Inf	100,99	0,49	-33,00	-49,75	-59,80	-66,50	-71,29	-74,88	-77,67
	12	Inf	56,77	-21,62	-47,74	-60,81	-68,65	-73,87	-77,60	-80,40	-82,58
		0,00	222,22	444,44	666,67	888,89	1111,11	1333,33	1555,56	1777,78	2000,00
		Engine speed to obtain 50 km/h for each motor size									

In the tables green indicates that the combination with motor size and gear works and the hydraulic system will have a higher velocity than the mechanical drive. The achievable speed in each gear from engine speed up to 2000 rpm can be seen in Table 9 below. When having zero rotational engine speed the equation says that a velocity of 50 km/h can be achieved by a motor of zero volume, this is of course incorrect and a value which should be discarded.

Table 9 Velocities

		Speed in km/h									
	Engine RPM	0,00	222,22	444,44	666,67	888,89	1111,11	1333,33	1555,56	1777,78	2000,00
Gear	1	0,00	1,06	2,12	3,18	4,24	5,30	6,36	7,42	8,48	9,54
	2	0,00	1,35	2,70	4,05	5,40	6,75	8,10	9,45	10,80	12,16
	3	0,00	1,75	3,51	5,26	7,02	8,77	10,53	12,28	14,04	15,79
	4	0,00	2,23	4,47	6,70	8,93	11,17	13,40	15,63	17,87	20,10
	5	0,00	2,86	5,72	8,58	11,44	14,30	17,16	20,02	22,88	25,74
	6	0,00	3,65	7,30	10,94	14,59	18,24	21,89	25,53	29,18	32,83
	7	0,00	4,61	9,21	13,82	18,43	23,03	27,64	32,25	36,86	41,46
	8	0,00	5,87	11,73	17,60	23,47	29,34	35,20	41,07	46,94	52,81
	9	0,00	7,63	15,26	22,89	30,52	38,16	45,79	53,42	61,05	68,68
	10	0,00	9,72	19,44	29,15	38,87	48,59	58,31	68,02	77,74	87,46
	11	0,00	12,44	24,88	37,32	49,75	62,19	74,63	87,07	99,51	111,95
	12	0,00	15,95	31,89	47,84	63,79	79,74	95,68	111,63	127,58	143,52

From the tables it can be tempting to choose a smaller motor in order make it possible to drive at top C.H.D speed with better fuel economies. But with smaller motors the torque will be lowered and be moving towards zero. There is therefore important to know the torque the different motors produce, maximum torque will be as previously stated when engaging four motors, see Table 10, and when engaging two motors the produced torque will be reduced to half.

Table 10 Torque four motors

		Motor size cc									
	Torque [Nm]	0,00	45,94	91,87	137,81	183,74	229,68	275,61	321,55	367,49	413,42
Mechanical efficiency	0.6800	0,00	795,43	1590,85	2386,28	3181,71	3977,13	4772,56	5567,99	6363,41	7158,84
	0.7022	0,00	821,42	1642,84	2464,26	3285,68	4107,11	4928,53	5749,95	6571,37	7392,79
	0.7244	0,00	847,42	1694,83	2542,25	3389,66	4237,08	5084,49	5931,91	6779,32	7626,74
	0.7467	0,00	873,41	1746,82	2620,23	3493,64	4367,05	5240,46	6113,87	6987,28	7860,69
	0.7689	0,00	899,40	1798,81	2698,21	3597,62	4497,02	5396,42	6295,83	7195,23	8094,64
	0.7911	0,00	925,40	1850,80	2776,20	3701,59	4626,99	5552,39	6477,79	7403,19	8328,59
	0.8133	0,00	951,39	1902,79	2854,18	3805,57	4756,96	5708,36	6659,75	7611,14	8562,53
	0.8356	0,00	977,39	1954,77	2932,16	3909,55	4886,94	5864,32	6841,71	7819,10	8796,48
	0.8578	0,00	1003,38	2006,76	3010,14	4013,53	5016,91	6020,29	7023,67	8027,05	9030,43
	0.8800	0,00	1029,38	2058,75	3088,13	4117,50	5146,88	6176,25	7205,63	8235,01	9264,38

As can be seen, with increased motor size, increased maximum torque follows as a result. Therefore a trade-off between achievable top velocity and maximum torque will in this case be needed to be considered.

The calculated torque can be used to calculate the startability of the trucks. The calculations for the startability have been done in two different states, when the truck is unloaded with trailers attached and when being loaded to maximum service/unladen weight. As stated in 5.1.3 when driving in unloaded condition, pusher axle for RAPDD-GR and tag axle for RADT-GR are lifted in order to increase weight on driven wheels and reduce rolling resistance, thus increasing startability.

Calculations for the tables above in this section can be seen in Appendix R, Appendix S and Appendix T.

5.2.2.1 RAPDD-GR

The RAPDD-GR is a heavier configuration than RADT-GR and would require higher torque in order to start. The startability when in unloaded condition for RAPDD-GR can be seen in Appendix U. Just as for the calculations for FH-1825 the bottom dark purple line is without any front wheel assistance and the red dashed line is when using mechanical front wheel drive together with the regular mechanical drive. The lines in between are the different contributions from the different sized motors. Each size is indicated with two lines, one where the motors are performing at the lowest efficiency of 68 % and the other at the relative stable 88 %. As can be expected the largest motor also produces most aid in startability and the smallest the lowest aid.

When loading the truck to maximum gross weight required torque to start will be increased, leading to reduced startability performance, see Appendix V. Also here the larger the motor size the larger aid is provided to the startability.

Summation of contribution for the different C.H.D configurations can be seen in next section 5.2.2.2, Table 11 and calculations for star ability regarding RAPDD-GR using equally sized motors on front axle and pusher axle can be seen in Appendix W and Appendix X.

5.2.2.2 RADT-GR

For the lighter combination RADT-GR the same arrangements regarding colouring, lines and how to analyse the graphs as in the RAPDD-GR case also applies in Appendix Y and Appendix Z. As can be seen, also here the larger the motor the higher is the aid in startability and the higher total mass that is to be moved also higher torque is needed. For the calculations regarding obtained results for RADT-GR's startability using equally sized motors can be seen in Appendix AA and Appendix BB.

If the graphs for RAPDD and RADT are compared and summarised into a table, see Table 11 it can be seen that the startability and contribution from C.H.D for the RAPDD combination is higher than RADT. This can be explained by that the axle pressure is higher for the RAPDD combination, allowing for higher traction force with the same torque. Also when the trucks are unloaded the torque is not a limiting factor and the truck should in theory be able to pull itself up a vertical wall if the fiction coefficient is neglected.

Table 11 Summary for RAPDD-GR and RADT-GR equally sized motors

Configuration	Unloaded / Loaded	Alfa at torque stop	Continuous improvement with C.H.D
RAPDD-GR	Unloaded	5,8 - 7,3 %	3 - 3,9 %
	Loaded	3,6 - 4,5 %	1,3 - 2 %
RADT-GR	Unloaded	5,2 - 6,9 %	2,1- 3 %
	Loaded	3,5 - 4,5 %	2 – 2,6 %

6.0 Dimensioning of hoses and fittings

In the following sections expected flow velocities in hoses and pipes are calculated, the criteria for choice of fittings is assessed and suggestion of suitable fittings for both RAPDD-GR and RADT-GR are presented.

6.1 Flow velocities in hoses

As stated in the 3.7.3, the volumetric flow, velocity and inner diameter of a hose, all correlate. With higher pressure and smaller inner diameter of the hose, higher velocities will be achieved which never should exceed 8 m /s.

6.1.1 Velocities on FH-1825

Due to confidentiality, illustrative pictures and designations of the flow through the system used in the thesis work cannot be presented in this section and thesis.

The flow which exits the pump will be divided equally to each motor if the truck is moving in a constant velocity, where each wheel is spinning in the same rotational speed. This means that each of the two wheels receives half the volumetric flow produced from the pump, losses in the routing is here neglected.

The feeder pump produces a pressure of 25 bar to the pump during usage and the two low pressure lines from the motors serves as leakage lines.

When the system is in freewheel mode, the feeder pump is set to produce a pressure of 6 bar. The volumetric flow which circulates to and from the motors is here 4,5 l/min in the feeder line, and 2,25 l/min in the high pressure lines.

When engaging the system the entire flow from the pump is equally divided over the motors if no slip is occurring. Since the motors do not have a hydraulic efficiency of 100 % some oil will leak and in the feeder line which now serves as an extra drainage the flow will be 2,25 l/min.

6.1.1.1 Velocities with hydraulics supplier’s specification

Using the hydraulics supplier’s recommendation on hose sizes and the described 90 cc pump and 19,6 cc feeder pump, results in flow velocities which can be seen in,

Table 12. The calculations have been simplified by dividing the flow equally over all lines.

Table 12 Flow velocities using the hydraulics supplier's recommendation

Line port - port	Type of line	Received volumetric flow [l/min]	Flow velocity [m/s]
Pump –>Valve block	High pressure	211	7.2
valve block <-> motors	High pressure	105	6.9
BPR –> (filter) -> Pump house	Feeder line	No data	-
Pump <-> valve block	Feeder line	9	0.85
Feeder line / return	Feeder line / return	4.5	0.66
Pump house -> (filter) -> tank	Feeder line	No data	-
Tank -> feeder pump	Suction line	46	0.8

The flow in the high pressure lines is above the recommended 5-6 m/s, but is lower than 8 m/s which is the velocity that is not to be exceeded.

The line which serves as both feeder line and return line has a flow which is below the recomended 3 m/s. From the motors to the pump housing the velocity is below 1 m/s in all cases, less than a third of recommended maximum. This seems very low but can be explained by a desire of having a low pressure loss after the pump, and therefore having higher torque since the differential pressure will be lower (Arvidsson, 2016).

The suction line is well below the maximum 1 m/s at 0.8 m/s.

Used calculations to calculate the flow velocities based on the recommended dimensions provided by the hydraulics supplier can be seen in Appendix CC.

6.1.1.2 Actual dimensions and flow velocities on FH-1825

The routing on FH-1825 has been conducted fully by Michael Andersson, where Volvo had little to none involvement in the specification of the routing of the system. Used dimensions of pipes and hoses on test truck FH-1825 can be seen in

Table 13.

Table 13 Used dimensions FH-1825

Description	Type of line	Inner diameter
Pump –>Valve block	High pressure	1”
valve block <-> motors	High pressure	5/8”
Pump -> BPR	Feeder line	18 mm pipe -> 19 mm hose ¾”
BPR –> (filter) -> Pump house	Feeder line	19 mm hose ¾”
Pump <-> valve block	Feeder pressure/leakage	5/8”
Valve block <-> motors	Feeder pressure/ leakage	½”
BPR -> Pump house	Goes to tank, atmospheric pressure	½”
Pump house -> (filter) -> tank	Suction line	¾”
Tank -> feeder pump	Suction line	1 ¼ “

Using the known flow rates and the used inner dimensions of the routed system on FH-1825, expected flow velocities can be calculated and are here presented in Table 14.

Table 14 Flow velocities on FH-1825

Type of line	Received volumetric flow [l/min]	Flow velocity [m/s]
High pressure	211	6.9
High pressure	105	8.9
Feeder line	No data	-
Feeder line	9	1.18
Feeder line / return	4.5	0.59
Feeder line	No data	-
Suction line	46	0.97

Compared to the hydraulics supplier’s specification which has been done with DN millimetre standard, the hoses on FH-1825 have been specified using inches. If comparing the high pressure lines from pump to valve block, feeder line/ return between valve block and motors and the suction line, these values differ somewhat from

Table 12. The difference between the tables is due to the difference in standard used to specify the system. If converting the DN specification used by the hydraulics supplier to inches the same dimensions would be achieved and also an equal result in the two cases.

The two lines which can be observed of having a higher flow velocity are the high pressure lines between the valve block and the motors and the line between feeder pump and valve block. The high pressure lines have received a relative high increase, an increase which exceeds the absolute maximum of 8 m/s and is therefore under dimensioned. The increase in feeder line velocity is also relatively high but could be increased further if necessary and a decrease in hose diameter is of desire.

Calculations used to obtain the flow velocities on FH-1825 can be seen in Appendix DD.

6.1.1.3 Upcoming four wheel motor flow velocities and diameters

Since the new upcoming solution can alter between freewheeling on all four motors, engaging two motors, or engaging all four motors, the flow in the lines will be changing in each of these cases. When setting all four motors to freewheel during driving, the feeder line flow will be divided over all four motors and is in this thesis work viewed as equally divided over all four motors. That means that when all four motors are in free wheel mode the flow to the motors through the feeder lines would be 2,25 l/min. But the previously 4,5 l/min flow is a set value of the system and should not be decreased, therefore this flow should remain unchanged for the four motor solution.

When engaging two motors the high pressure maximum flow will be equal to the flow in previous calculations, 105 l/min, and when engaging all four motors this flow will be halved since the number of motors are doubled, 52.5 l/min.

If there is a possibility to enable only the front motors or the rear motors and letting the other pair free wheel, the maximum flow velocities in the system will be unchanged. If the option of only having one of the motor pairs engaged and the other pair only engaged together with the first motor pair, i.e. the entire system is engaged, the maximum flow in the motor pair which only is engaged when engaging all four motors will be lower, see Table 15.

Table 15 Reduced flow velocity

Type of line	Received volumetric flow [l/min]	Flow velocity [m/s]
High pressure	52.5	4.44
Feeder line / return	2.25	0.59

Since it is desired to have the possibility to engage each motor pair individually the maximum flow will remains the same. Further, used dimensions for the high pressure lines to the motors and the feeder line will be kept the same disregarding the high flow velocities. The reasoning for this is that in a previous test truck a 140 cc pump was used and high pressure hoses with the inner diameter ¾” and feeder hoses with the inner diameter 5/8” which resulted in a maximum flow velocity of 10.7 m/s for the high pressure line and 0.6631 m/s for the feeder line. During the testing of that test truck no problems caused by flow velocity could be observed and therefore determined to be a working solution.

Calculations used to obtain the flow velocities with the reduced flow velocity and the flow velocities using the 140 cc pump see Appendix EE and Appendix FF.

6.2 Analysing criteria for nipples & fittings

As described in the theory chapter, in order to choose correct fitting it is important to view the different design criteria of the system. But since the hoses are predefined to parkers hoses 797 TC and 462 ST with predefined compatible fittings the most criteria are already fulfilled and lesser concern to these can be taken.

The recommended crimp fittings for the high pressure hoses 797 TC are fittings from the 77 series from Parker and for low pressure and feeder hoses 462 ST Parkers crimp fittings from series 46/48 is recommended.

Taking Jörgen Ahlberg’s statement that metal to metal fittings are highly sensitive to scratches and very fine surface roughness is needed, these solutions are if possible discarded and solutions with O-rings are of main interest. Standard O-rings are specified to work in a range between -40^{o}C and +105^{o}C, which is within the set temperature range for the system.

Analysing Parkers 77 series fittings, the choice of possible fittings can be narrowed down to three different fittings; Flange code 62, Seal-Lok and Metric 24^{o} cone, where Seal-Lok and Metric 24^{o} cone are relative equal in configuration. Table over suitable candidates from the 77 series can be found in Table 16.

Table 16 Possible 77 series fittings

Fitting	Name	Axial Length	Max diameter	Radial Length from centre line
Flange code 62	16A77-12-10	74.9	40.7	20.35
Seal-Lok, female	1JS77-10-10	81.5	30	15
Seal-Lok, male	1J077-12-12	75.4	30	15
Seal-Lok, female, 45^{o} Elbow	1J777-10-10	83.3	30	16 (to the 45^{o} centre line)
Seal-Lok, female, 90o Elbow	1J177-10-10, 1J977-10-10	81.5	30	70 long drop, 32 short drop
Metric 24^{o} cone, straight female	1C977-20-10	67	36	18
Metric 24o cone, straight male	1D277-20-10, 1ZM77-22-10	72.4, 71.8	30, 36	15, 18
Metric 24^{o} cone, 45^{o} Elbow	10C77-20-10	106.9	36	28

Flange fittings require flange clamps to be properly tightened, working clamps for the flange code 62 fitting provided by parker can be found in

Table 17.

Table 17 Clamps flange code 62

Clamp name	Length	Width	Height	Screw type
FUS	60	71.4	28	M10x35, 3/8x1 1/2
FUSM	60	71.4	28	M 10
FHSF	30*2	71	20	M10x35, 3/8x1 1/2
FUSF	60	71	20	M10x35, 3/8x1 1/2

As for the low pressure hoses to and from the motors, the fittings will be smaller since the hoses are of smaller size. To reduce confusion, the fittings for these hoses should match the types chosen for the high pressure hoses. In the 46/48 series Parker's Seal-Lok solution does not exist, but this solution is more or less an ordinary seal lock solution with O-ring and UNF thread which is why ORFS fitting will here instead be considered (Hannifin, n.d.).

Following dimensions of fittings in series 46/48 have been found:

Table over suitable candidates from the 46 and 48 series can be found in

Table 18 (Parker Hannifin, n.d.).

Table 18 Possible 46 & 48 series fittings

Fitting	Name	Axial Length	Max diameter	Radial Length from centre line
Metric 24o cone, straight female	1CA46-18-8	49	32	16
Metric 24^o cone, straight male	1D046-18-8, 1D048-18-8	53	27	13.5
Metric 24o cone, 45 o Elbow	1CE46-18-8	71	32	22
Metric 24^o cone, 90^o Elbow	1CF46-18-8	65	32	43
Flange Straight	11546-12-8	51	38	19
Flange 45 ^{o} Elbow	11746-12-8, 11748-12-8	70	38	21
Flange 90 o Elbow	11946-12-8, 11948-12-8	70	38	43
ORFS, female straight	1JC46-12-8	52	36	18
ORFS, male straight	1JS46-10-8	63	30	15
ORFS, 45 ^{o} Elbow	1J746-12-8	77	36	21
ORFS, 90 o Elbow	1J949-12-8	58	36	48

(Parker Hannifin, n.d.)

The final choice of fittings can be seen in section 8.5.

7.0 The hydraulic system and schematics

The following sections include the criteria of the hydraulic system, existing schematics for FH-1825 and also concepts of the schematics for the valve block for the C.H.4WD system.

7.1 The criteria of the hydraulic system

To fully understand how the hydraulic scheme should work and be designed some sort of logic of what the system should be capable to do is needed to be defined.

There are three different modes which should be possible to obtain:

1. Driving without the C.H.D engaged
2. Driving with only C.H.FWD engaged
3. Driving with C.H.4WD engaged

The system should also automatically be disengaged if velocity exceeds predefined top velocity for both when only engaging C.H.2WD and C.H.4WD. The system should then be automatically reengaged if the velocity goes below set top velocity, given that the system is turned on. Maximum possible velocity will be higher if only two wheels are engaged, which also would differ between front axle and pusher axle if the choice of differently sized motors on each axle is chosen. The system would also need to identify these scenarios and adapt the maximum allowed engaged speed accordingly.

7.2 The schematics of today

Due to confidentiality, illustrative pictures of the schematics provided by the hydraulics supplier are not included in this report, designations of the ports as A, B, etc are however included.

The block has two high pressure ports which are connected directly to the pump, port A and B; the pressure these two ports are subjected to is the full pressure of 420 bar, with 450 bar spikes. When the C.H.FWD is engaged the high pressure oil is lead to the wheel motors through either port AM1 and AM2 or BM1 or BM2. The port pair which is not used to transport high pressure oil to the wheel motors receives low pressure oil from the wheel motors which eventually is lead back to the pump. During freewheeling, flow through both port pair A1, A2 and B1, B2 is lead to the tank via exit port L, emptying the oil within the pistons.

Port G is connected to the feeder pump which while the C.H.FWD is engaged serves as a leakage line together with LM1 and LM2 from the piston house back to the feeder pump which feeds the main pump with new oil. In free wheel mode the flow in this line is reversed, resulting in filling of the piston house and an increased pressure, forcing the pistons to contract.

The controlling of engaging or disengaging the C.H.FWD is done through activating or deactivating solenoids 2.6.1, 2.6.2 and 2.11. When activating solenoid 2.6.1 the direct connection between line A and B is removed and directed towards the hydraulic controlled valve. When activating solenoids 2.6.2 and 2.11 the flow from the feeder pump is redirected through 2.11 to 2.6.2 to the hydraulic controlled pilot valve which is engaged and opens the passage for line A and B, thus enabling high pressure to the wheel motors (Hydraulics & Pneumatics, 2006) & (Arvidsson, 2016).

7.3 The new valve block

In the following section the generation of different concepts and the final choice of concept is described and explained.

7.3.1 Concept generation

In order to keep the required development cost as low as possible the new solutions was aimed to be as similar as the existing solution of today as possible. The idea generation concluded into three different concepts which all could solve the three described requirements in section 7.1 and two of the concepts also could enable the fourth desire.The idea generation was primary based on brainstorming and discussion with experts in the area.

Illustrative pictures over the concepts of the schematics for the new valve block(s) had to be excluded in this report due to confidentiality.

7.3.1.1 Concept 1

The first concept would require a completely new hydraulic valve block, where both the front and the rear hydraulic drive would be controlled. The block has two entry ports for the high pressure and one entry port for the feeder line and eight exit ports to the motors, doubled compared to the previous valve block which have four ports.

The basic idea of the block is the same as the one used for FH-1825, with the exception that there are two sets of control valve systems, one for the front motors and one for the rear motors. The feeder flow enters in one port where it is then divided into two lines for front and rear motors and depending on the settings of the solenoid the flow will either open the pilot valve or fill the motor house and contract the pistons.

The flow from the feeder pump could just as for the high pressure line be divided into two ports, and the split of lines inside the block would in that case be removed. The advantage of having the splits inside the block is that fewer nipples are needed; a drawback is that it will probably make the valve block more complex and also higher losses can be expected. Using this configuration, enabling all four motors, only the front motors or only the rear motors can be achieved.

7.3.1.2 Concept 2

Concept 2 is designed to be connected with the primary valve block via one of the high pressure ports AM1 or AM2 and BM1 or BM2. This means that in these two ports leading to the motors and to the valve block, tee fittings needs to be mounted. Since the connection is of parallel type the pressure will be kept constant and since all wheels are connected to the surface the flow will be equal to all motors if no slip is occurring. The concept originates from the existing valve block where the only difference is the lack of valves 2.10 and 2.6.1 described in the existing schematics. The clutch function which valve 2.6.1 performs is not needed in this block since; if the entire system is disabled the high pressure flow will circulate in the primary valve block. Using this configuration only four wheel drive and front wheel drive can be achieved.

7.3.1.3 Concept 3

Concept 3 is simply to add a second identical valve block where the high pressure flow from the pump and the flow from the feeder pump are divided with tee fittings to both valve blocks, resulting in an equal split of flow. To the tank from the valve blocks the two lines needs to be contracted into one line using a tee fitting. Each motor pair will be controlled by one valve block, i.e. one valve block controls the front motors and one valve block the rear motors. The two valve blocks are then individually controlled by the electric control unit. This solution would enable four wheel drive, front wheel drive and rear wheel drive. Further it would also be the most cost effective solution since it would not require development of a new valve block. The only thing needed is the development of new software, but that is required for all concepts.

7.3.2 Final choice of hydraulic schematics

Together with the hydraulics supplier the three different concepts were discussed and reviewed in order to decide which would be most suitable for this project. Since concept 2 did not fulfil all requirements and the desire, this concept was removed. The remaining two concepts were judged depending on respective pros and cons. Concept one is more compact and offers a more sustainable solution for the future, but it would also cost more since it has to be developed and manufactured. For concept 3 all hardware already exists and the only new thing that is needed to be developed is new software. The con with this solution is that it would require more space which result in possible issues with packaging, also an increased number of hoses and fittings would be required in order to realise this solution.

With the presented pros and cons, concept 3 was chosen to be the most sensible solution for upcoming test vehicles. The major factor of the choice was cost, where the hydraulics supplier wants to keep the costs as low as possible. If this project were to be taken to serial production concept 1 would be of higher interest.

8.0 Packaging study

The following chapter focuses on the conducted packaging study containing concept development, concept screening and presentation of final concept for both RAPDD-GR and RADT-GR.

8.1 Tolerances and obstacles

In order to produce a feasible result regarding the packaging concepts it is important to know what kind of tolerances and obstacles which needs to be concerned.

The combinations, RAPDD-GR and RADT-GR, each have individual issues which need to be concerned, but some areas are more of a general nature and apply for both:

- The closest allowed distance an object is allowed to be placed from the tire is 60 mm, where 50 mm is due to the snow chains used during the winter and the remaining 10 mm due to the aging of the tire. Objects fitted close to the inside of the rims have to have at least 10 mm distance from the rims (Söder, 2016).
- Rule of thumb free distance between parts:
    - Two moving parts next to each other – 1” free space
    - Moving part next to rigid part – ½ “ free space
    - Rigid part next to rigid part – ¼” free space (Söder, 2016)
- A rule of thumb in the area of the suspension is; fasteners which are holding a part which rotates around the fasteners centre axle are so called sensitive fasteners and should not be used to add additional load.
- When adding parts to be fastened with the screws that connect the axles with the suspension it is important to us a thick material, approximately minimum of 20-25 mm, which have a high hardness in order to have low deformation when tightening the fasters (Hendriks, 2016).

However the majority of the packaging issues will be individual for each combination and are discussed in the following two sections.

8.1.1 RAPDD-GR

At the wheel ends of the pusher axles there are three air bellows located on each side, these air bellows can increase in diameter when compressed during lift/roll of the axles. Therefore there needs to be an area around the bellows which is free from other parts to allow the bellows to swell unrestricted. For the air bellows mounted on the driven axles, part number 21244937, the free distance is a diameter of 275 mm from bellow centre, regarding the two bellows for the pusher axle, no information has been found and therefore the distance rule of thumb previously described is here applied.

Max/min lift in millimetre and max/min roll in degrees defined by Volvo is following:

- Max steering angle
    - $\pm 19.9^{o}$
- Lateral motion max/min
    - + 160 mm

-
  - – 90 mm
- Max roll angle
  - ± 6 º
- Max lateral motion at max roll
  - ± 50 mm

8.1.2 RADT-GR

The tie rod arms which have been developed and are used for the RADT-GR installation imposes problems and would be impossible to use together with the C.H.D. This is due to a knob on the tie rod arms where the tie rod is connected, see Figure 15.

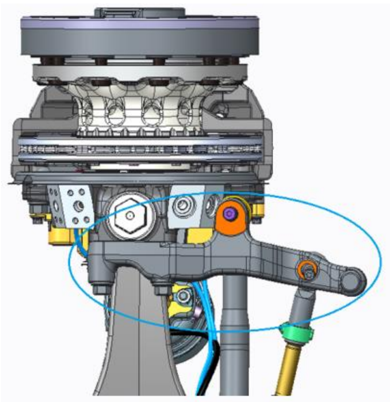

Figure 15 Tie rod

One solution to this is to change the tie rod arms and tie rod to another model where this knob does not exist. By directive from Sten Ragnhult at Volvo Trucks the tie rod arms and tie rod from RADDT-G2 was chosen to replace the otherwise standard tie rod on RADT-GR. This shorter tie rod from RADDT-G2 will result in an incorrect Ackermann angle which will result in more slip for both wheels on the tag axle.

The tag axle is fitted with several holes intended to fasten cables and hoses to these are all allowed to use for intended purpose.

Air bellows which can be of concern of the routing are the bellow in front and behind the tag axle on each side. The front air bellow, part number 22101721, has a free clearance diameter of 320 mm from the bellow centre and the rear air bellow, part number 22101719, has a clearance of 275 mm from the bellow centre.

Max/min lift in millimetre and max/min roll in degrees defined by Volvo is following:

- Steering angle
  - Now: 11.5º
  - Future desire: 25º
- Lateral motion max/min
  - + 130 mm
  - – 180
- Max roll angle

- +- 6 at maximum
- Max lateral motion at max roll
  - ± 50 mm

8.2 Concept Generation of routing

Concepts for RAPDD-GR were generated during a concept generation session, which included five participants from different areas and lasted three hours. The session was divided into three parts, brainstorming, categorisation and evaluation.

During the brainstorming all participants had in total 30 minutes with small breaks to come up with ideas which then were to be sketched and described on post-it notes. The brainstorming resulted in a total of 16 possible solutions of how the routing could be solved.

The 16 different solutions were then discussed and grouped together if deemed to have high enough similarity. Also the solutions were divided into two categories; Solutions to the frame, named A- and AA-concepts and solutions at the wheel end, named B-concepts. The difference between A- and AA-concepts is that the A-concepts were developed for the pusher axle, and the AA-concepts to the tag axle, where there were four A-concepts and four AA-concepts.

The four A-concepts in general terms were following:

- Following axles front side, then connect in some sort of adapter and vertically ascend to chassis side
- Following axle on rear side, then connect in some sort of adapter and vertically ascend to chassis side
- Follow cabling for AUXPark (Auxiliary parking brake) to chassis side.
- Going from air bellow for pusher to chassis side.

The four AA-concepts in general terms were following:

- Use the holes in the tag axle to fit some bracket to hold hoses. Change to pipes to middle of axle where change to hoses is made. Hoses routed along the blue V-stays to chassis
- Routing on the front side of the axle. Change to pipes to middle of axle where change to hoses is made. Hoses routed along the blue V-stays, group name 22238525_rear to chassis, see Figure 16.
- Routing towards the front directly to the chassis and down under the frame sides before the driven axle
- Routing backwards directly to the chassis and down under the frame sides

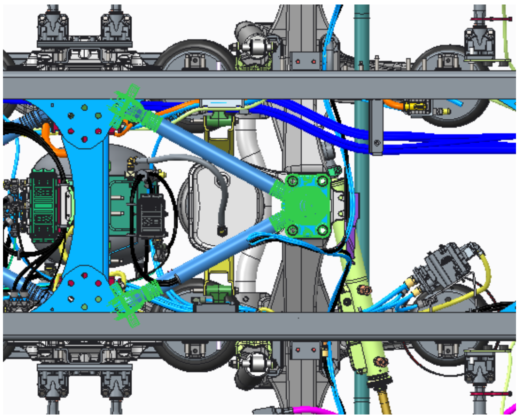

Figure 16 V-stays

The eight B-concepts in general terms were following:

- Routing all four hoses between AUXPark and axle.
- All four hoses are routed around the AUXPark where before and after AUXPark the hoses are as close the wheels as possible to increase bending radius and hose length.
- Using swivel fittings to handle the rotational motions.
- Bendable fitting to handle the rotational motions.
- Using as small radius as possible from the spindle ports to reach the axle/chassis.
- Two hoses are led below AUXPark and above the axle; the other two led upwards then back down.
- Routing over the wheel brake entering from the front side.
- Routing over the wheel brake entering from the rear side.

The B concepts was said to be realised by either having hoses directly from the ports of the wheel spindle, or by first using pipes then switch to hoses. Having hoses from the start reduced the required space for fittings in the change from pipes to hoses, but could be difficult to fit due to the hoses minimum bending radius. Also, since hoses are more vulnerable, a protecting bracket would be required, something which is not desired since the wheel should be able to be submerged up to 40 % in mud.

8.3 Concept evaluation

Since the A-, AA- and B-concepts all are sub solutions to complete solutions; a variant of the morphological matrix was used to identify the compatibility between A- and B-concepts, and AA- and B-concepts, see Table 19. The compatibility between a few of the concepts could immediately be determined with logical reasoning, other needed further assessment in CAD in order to be determined whether these were compatible or not. The areas filled with red & "No1" indicate that there is no compatibility between the two sub concepts from each row and column and the choice was taken without the usage of CAD support. The areas filled with pink colour & "No2" indicates that there is no compatibility between the sub concepts and the choice was taken once analysed in CAD environment. Green & "Yes" area indicates that there is compatibility between two sub concepts which have been verified in CAD. The area filled with orange & "Maybe" indicate too risky to continue, but might be possible with further evaluation.

Table 19 Morphological matrix

	1A	1A2	2A	3A	1AA	2AA	3AA	4AA
1B	No1	Yes	Yes	Yes	Yes	Yes	Yes	Yes
2B	No1	Yes	Yes	Yes	No1	No1	Yes	Yes
3B	No1	Maybe	Maybe	Maybe	Maybe	Maybe	Maybe	Maybe
4B	No1	No1	No1	No1	No1	No1	No1	No1
5B	No1	No2	No2	No2	No2	No2	No2	No2
6B	No1	No2	No2	No2	No2	No2	No2	No2
7B	No1	No2	No2	No2	Yes	No1	Yes	Yes
8B	No1	No2	No2	No2	No1	Yes	Yes	Yes

8.3.1 The non-compatible concepts

In the following section the concepts indicated with red & "No1", pink & "No2" and orange & "Maybe" are discussed and explained why these have been eliminated.

8.3.1.1 Concepts eliminated without CAD assessment (red & "No1")

The concepts that were deemed to be non-compatible even before assessment in CAD environment involved 21 different concepts. The argument to why the different concepts were discarded differs between concepts to concepts.

The 1A concepts were all discarded since the pusher is located lower than the front axle resulting in that the front side of the pusher is more exposed for possible debris and similar, see Figure 17. In the picture the lowest transverse part is the pusher axle. Since the system handles highly pressurised hot oil this risk was viewed to high and was therefore eliminated.

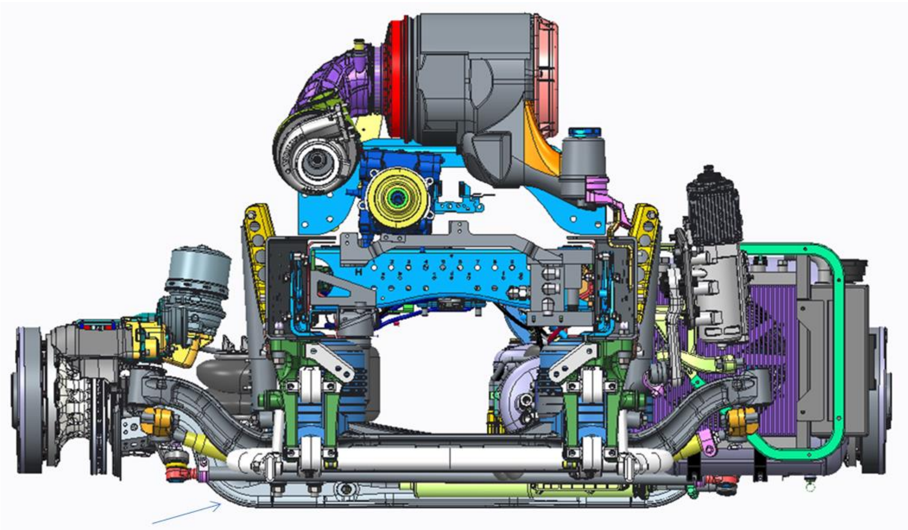

Figure 17 Illustration pusher axle

All 4B concepts were also eliminated without further inspection in CAD environment, this decision was based on the fact that an existing solution of this type could not be found and is therefore viewed as non-existent.

The combinations 2B1AA and 2B2AA were eliminated with the reasoning that the minimum bending radius would be a problem, resulting in that an unreasonable hose length and routing would be required.

The 3B concepts, using a swivel fitting were also eliminated before any assessments in CAD was performed. This decision was based on the study visit at VCE (Volvo Construction Equipment) where Jörgen Ahlberg expressed his concerns of the shorter life time of swivel fittings for this particular application and the increased risk of premature leakage. The reason why these concepts are orange and not red or pink is due to the fact that removing twisting in the hoses completely can be proven difficult and using swivel connections might be the only way to solve this problem. Further, encapsulate the swivel fittings in an elastic material would remove the swivels contact with dirt and could increase the lifetime.

7B2AA and 8B1AA were eliminated due to the expected complexity and the problems expected with that type of installation.

8.3.1.2 Concepts eliminated with CAD assessment (pink & "No2")

All 5B concepts were eliminated once routed in CAD environment, the reason is the using as small hose length and radius as possible would result in that the hoses highest position would be over the chassis side, which is not allowed. The same reasoning applies for all 6B concepts, where two hoses will have the highest point above the chassis side in driving height, see Figure 18.

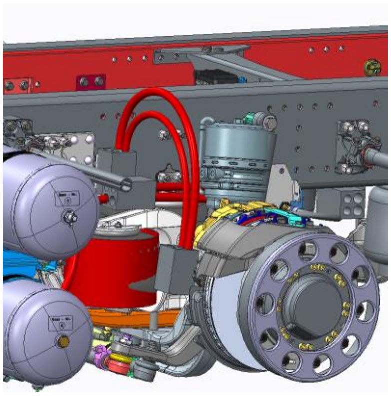

Figure 18 5B concepts

Both 7B and 8B for all #A solutions were eliminated due to the location of the AUXPark making further extensions of parts in vertical direction prohibited, see Figure 19.

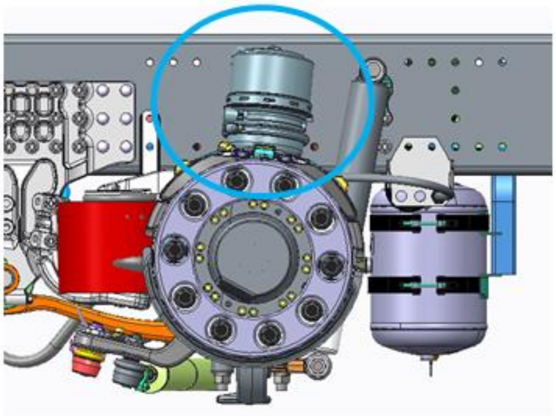

Figure 19 Side view, wheel end

8.3.2 Description of compatible concepts

The compatible concepts which had been derived can be viewed in Appendix GG where both a descriptive text and illustrative picture is presented for each concept.

8.4 Elimination of concepts

In order to have the attributes of each complete concept well-defined and understood a pros and cons list was derived for both the RAPDD-GR concepts and the RADT-GR concepts, see Appendix HH. Due to the size of the pros and cons list it was deemed too large and complex to use as an elimination method. The pros and cons list was instead used as a foundation to the Kesselring matrices, where the different pros and cons for the concepts were used as criteria. The concepts for RAPDD-GR and RADT-GR were evaluated separately in individual Kesselring matrices, where one concept for RAPDD-GR and two concepts for RADT-GR were chosen for detailed design.

As can be seen in Appendix II and Appendix JJ, the matrices has two total scores and ranking, with a criteria called "perceived feeling" separating the two. The desired effect from adding this criterion which usually is not seen in a Kesselring is that the correct concepts is also the winning concepts, the concepts which has the highest criteria fulfilment but also the highest perceived feeling.

The winning concept from the RAPDD-GR candidates was proven to be concept 1B2A and on second place concept 2B2A where only concept 1B2A was chosen to be further developed.

From the elimination for RADT-GR two concepts were chosen to be further evaluated with detailed designs, the two winning concepts were 1B4AA and 2B4AA. As can be seen concept 2B3AA also received a top ranking if "perceived feeling" is neglected. However after further evaluation it was noted that there are little to no possibility to route the hoses from the backside of the axle, due to the tie rod arm configuration, see Figure 20. This lead to that concept 2B3AA received lower rating in "perceived feeling", and received therefore a lower total score which resulted in a fourth place.

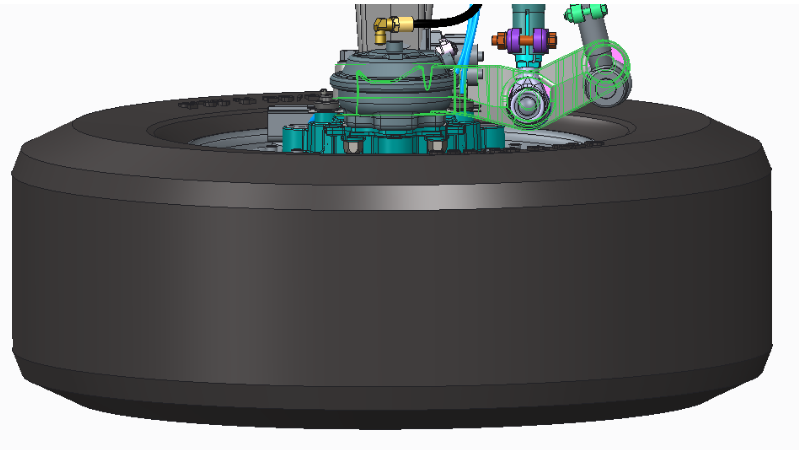

Figure 20 Top view wheel end tag

Note here that the tie rod arm seen in Figure 20 is not standard on RADT-GR but on RADDT-G2.

8.5 Detailed design / Final concept

For all transitions between pipes and hoses bulkhead unions was chosen, based on consultation with both Jörgen Ahlberg and Michael Andersson, due to the fact that these fittings offer a more compact solution which suits this project.

The development of the detailed and final design started with solving how to rout the hoses from the wheel end to the chassis which, when was performed enabled the routing of the pipes from the spindle and the pipes along the chassis sides. In section RAPDD-GR and RADT-GR the final design of the concepts for the two truck combinations are described.

A BOM (bill of material) for the routing solutions of pipes and hoses for both RAPDD-GR and RADT-GR has been established. The two BOMs are due to confidentiality not presented in this thesis report.

8.5.1 RAPDD-GR

The pipes which are connected to the spindle are bent to a bracket located between the AuxPark and the axle, see Appendix KK. The pipes have the outer diameter of 15 mm and 20 mm, where the 15 mm pipes are connected to the spindle with straight GEO15LM fittings and the 20 mm pipes with the 90 degree SAE fitting PAFG-90L.

The layout of the pipes in the picture and in the Creo Parametric model should only be used as an illustration of the path for the pipes from the spindle to the bulkhead fittings and not as an exact layout.

The bracket, a208649_bracketauxspindlerapdd2, which the bulkhead unions are mounted on to, is a bent sheet metal piece with the thickness 4 mm; the bracket is secured to the spindle with two M22 screws, see Appendix LL.

The hoses are connected to the bulkhead unions with female, metric threaded crimp fittings, 1C977-20-10 for the 5/8” hoses and 1CA46-18-8 for the 1/2” hoses. The four hoses on each side have a length from 621 – 706 mm with the total length of 2691 mm on both sides. The other end the hoses have been fitted with 45^{o} fittings, which are connected to bulkhead fittings which all is supported by a bracket mounted to the chassis side see Appendix MM.

A concern was that the hoses would interfere with the mud flaps which would require modification of the mud flaps. This was proven in Creo Parametrics that it should not be a problem and all possible axle heights, roll angles and steering angles should be possible without collision, see Appendix NN.

To the bulkheads which are mounted to the bracket on the chassis side, pipes are connected which are lead to a connection block. The reason why there is a transition to pipes is due to the fact that the pipes have a smaller outer diameter, thus occupying less packaging space. On the left side the pipes are led directly to the connection block and on the right side to the connection block via a clamp which is attached to the crossbeam.

The connection block will, if the project is realised, be produced by an external company specialised in the area which will do the final design, a recommended company by the hydraulics supplier is Side System AB. The block is therefore within this project only specified with the following:

- Outer dimensions
- Locations of hydraulic ports with required M-threads
    - Four M27 ports
    - Six M18 ports located on opposite sides
    - Two M42 ports
- Holes for attaching four M8 screws to attach the block to holding bracket
- Schematics of the flow lines see Appendix OO and Appendix PP

For the four M8 holes only three holes will be used when mounted, which three depends on whether mounted on RAPDD-GR or RADT-GR, see Appendix QQ and Appendix RR.

From the side where the M42 and M18 ports are located, two 30 mm and 15 mm in diameter pipes are to be connected. These pipes are led straight forward towards the valve block, supported by two clamps placed at a distance from the connection block of approximately 1,4 m to a bracket which is mounted to a cross member. The choice of having an inner diameter of 25 mm was based on the derived velocities for the high-pressure lines between ports A - A and B – B where the maximum flow velocity was calculated to 6.9 m/s.

The only part already fitted on the truck which has been identified to be modified or moved is the air tank assembly number 21944726 where one of the pipes clashes with the holding bracket off the assembly, see Appendix SS, here four possible solutions have been identified. 1) Move the assembly one hole row towards the left chassis side if possible, 2) develop a new bracket to attach the air bellow in the cross member, 3) relocate air bellow to another place on the truck or 4) completely remove the air tank from the truck, see Appendix SS. According to Mats Sanborn, this tank could be removed and the minimum air volume would still be exceeded.

In Appendix TT, the routing for RAPDD-GR is illustrated, without other components and parts. Note here that only the left side is completely routed, the right side is only routed to the bulkhead fittings mounted to the bracket on the chassis side.

8.5.2 RADT-GR

To the spindle 15 mm and 20 mm pipes are connected, leake/return flow and high pressure respectively. The pipes are just as the case for the RAPDD-GR solution connected with straight GEO15LM fittings and the 90 degree SAE fitting PAFG-90L. The pipes are then secured to clamps mounted to the bracket mounted onto the spindle which also the bulkhead fittings where the transition from pipes to hoses occur are held into place, see Appendix UU. The layout of the pipes in the picture and in the Creo Parametric model should only be used as an illustration of the path for the pipes from the spindle to the bulkhead fittings and not as an exact layout.

The bracket which hold the pipes and hoses at the wheel end is an assembly by one 5 mm thick sheet metal part and three 4 mm thick sheet metal parts which are to be welded together. Welding is seldomly of preferense within Volvo, but after a study visit at Volvo's own prototype workshop, a welded assembly was considered the only viable solution. For illustrating pictures see Appendix VV.

Both sides of the hoses are fitted with straight female fittings, 1CA46-18-8 and 1C977-20-10; these fittings are in turn fitted to bulkhead fittings SV18L and SV20S respectively. The hoses will clash with the mud flaps on different degree depending on steering angle, axle height and axle roll, see Appendix WW.

It is a desire by Volvo to have a 25^{o} degree turn angle on the tag axle in the future, which the C.H.D should be compatible with. After several virtual tests in Creo Parametrics a maximum steering angle of 15^{o} should not impose any problems, angles above 15^{o} needs real physical testing to verify if possible or not. Larger steering angle increases risk of exceeding the requirement of free distance from the wheel of 10 mm without snow chains and 60 mm with snow chains, in Appendix XX, the 60 mm free distance is indicated with a blue line.

Once the transit to pipes at the chassis sides has occurred the pipes are led to a connection block which branches the hydraulic flow. On the left side the pipes are held into place by four clamps which are mounted to a bracket which in turn is fastened to the chassis side, on the right side the pipes are routed directly to the connection block, see Appendix YY.

The connection block is mounted to the cross member behind the tag axle whereas the case for the RAPDD-GR the four 20 mm pipes are reduced to two 30 mm pipes, see Appendix RR. The pipes are from the connection block led along the right chassis side past all axles. Along this path the pipes are secured at three locations using two twin clamps stacked on each other see Appendix ZZ. Along this path some hoses and cables will be needed to be moved a shorter distance, something that for a prototype should be possible as long as the total length of the hoses or cables are not changed (Sanborn, 2016). In Appendix AAA and Appendix BBB cross section view on the routing along the chassis side are shown, where in the second picture the parts which are definitive to be needed of relocation or rework are indicated with green. The part numbers for the two market parts are:

- 21815804.asm
- 21666585.prt

Further the 30 mm pipes will have to have small local bends at two locations in order to get around existing parts, see Appendix CCC. The bends are not illustrated in the pictures and therefore are interfering with the existing parts, but the bends should not be a problem to design (Andersson, 2016).

8.6 Simulation of final concepts

In order to validate the strength of the developed brackets which will support the hydraulic routing both modal and static assessment has been performed on the more complex parts. The frequency which the truck is being subjected to during driving in normal conditions is by Volvo set to be from 0 Hz to 25 Hz, where a safety factor of 5 Hz should be applied, leading to a maximum frequency of 30 Hz. A component which natural frequency is within this area will start to oscillate which eventually will result in fatigue failure. All components on the truck need therefore to have a natural frequency higher than 30 Hz (Söder, 2016) and (Olsson, 2016).

When assessing the static load cases it is of value to know how large the acceleration is in the different directions; x, y, z, in order to perform an as accurate calculation as possible. In an internal Volvo report series of tests regarding the acceleration in the different directions have been documented. The acceleration increases with increased axle load, which means the worst case will be when the pusher and tag axle are loaded to specified maximum, 6720 kg and 7475 kg.

8.6.1 RAPDD-GR

As previously described, there are three brackets along the routed path from the wheel end to the connection block holding hoses and pipes regarding the RAPDD-GR solution. These brackets are also the parts that were deemed to be of interest to perform the simulations of.

8.6.1.1 Modal analysis

A208649_bracketauxspindlerapdd2, the bracket which is located directly onto the spindle is also the bracket which will only have the tires as dampers. When no load was applied, i.e. when fittings and hoses are not mounted, the lowest natural frequency is 197 Hz, vibrating in the driving direction. When applying an estimated load of 5.16 kg the natural frequency was measured to 41 Hz. The weight of the fittings were estimated using the mass property tool in Creo Parametric and the hose mass by multiplying the length with the known weight per meter, 0.8kg/m for hose 797TC and 0.52 kg/m for 462ST. Also the mass centre was projected a distance from the bracket in negative direction of travel due to masses of the hoses. As a safety factor the measured weights of the fittings were doubled and the entire hose lengths were used.

Bracket A208649_bracketchassirapdd2_1 which is mounted on the chassis side and the location where the hoses transcends to pipes has when applying the same load as for bracket A208649_bracketauxspindlerapdd2, a natural frequency of 32 Hz in lateral direction of the truck.

Bracket A208649_bracketcrossbeamrapddc2 which is located on the cross member shows a natural frequency of 26 Hz when the connection block with nipples is mounted. This is higher than the minimum of 25 Hz but lower than the safety mark of 30 Hz. Performed analysis did not however take the connected pipes into consideration, something that could change the result, which is why further evaluation of this bracket is of recommendation.

Pictures from the modal simulations of A208649-bracketchassirapdd2_1 and A208649_bracketcrossbeamrapdd2 can be seen in Appendix DDD and Appendix EEE.

8.6.1.2 Static analysis

The static analysis for the three brackets has been compiled in Table 20 with the maximum stress indicated in three directions; x, y and z.

Table 20 Static RAPDD-GR

Part	X - direction	Y - direction	Z - direction
A208649_bracketauxspindlerapdd2	250 MPa	137 MPa	882 MPa
A208649_bracketchassurapdd2_1	257 MPa	61 MPa	13 MPa
A208649_bracketcrossbeamrapddc2	41 MPa	60 MPa	10 MPa

The measured values are relatively low with the exception of the 882 MPa in Z – direction for A208649_bracketauxspindlerapdd2. If this is a true value this bracket will require high strength steel, but as previously stated, the values in Z – direction have been noted to be higher than reality and further test should be conducted before a final decision is made. Regarding the two brackets mounted onto the chassis side and the cross beam, a common structural steel should be sufficient. For illustrating picture see, Appendix FFF.

8.6.2 RADT-GR

As previously described, there are three brackets along the routed path from the wheel end to the connection block holding hoses and pipes regarding the RADT-GR solution. These brackets are also those parts that were deemed to be of interest to perform the simulations of.

The mass due to the fittings and hoses has been derived in the same way as in previous section 8.6.1.1, where the used mass in the calculations was 6.4 kg.

8.6.2.1 Modal analysis

The bracket assembly located at the wheel end, A208649_bracketspindleradt.asm, has a natural frequency of 38 Hz in the travel direction when fully loaded.

Bracket A208649_bracketchassiradt3_2.prt was measured to have a natural frequency of 30 Hz in the travel direction of the truck when fully loaded, see Appendix GGG. Note here that in both these cases the full length of the hoses as in previous section has been used in the simulations.

The bracket which is located onto the crossbeam, holding the connection block for the hydraulic system was measured to have a lowest natural frequency of 49 Hz in the travel direction of the truck.

8.6.2.2 Static analysis

The static analysis for the three brackets has been compiled in Table 21 with the maximum stress indicated in three directions; x, y and z.

Table 21 Static RADT-GR

Part	X - direction	Y - direction	Z - direction
A208649_bracketspindleradt.asm	583 MPa	263 MPa	373 MPa
A208649_bracketchassiradt3_2.prt	165 MPa	50 MPa	60 MPa
A208649_bracketcrossbeamradtcb.asm	49 MPa	33 MPa	11 MPa

As can be seen, the bracket assembly located at the wheel end is the only structure which receives relatively high stress, see Appendix HHH and Appendix III. But just as in the case for A208649_bracketauxspindlerapdd2 further assessments is recommended and for the two brackets mounted onto the chassis side and the cross beam, a common structural steel should also here be sufficient.

9.0 Final words

In this chapter the derived results and final concepts are discussed where requirements and the aims of the project are evaluated. Further, future recommendations within the subject are presented and finally the thesis is summarised in a conclusion.

9.1 Assessment of solution

The assessment of the results have been divided into four separate parts; Motors, Schematics, Hoses and fittings, and Packaging, the four discussed topics in this report. These four areas are here discussed with regard to the stated questions in section 1.3.

9.1.1 Motors

From the conducted simulations regarding the motors, it has firstly been theoretically proven that it is in fact possible to achieve a velocity of 50 km/h if a hydraulic gear box of two motor pairs is used, see section 5.2. The 50 km/h can be achieved simultaneously as the total produced torque, if the entire system is engaged, is increased, which will automatically result in higher startability.

The requirement of starting in a slope with the inclination 12 % on dry asphalt has been shown to be fulfilled even without engaging the C.H.FWD or C.H.D. However it has shown that when driving on slippery surface the startability from the hydraulic motors are offering a significant contribution.

If the highest startability is desired to be achieved, a motor size of approximately 413 cc should be chosen. If a columetric displacement of 413 cc is chosen the top velocity when only two motors are engaged would be approximately 41 km/h, 9 km/h short from the 50 km/h goal. If the 50 km/h mark is to be passed a motor size of approximately 367 cc should be chosen, this would however result in a reduction of total available torque of 9264 Nm to 8235 Nm if compared to the 413 cc motor.

9.1.2 Schematics

Regarding the valve block, three concepts were developed, where concept 1 and concept 3 were said to be of further interest. Concept 3 would be a preferred choice for a first prototype to test the system at a low cost. Concept 1 is a more sustainable long term solution which would be of preference more towards later prototypes and production.

9.1.3 Hoses and fittings

Depending on how the final solution is configured, the same dimensions on hoses and pipes as FH-1825 can be used. The reason why it can be of advantage to have the same dimensions on hoses and pipes as a previous test truck is that knowledge regarding required space and behaviour of the system is then known to a higher degree compared to other dimensions.

If the three requirements and the desire in section 7.1 are all fulfilled, the system can have the same dimensions of hoses to the wheel ends as the FH-1825 truck since the maximum flow will in that case be equal in some situations. If instead only the requirements are fulfilled, the maximum flow in the motor pair mounted on the pusher/tag axle will be reduced and therefore also the hoses and pipes in this area can be reduced.

Further the choice of having large pipes with an outer diameter of 30 millimetres for the high pressure lines when routing along the chassis was taken, which is not what is used today. But making this decision resulted in a significant advantage in packaging of the system.

9.1.4 Packaging

The final concepts for RAPDD-GR and RADT-GR have both been verified to work by Michael Andersson and should be possible to realise.

The solution of the routing for RAPDD-GR, was possible to perform with seemingly only one part that needs to be modified or relocated, the air tank mounted to the cross member. In general, available space to route along the chassis for RAPDD-GR is relatively high where the only possible problem is occupied space from the propeller shaft. But as presented in 8.5.1 does this seem to be manageable and it should be possible to rout as illustrated. The routing at the wheel end is also an area where available space is low, but the shape of the used bracket holding the pipes and hoses should still enable possibility to easily attach and detach the fittings.

The suggested routing for the RADT-GR combination imposes higher degree of difficulties where several existing parts will be needed to be moved or modified compared to the RAPDD-GR solution. The clash between hoses and mud flaps does impose quite large complications, the tool used to develop this part costs about 1,5-2,0 million SEK and is classified as an A-surface by the design department (Wiktorsson, 2016).

9.2 Future work and recommendations

From the conducted pre study of implementing C.H.D on pusher and tag axle it is recommended that the future work should firstly be performed for the RAPDD-GR. This is mainly due to the fact that the routing does seem to be easier to implement on the RAPDD-GR combination, where only one part will be needed to be modified. Taking this pre-study to the next phase where a study and implementation of the entire hydraulic system is performed could be a good step.

For the RADT-GR large portion of the mud flaps would be needed to be removed and also parts along the chassis side, where the pipes would have several bends along the routing path. Also the RADT-GR solution requires a tie rod from the RADDT-G2 combination, which would result in an inaccurate Ackerman angle. Achieving accurate Ackerman angle with the C.H.D installed could be the next research area specific for the RADT-GR configuration

Neither the RAPDD-GR combination nor the RADT-GR combination offers easy routing possibilities where a perfect solution does seem to be hard or even impossible to achieve. Developing new wheel end configurations where the hydraulic drive has been considered from the beginning is therefore recommended. Having the high pressure lines through the king pin and a custom made swivel fitting as Mercedes has allowed them to have a compact system where the high pressure lines should not be subjected to any twisting motion. Utilizing the hydraulic dampers and developing a new damper which also can transport high pressure oil flow can also be an area of further research.

For the developed routing of hoses to the wheel end, all hoses will most likely be subjected to twisting during their lifetime. It could therefore be of interest to test whether swivel fittings would work or if these fittings would as suspected by Jörgen Ahlberg start leaking relatively fast. Also quick couplings could be of interest of further investigations, but high negativity towards these types of fittings with the high pressure and application has been noted.

9.3 Conclusion

At least in theory does having two motors on the front axle and two motors on the pusher/tag axle resulted in both higher possible maximum velocity and also a good startability. The RAPDD-GR does indicate less difficulties of implementation of the system compared to the RADT-GR, but both are possible candidates for the C.H.D.

If the requirement list in section 4.0 is reviewed it can be seen that the majority of the requirements are fulfilled where only two have not been fulfilled and seven are still unknown and cannot be answered. The two requirements which have not been met are maximum rolling resistance and torque on pump. To manage a rolling resistance of 40 % does seem to require high torque and traction which the hydraulic motors are far away from achieving. Regarding allowed torque on pump, which was specified to maximum 572 Nm, calculations indicate a highest torque of 595 Nm. Even though the hydraulics supplier most probably have set a lower value than the pump actually can be used for and the 595 Nm is the highest peak value, it is still higher than set requirement and therefore is unachieved.

The requirements which are unknown whether possible to be met today are all except one requirements which the hydraulics supplier has to conduct further tests on. The one that does not regard testing by the hydraulics supplier is the free distance from wheel by hoses when the wheels are fitted with snow chains. Virtual tests does indicated that a 15º steering angle could work, and higher angles would most possible not work. However, since the virtual environment is not exactly as reality it should be used as such and physical test should be conducted to answer this requirement fully.

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Appendix A MAN – routing

Source:http://www.at-aandrijftechniek.nl/technologie/wielaandrijvingen-voor-betere-tractie-of-betere-efficiency/3094/ 2016-05-12

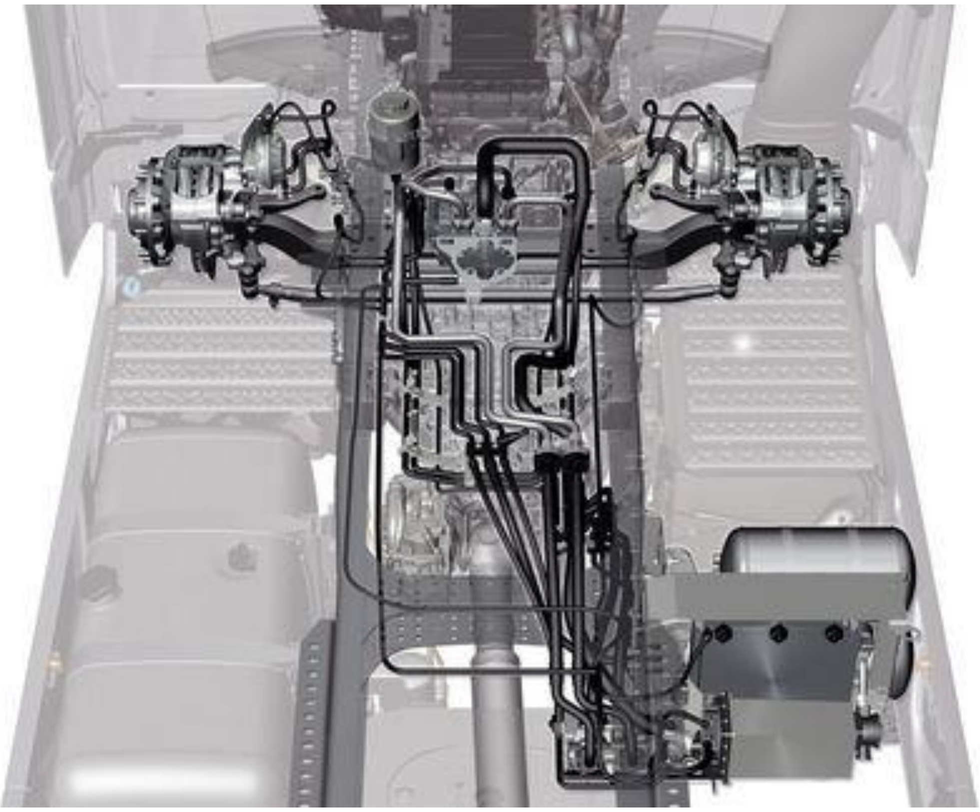
The high pressure pipe lines are attached to the spindle with sae flanges and the low pressure lines with crimp able 90^o elbow fittings. All lines are then together directed upwards above the brake clock. In order to know how the hoses are connected to the chassis frame side more pictures are needed.

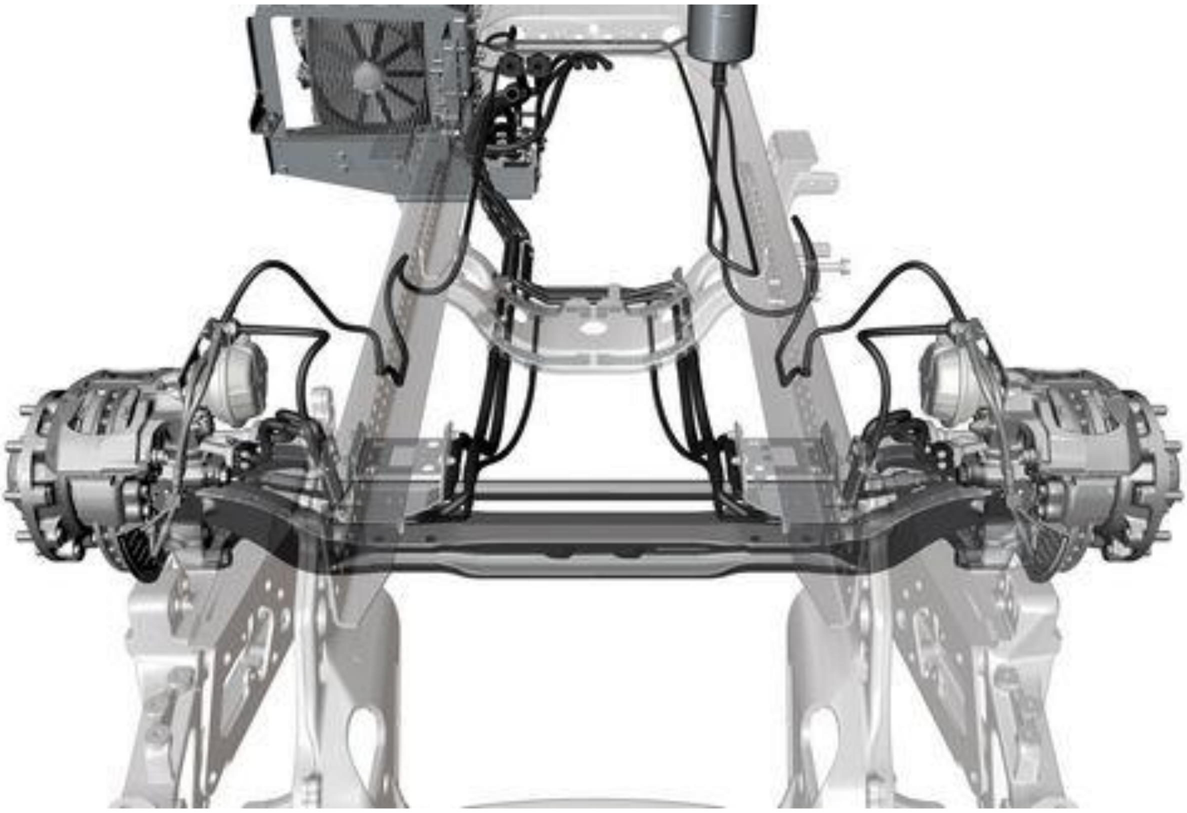
Appendix B Optitrack - routing

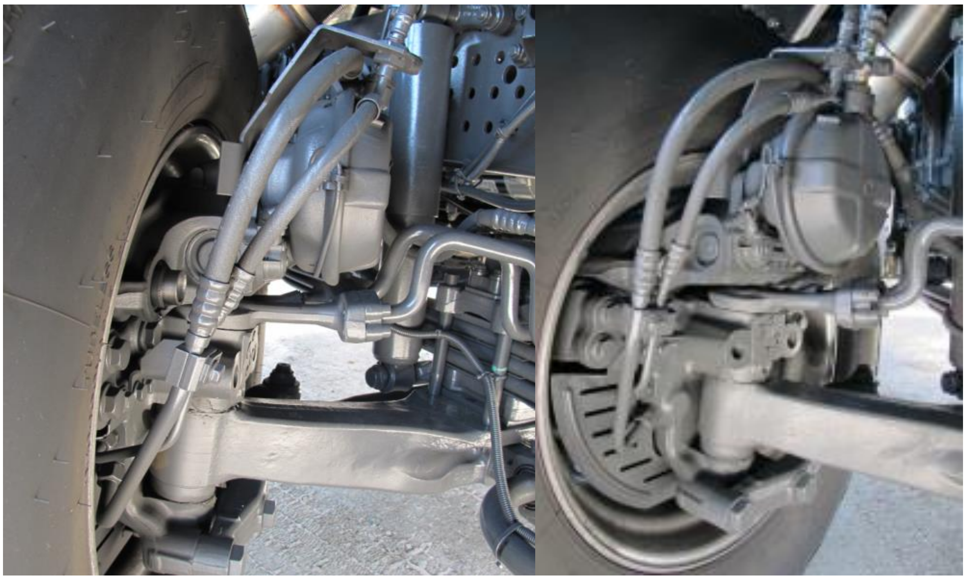
Source: *http://corporate.renault-trucks.com/en/press-releases/2014-11-20-optitrack-on-the-renault-trucks-c-optimised-mobility-and-consumption.html 2016-05-12*

The two high pressure lines are connected to the wheel spindle with SAE flanges and pipes, these pipes ascends vertically and bent backwards with a 90^{o} *angle, along the ascending path the pipes are supported by two clamps which are attached to a bracket mounted to the spindle. The low pressure lines are connected to the spindle with* 90^{o} *crimp elbow fittings; all lines are then bundled together and led to the chassis frame side through a hole in the mud flaps. This solution does most likely result in twisting in the hoses and will eventually lead to fatigue failure.*

Appendix C HAD - routing







Source: *http://blog.mercedes-benz-passion.com/2015/05/allrad-auf-knopfdruck-der-mercedes-benz-arocs-mit-zusatzantrieb-hydraulic-auxiliary-drive-had/* *2016-05-12*

The two high pressure lines are connected to the steering spindle and motors via a swivel connection located at the top of kingpin. From the swivel two pipes are then led via the leaf feather to a cross member and then the tank. The two low pressure lines are both routed to be as in line with the rotational axle as possible where the two are clamped and lead in different directions. One perpendicular to the chassis frame side to follow the high pressure lines and the other backwards some distance to later be led under the chassis frame side. The routing from wheel to chassis frame side is done equally on both sides. By routing the high pressure though a swivel and then to the chassis frame side, wear on the hoses is significantly reduced, however swivels are much more dirt sensitive and there might be risk of leakage in the fitting.

Appendix D Friction

Type of surface	Coefficient of Traction
Clay, wet Ice Snow road solid Dirt road, deeply rutted, loose Sand or gravel, loose Stripped arable land, sticky wet	FRIC-0.1 (μ ≤ 0,3)
Clay, dry Moraine, dry Soil backfill, soft Wet asphalt road Paved road (Belgian pavé), dry Dirt road, compacted Dirt road, firm rutted Sand or gravel, compactible Soil road, packed Stripped arable land, loose, dry	FRIC-0.4 (μ = 0,4-0,5)
Gravel road, compacted Macadam Woodland pastures, grassy banks Stripped arable land, firm, dry	FRIC-0.6 (μ = 0,6-0,7)
Asphalt, dry Concrete, dry	FRIC-0.8 (μ ≥ 0,8)

Source: Internal Volvo GTA parameters

Appendix E Rolling resistance

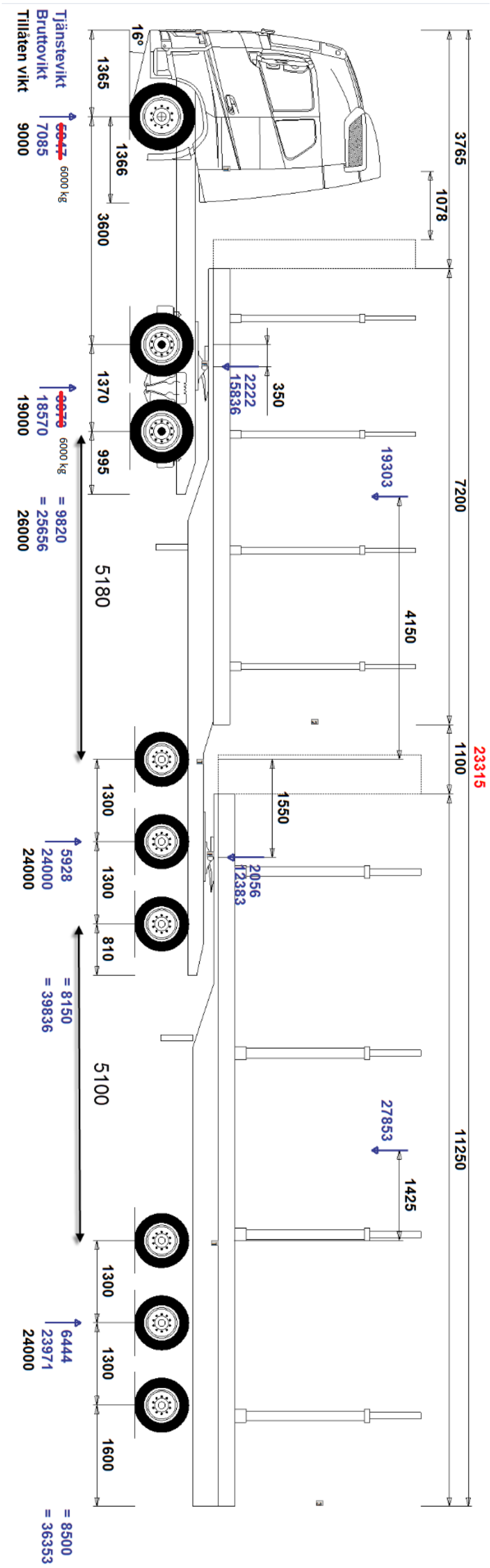
Type of surface	ROLLING RESISTANCE
Ice Asphalt Paved road (Belgian pavé) Concrete	ROLR-2 (0,4-2%)
Gravel road, compacted Dirt road, compacted Dirt road, firm rutted Macadam Soil road packed Snow 100 mm	ROLR-5 (3-5%)
Soil backfill, soft Stripped arable land, firm, dry Stripped arable land, loose, dry	ROLR-12 (6-12%)
Stripped arable land, sticky wet Woodland pastures, grassy banks Pasture soil grass Sand or gravel, compactible Dirt road, deeply rutted, loose	ROLR-25 (13-25%)
Clay, loose Moraine, loose Sand, dry and loose	ROLR-40 (26-40%)

Source: Internal Volvo GTA parameters

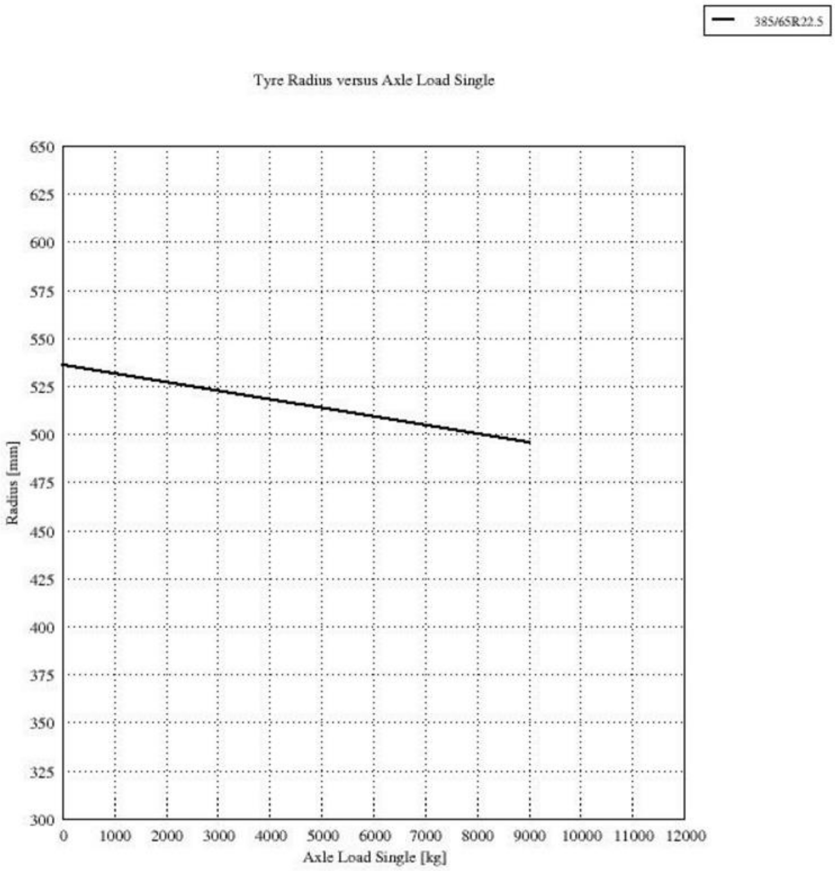
Appendix F Optimum PTO ratio

```
np = 3050 % max allowed rpm for pump
ne = 2000 % max possible rpm when accelerating
ipto = np/ne % meaning the rotational speed for the pump will be ipto times
larger than rotational speed for the engine
```

Appendix G Weight distribution FH-1825



Appendix H Tyre radius

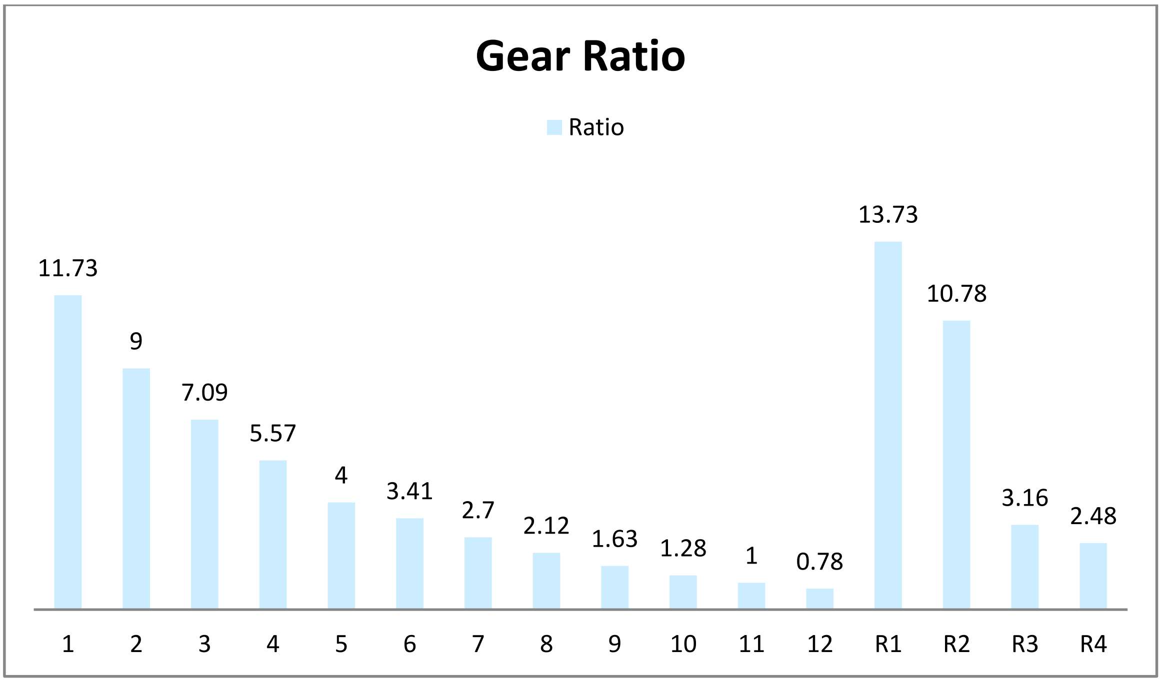


Source: Volvo EDB

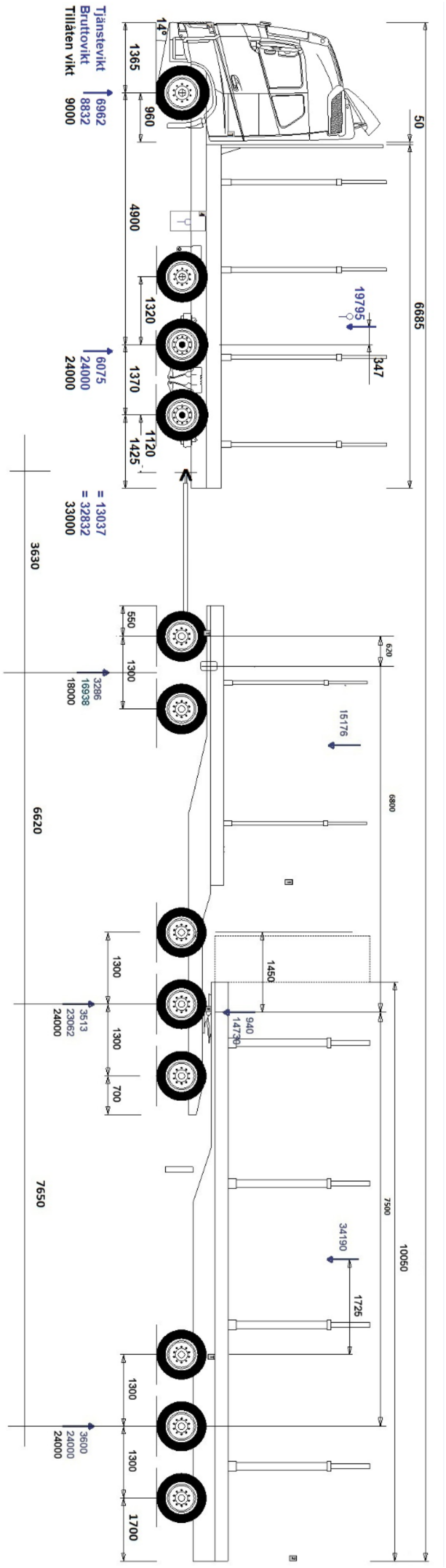
Appendix I Rotational ratio for pump with FH-1825 spec.

```
npmax = 2000*1.26;
Rotrat = npmax/3050 % The rotational ratio for the pump where Max allowed
rotation for pump is 3050 rp
```

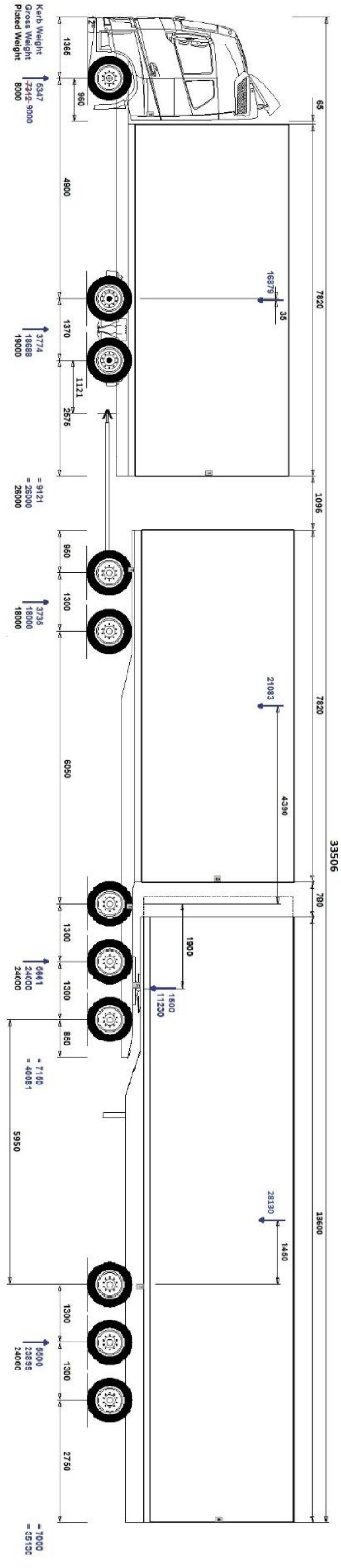
Appendix J Gear ratio



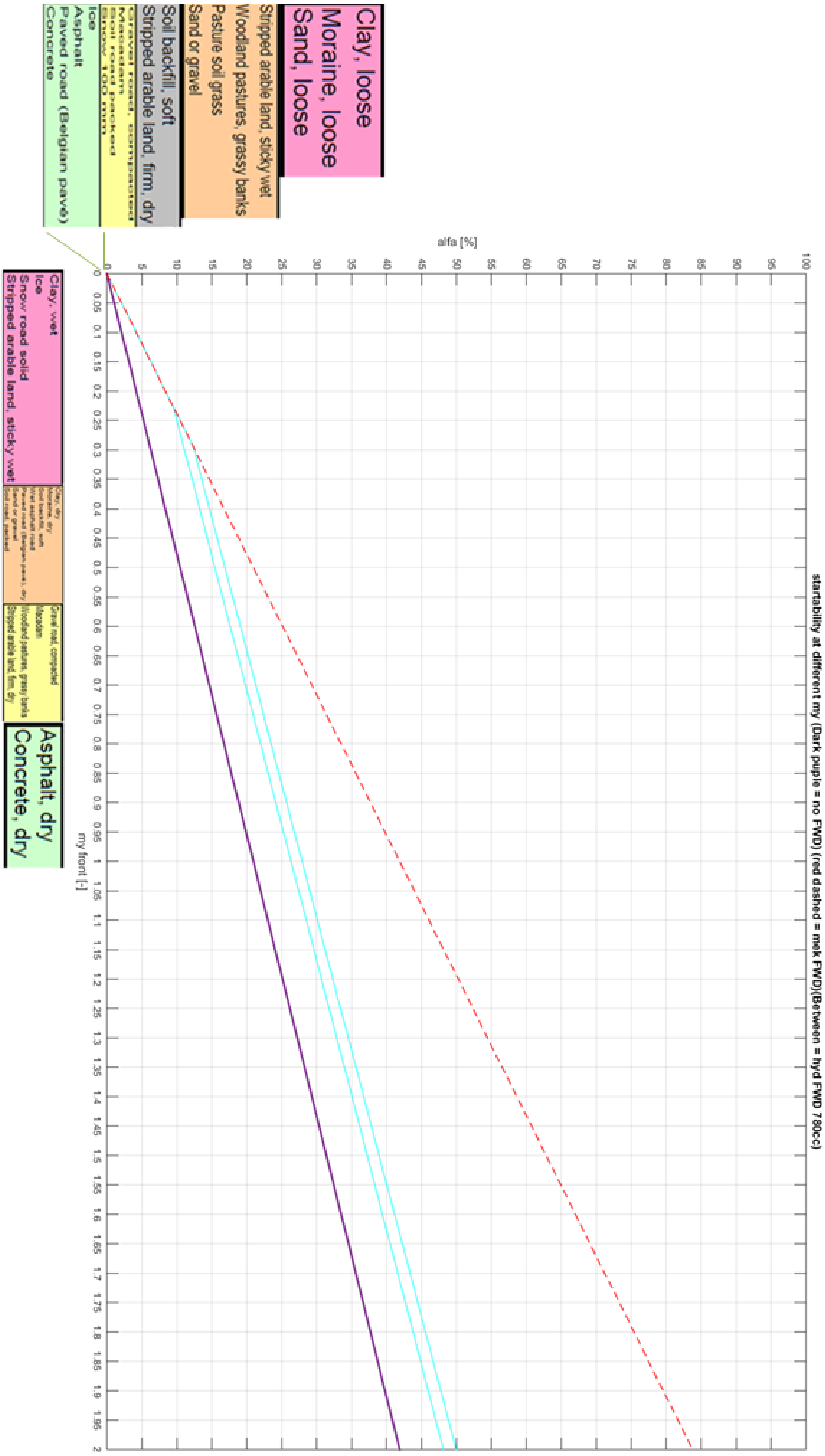
Appendix K Weight distribution RAPDD-GR



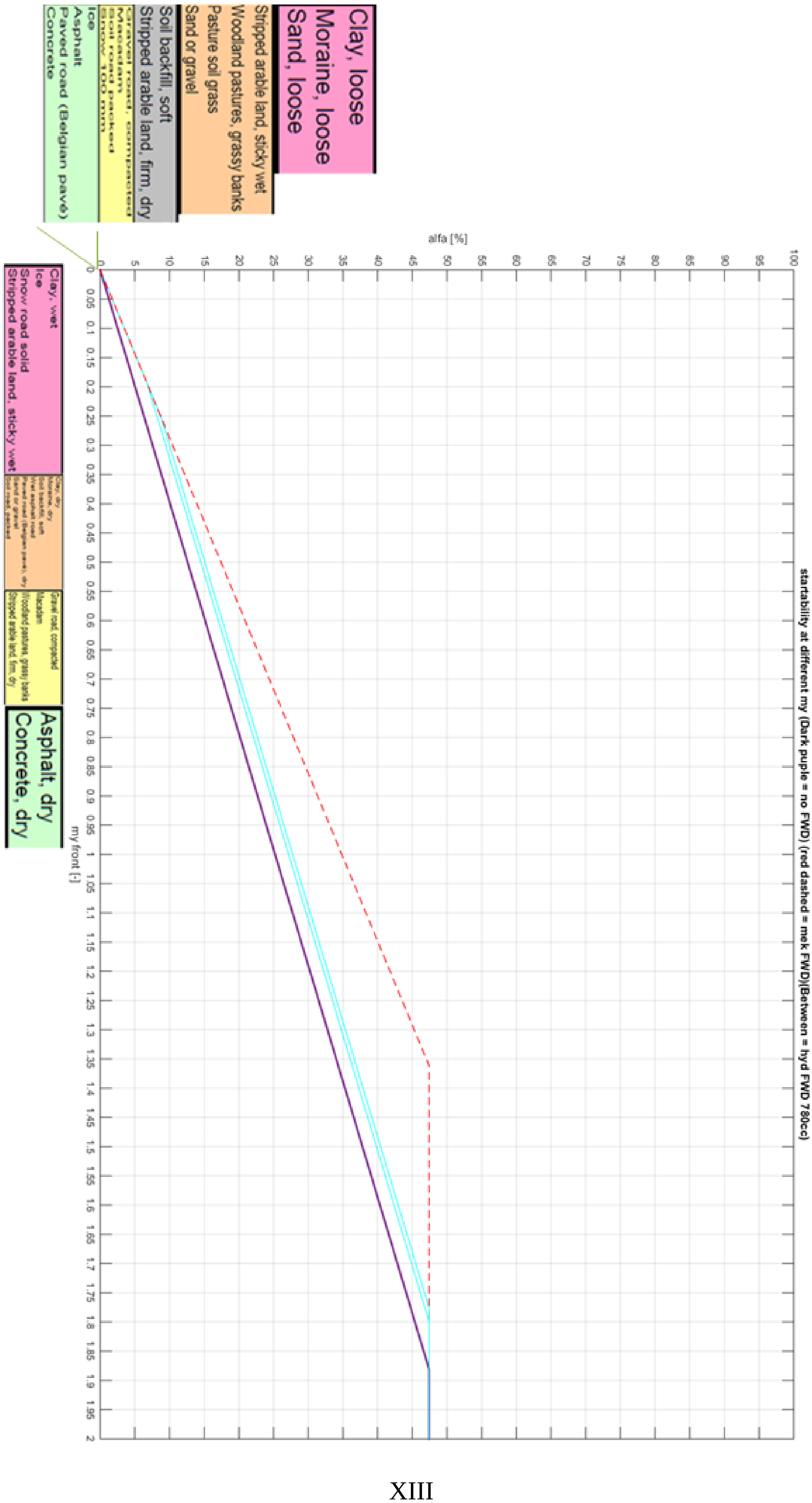
Appendix L Weight distribution RADT-GR



Appendix M FH-1825 unloaded



Appendix N Fh-1825 loaded



Appendix O Calculations for FH-1825 velocity and torque

```
clear all
Dp = 90; % displacment for pump in cm^3 (90cc)
Dm = 780;
ipto = 1.26; % ratio for PTO calculated as wout/win= ipto...
R  =  (((385*0.65*2)+(22.5*25.4))/1000)/2;  %  wheel  radius  in  meters  for
unloaded non-driven wheel
R2  =  (((525-500)/(2500-8000))*7085+536)/1000;  %  Radius  front  wheel  fully
loaded 7500 kg, curve equation approximated from figure 1
nvolp  =  xx;  %  The  efficinecy  when  having  max  rotational  speed  3050  and
delta p = 400
% source the hydraulics supplier data sheet
% engine rotatinal speed will be 3050/1.26 = 2420.63 if max allowed pump
% speed is used. But max engine rev. is 2000 rpm when accelerating
% span which this can be used is thus 0-2000 rpm
ne=2000;
nvolm  =  0.97;  %  approximated  volymetric  efficiency  for  wheel  motors  based
from data sheet provided by the hydraulics supplier
deltap = 400; % Bar

nhmp = xx; % The mechanical efficinecy when having max rotatinal speed 3050
and delta p = 400
Tpto = (Dp*deltap)/(20*pi*nhmp); % [Nm]

vunloaded = ((Dp*ne*nvolp*nvolm*R*2*pi*ipto)/(2*Dm*60))*3.6
vloaded = ((Dp*ne*nvolp*nvolm*R2*2*pi*ipto)/(2*Dm*60))*3.6

nm = linspace(0,25,10); % Rotational speed from 0-25 RPM, low revs used for
start, has the highest influence on efficiency
nhm  =  ((0.88-0.68)/(25-0))*nm+0.68;  %  Efficiency  curve  approximated  to  a
line based of data sheet

ii=0;

for ii=0:9
    ii=ii+1;
T(ii) = (deltap*Dm*nhm(ii))/(20*pi); % Torque for one motor

end
T=T*2; % Total torque produced by the two motors
ig=  [11.73  9.21  7.09  5.57  4.35  3.41  2.7  2.12  1.63  1.28  1  0.78];  %  gear
ratios for gear 1-12 in that order
ifinal = 3.61; % Source RAT3.61 EDB
itot=ig*ifinal;
jj=0;

for jj=0:11
    jj=jj+1;
X(jj) = ((Dp*nvolp*ipto*itot(jj)*nvolm)./(2*Dm)-1)*100;
end
%plot(T,nhm)
X=X';
```

Appendix P Calculations for FH-1825 friction and inclination loaded

```
clc
clear all
i=0;
j=0;
Thyd=[6753.262545 8739.516];
colour = ['c'];
for i=0:1
    i=i+1;
    for j=0:1
        j=j+1;
Tice=3550; %torque from combustion engine
ratio=11.73*3.61; %total ratio on first gear
mice=18570; %total axel load on the rear driven wheels. 36% weight on rear
axles of total 24 tons *2 axles
mhyd=7085; %axel load on front wheel
mtot=73656; %total weight for the trailen and truck
radii=(((525-500)/(2500-8000))*mhyd+536)/1000;%rolling radius on wheels.
based on graf from EDB at mhydfront
radiiice=(((525-500)/(2500-8000))*mice+536)/1000;%rolling radius on wheels.
based on graf from EDB at mice
g=9.81; %grofity constant
e=0.93; % efficency on ICE drivetrain
% 93 % verkningsgrad verifierad of Anders Hedman
Fice=((Tice*e*ratio)/radiiice); % Driving force produced by engine
amax=asin((Fice)/(mtot*g)); % max slope angle
myice=Fice/(mice*g*cos(amax)); % Limiting friction
Fhyd=(Thyd(1,i)/radii); % Driving force produced by hydraul motors
amaxhyd = asin((Fhyd)/(mtot*g));
myhyd1=(Fhyd)/(mhyd*g*cos(amaxhyd)); % Limiting friction
a2=asin(((mice+mhyd)*myhyd1*g*cos(amaxhyd))/(mtot*g)); % Max slope angle
with C.H.D where the wheels spin
my3=(Fice-Fhyd)/(mice*g*cos(amax));
mymek=(Fice)/((mice+mhyd)*g*cos(amax)); % Limiting friction mechanical
drive
y1=[0 tan(amax)*100 tan(amax)*100];
x1=[0 myice 2];
plot(x1,y1,'linewidth',2)
xlabel('my front [-]')
ylabel('alfa [%]')
xlim([0 2])
ylim([0 100])
grid on
set(gca,'XTick',0:0.05:2)
set(gca,'YTick',0:5:100)
title('startability at different my (Dark puple = no FWD) (red dashed = mek
FWD)(Between = hyd FWD 780cc)')
hold on
y2=[0 tan(a2)*100 tan(amax)*100 tan(amax)*100];
x2=[0 myhyd1 my3 2];
plot(x2,y2,colour(1),'linewidth',0.5)
y3=[0 tan(amax)*100 tan(amax)*100];
x3=[0 mymek my3];
plot(x3,y3,'r--','linewidth',1)
    end
end
```

Appendix Q Calculations for FH-1825 friction and inclination unloaded

```
clc
clear all
i=0;
j=0;
Thyd=[6753.262545 8739.516];
colour = ['c'];
for i=0:1
    i=i+1;
    for j=0:1
        j=j+1;
Tice=3550; %torque from combustion engine
ratio=11.73*3.61; %total ratio on first gear
mice=6000; %total axel load on the rear driven wheels. 36% weight on rear
axles of total 24 tons *2 axles but rear lifted, lift and declutch
mhyd=6000; %axel load on front wheel
mtot=28650; %total weight for the trailen and truck 12000+8150+8500
radii=(((525-500)/(2500-8000))*mhyd+536)/1000;%rolling radius on wheels.
based on graf from EDB at mhyd
radiiice=(((525-500)/(2500-8000))*mice+536)/1000;%rolling radius on wheels.
based on graf from EDB at mice
g=9.81; %grofity constant
e=0.93; % efficency on ICE drivetrain
% 93 % verkningsgrad verifierad of Anders Hedman
Fice=((Tice*e*ratio)/radiiice); % Driving force produced by engine
amax=asin((Fice)/(mtot*g)); % max slope angle
myice=Fice/(mice*g*cos(amax)); % Limiting friction
Fhyd=(Thyd(1,i)/radii); % Driving force produced by hydraul motors
amaxhyd = asin((Fhyd)/(mtot*g));
myhyd1=(Fhyd)/(mhyd*g*cos(amaxhyd)); % Limiting friction
a2=asin(((mice+mhyd)*myhyd1*g*cos(amaxhyd))/(mtot*g)); % Max slope angle
with C.H.D where the wheels spin
my3=(Fice-Fhyd)/(mice*g*cos(amax));
mymek=(Fice)/((mice+mhyd)*g*cos(amax)); % Limiting friction mechanical
drive
y1=[0 tan(amax)*100 tan(amax)*100];
x1=[0 myice 2];
plot(x1,y1,'linewidth',2)
xlabel('my front [-]')
ylabel('alfa [%]')
xlim([0 2])
ylim([0 100])
grid on
set(gca,'XTick',0:0.05:2)
set(gca,'YTick',0:5:100)
title('startability at different my (Dark puple = no FWD) (red dashed = mek
FWD)(Between = hyd FWD 780cc)')
hold on
y2=[0 tan(a2)*100 tan(amax)*100 tan(amax)*100];
x2=[0 myhyd1 my3 2];
plot(x2,y2,colour(1),'linewidth',0.5)
y3=[0 tan(amax)*100 tan(amax)*100];
x3=[0 mymek my3];
plot(x3,y3,'r--','linewidth',1)
    end
end
```

Appendix R Calculations of motor size and speed relation to mechanical drive engaging two motors

```
clear all
Dp = 90; % displacment for pump in cm^3 (90cc)
ipto = 1.26; % ratio for PTO calculated as wout/win= ipto...
R = (((385*0.65*2)+(22.5*25.4))/1000)/2; % wheel radius in meters for both
front and rear
v = 50/3.6; % max desired speed in m/s

nvolp = xx; % The efficinecy when having max rotational speed 3050 and
delta p = 400
% source the hydraulics supplier data sheet
% engine rotatinal speed will be 3050/1.26 = 2420.63 if max allowed pump
% speed is used. But max engine rev. is 2000 rpm when accelerating
% span which this can be used is thus 0-2000 rpm
ne = linspace(0,2000,10); % Angular speed for engine from 0-2000 in 1x10
matrix
nvolm = 0.97; % approximated volymetric efficiency for wheel motors based
from data sheet provided by the hydraulics supplier

Dm = ((Dp*ne*nvolp*nvolm*R*2*pi*ipto)/(2*v*60)) % Different wheel motors
displacements in cm^3 with set velocity and variable engine speed divided
by 2 since the high speed should be achieved with 2 motors and high torque
with 4

% Will PTO mange the torque? Max allowed torque on PTO = 1000 N
nhmp = xx; % The mechanical efficinecy when having max rotatinal speed 3050
and delta p = 400
deltap = 400; %deltap = p-lp where p = 425 bar and lp = {low pressure} = 20
bar
Tpto = (Dp*deltap)/(20*pi*nhmp); % [Nm]

ig= [11.73 9.21 7.09 5.57 4.35 3.41 2.7 2.12 1.63 1.28 1 0.78]; % gear
ratios for gear 1-12 in that order
ifinal = 3.61; % Source RAT3.61 EDB
itot=ig*ifinal;
i=0;
j=0;
X=zeros(12,10);
for i=0:11
    i=i+1;
    for j=0:9
        j=j+1;
X(i,j) = (((Dp*nvolp*ipto*itot(i)*nvolm)./(2*Dm(j)))-1)*100
vtruck(i,j) = (ne(j)*2*pi*R*60)/(itot(i)*1000)
    end
end
```

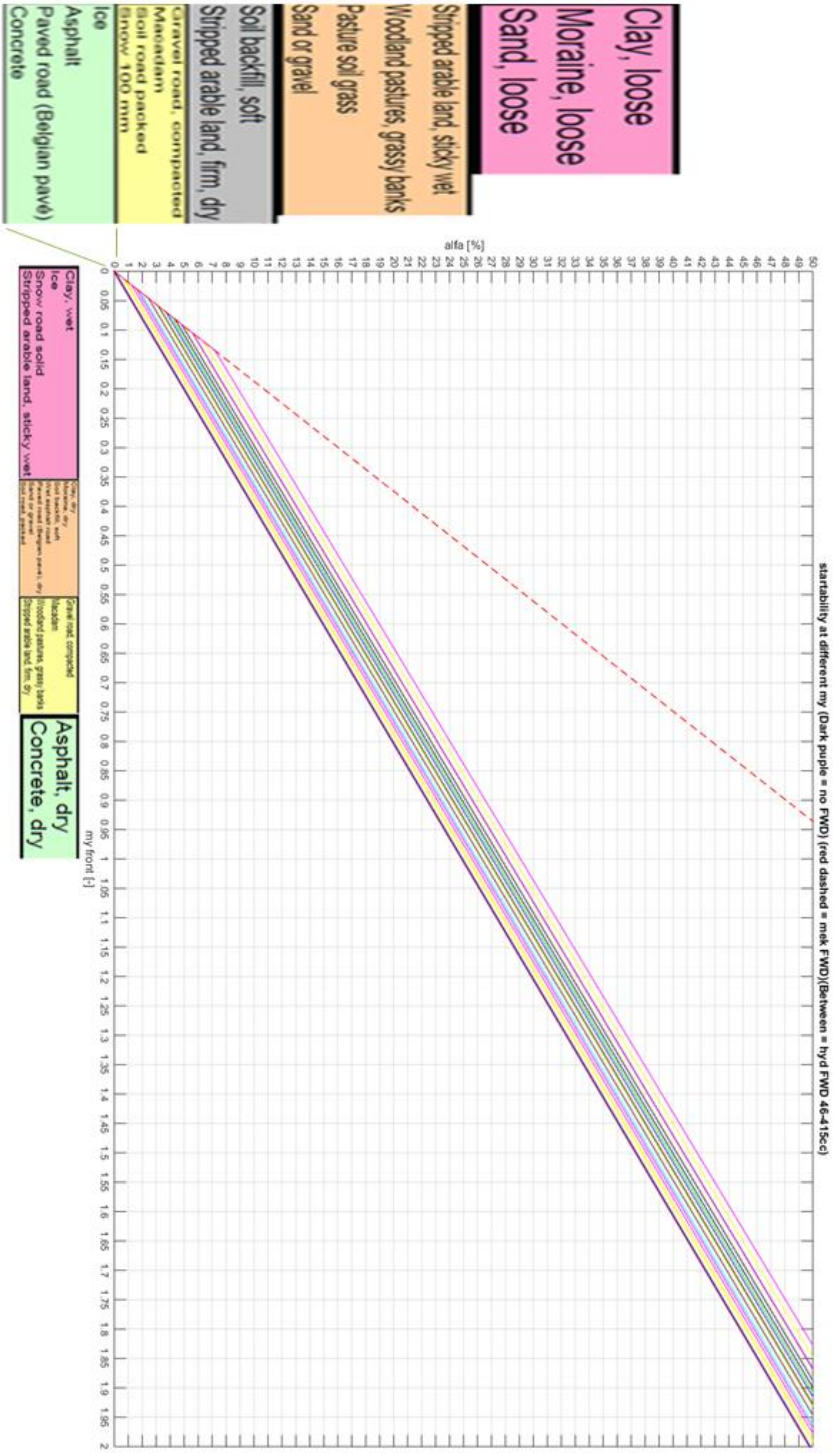
Appendix S Torque four motors

```
clear all
Dp = 90; % displacment for pump in cm^3 (90cc)
ipto = 1.26; % ratio for PTO calculated as wout/win= ipto...
R = (((385*0.65*2)+(22.5*25.4))/1000)/2; % wheel radius in meters for both
front and rear
v = 50/3.6; % max desired speed in m/s
nvolp = xx; % The efficinecy when having max rotational speed 3050 and
delta p = 400
% source the hydraulics supplier data sheet
% engine rotatinal speed will be 3050/1.26 = 2420.63 if max allowed pump
% speed is used. But max engine rev. is 2000 rpm when accelerating
% span which this can be used is thus 0-2000 rpm
ne = linspace(0,2000,10); % Angular speed for engine from 0-2000 in 1x10
matrix
nvolm = 0.97; % approximated volymetric efficiency for wheel motors based
from data sheet provided by the hydraulics supplier

Dm = ((Dp*ne*nvolp*nvolm*R*2*pi*ipto)/(2*v*60)); % Different wheel motors
displacements in cm^3 with set velocity and variable engine speed
deltap = 400; %deltap = p-lp where p = 425 bar and lp = {low pressure} = 20
bar

n = (v*60)/(R*2*pi); % required roational speed of motors for given v and R
nm = linspace(0,25,10); % Rotational speed from 0-25 RPM, low revs used for
start, has the highest influence on efficiency
nhm = ((0.88-0.68)/(25-0))*nm+0.68; % Efficiency curve approximated to a
line based of data sheet
ii=0; jj=0;
for ii=0:9
    ii=ii+1;
    for jj=0:9
        jj=jj+1;
T(ii,jj) = (deltap*Dm(jj)*nhm(ii))/(20*pi); % Torque for one motor
    end
end
T2=T*2; % Torque using 2 motors
T4=T*4; % Torque using 4 motors

plot(T2,nhm)
hold on
plot(T4,nhm)
```

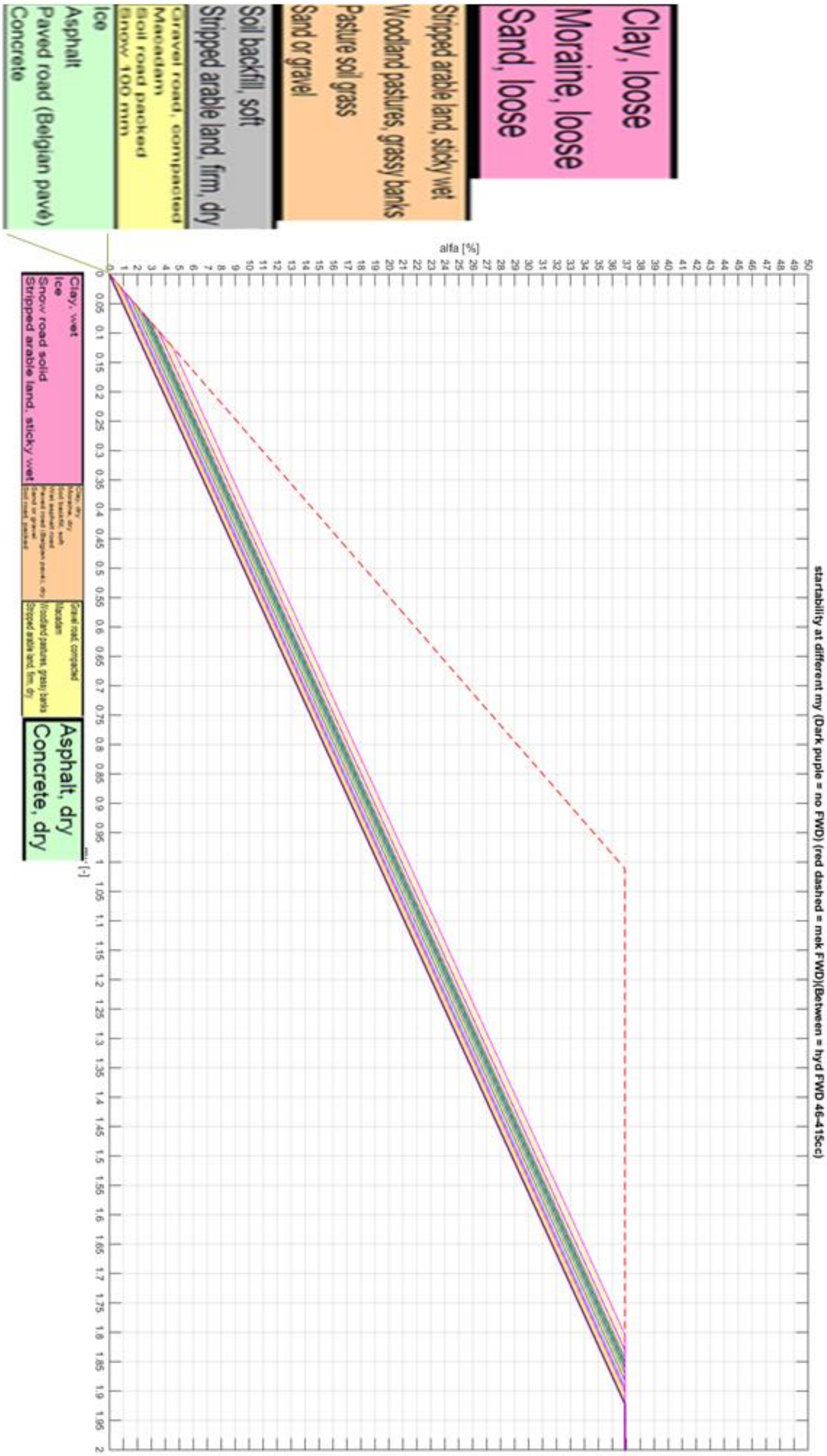
Appendix T Calculations of motor size and speed relation to mechanical drive engaging four motors

```
clear all
Dp = 90; % displacment for pump in cm^3 (90cc)
ipto = 1.26; % ratio for PTO calculated as wout/win= ipto...
R = (((385*0.65*2)+(22.5*25.4))/1000)/2; % wheel radius in meters for both
front and rear
v = 50/3.6; % max desired speed in m/s
nvolp = xx; % The efficinecy when having max rotational speed 3050 and
delta p = 400
% source the hydraulics supplier dta sheet
% engine rotatinal speed will be 3050/1.26 = 2420.63 if max allowed pump
% speed is used. But max engine rev. is 2000 rpm when accelerating
% span which this can be used is thus 0-2000 rpm
ne = linspace(0,2000,10); % Angular speed for engine from 0-2000 in 1x10
matrix
nvolm = 0.97; % approximated volymetric efficiency for wheel motors based
from data sheet provided by the hydraulics supplier

Dm = ((Dp*ne*nvolp*nvolm*R*2*pi*ipto)/(2*v*60)) % Different wheel motors
displacements in cm^3 with set velocity and variable engine speed divided
by 2 since the high speed should be achieved with 2 motors and high torque
with 4

% Will PTO mange the torque? Max allowed torque on PTO = 1000 N
nhmp = xx; % The mechanical efficinecy when having max rotatinal speed 3050
and delta p = 400
deltap = 400; %deltap = p-lp where p = 425 bar and lp = {low pressure} = 20
bar

Tpto = (Dp*deltap)/(20*pi*nhmp); % [Nm]
ig= [11.73 9.21 7.09 5.57 4.35 3.41 2.7 2.12 1.63 1.28 1 0.78]; % gear
ratios for gear 1-12 in that order
ifinal = 3.61; % Source RAT3.61 EDB
itot=ig*ifinal;
i=0;
j=0;
X=zeros(12,10);
for i=0:11
    i=i+1;
    for j=0:9
        j=j+1;
X(i,j) = (((Dp*nvolp*ipto*itot(i)*nvolm)./(4*Dm(j)))-1)*100
vtruck(i,j) = (ne(j)*2*pi*R*60)/(itot(i)*1000)
    end

end
```

Appendix U RAPDD-GR unloaded four motors



Appendix V RAPDD-GR loaded four motors



Appendix W Calculations for RAPDD-GR friction and inclination unloaded

```
clc
clear all
i=0;
j=0;
% Calculating weight distribution
A = 6962*9.81;
B = 0*9.81;
C = 6075*9.81;
D = 13037*9.81;
p=0;
yu = (D*2283.31-B*p*3580)/(C*5585);
xu = (D-C*yu-B*p)/A;
%--------------------------------
Thyd=[795.4266894       1590.853379   2386.280068   3181.706758   3977.133447
4772.560137 5567.986826 6363.413515 7158.840205
1029.375716  2058.751431  3088.127147  4117.502863  5146.878579  6176.254294
7205.63001  8235.005726 9264.381442];
Thyd=Thyd/2; % only two motors are engaged
colour = ['y' 'm' 'c' 'r' 'g' 'b' 'k' 'y' 'm'];
for i=0:8
    i=i+1;
    for j=0:1
        j=j+1;
Tice=2000; %torque from combustion engine
ratio=11.73*3.61; %total ratio on first gear
mice=6075; %total axel load on the rear driven wheels.
mhydfront = 6962; % From picture weight distribution RAPPD
mhydrear = 0*0.28;
mhyd = mhydfront+mhydrear;
mtot=24400; %total weight for the trailen and truck
radii=(((525-500)/(2500-8000))*mhydfront+536)/1000;  %rolling  radius  on
wheels. based on graf from EDB at mhyd
radiiice=(((525-500)/(2500-8000))*mice+536)/1000;%rolling radius on wheels.
based on graf from EDB at mice
g=9.81; %grofity constant
e=0.93; % efficency on ICE drivetrain
% 93 % verkningsgrad verifierad of Anders Hedman
rr=55*mtot/1000; %rolling resistance on asphalt (N/ton*ton)/1000???
Fice=((Tice*e*ratio)/radiiice); % Driving force produced by engine
amax=asin((Fice)/(mtot*g)); % max slope angle
myice=Fice/(mice*g*cos(amax)); % Limiting friction
%_________________________________________________________________

Fhyd=(Thyd(j,i)/radii); % Driving force produced by hydraul motors
amaxhyd = asin((Fhyd)/(mtot*g));
myhyd1=(Fhyd)/(mhyd*g*cos(amaxhyd));  %  Limiting  friction  %atang(0.09)
changes the 9% slope to radians
a2=asin(((mice+mhyd)*myhyd1*g*cos(amaxhyd))/(mtot*g));  %  Max  slope  angle
with C.H.D where the wheels spin

%____________________________________________________________________
my3=(Fice-Fhyd)/(mice*g*cos(amax));
mymek=(Fice)/((mice+mhyd)*g*cos(amax));  %  Limiting  friction  mechanical
drive
y1=[0 tan(amax)*100 tan(amax)*100];
x1=[0 myice 2];
plot(x1,y1,'linewidth',2)
```

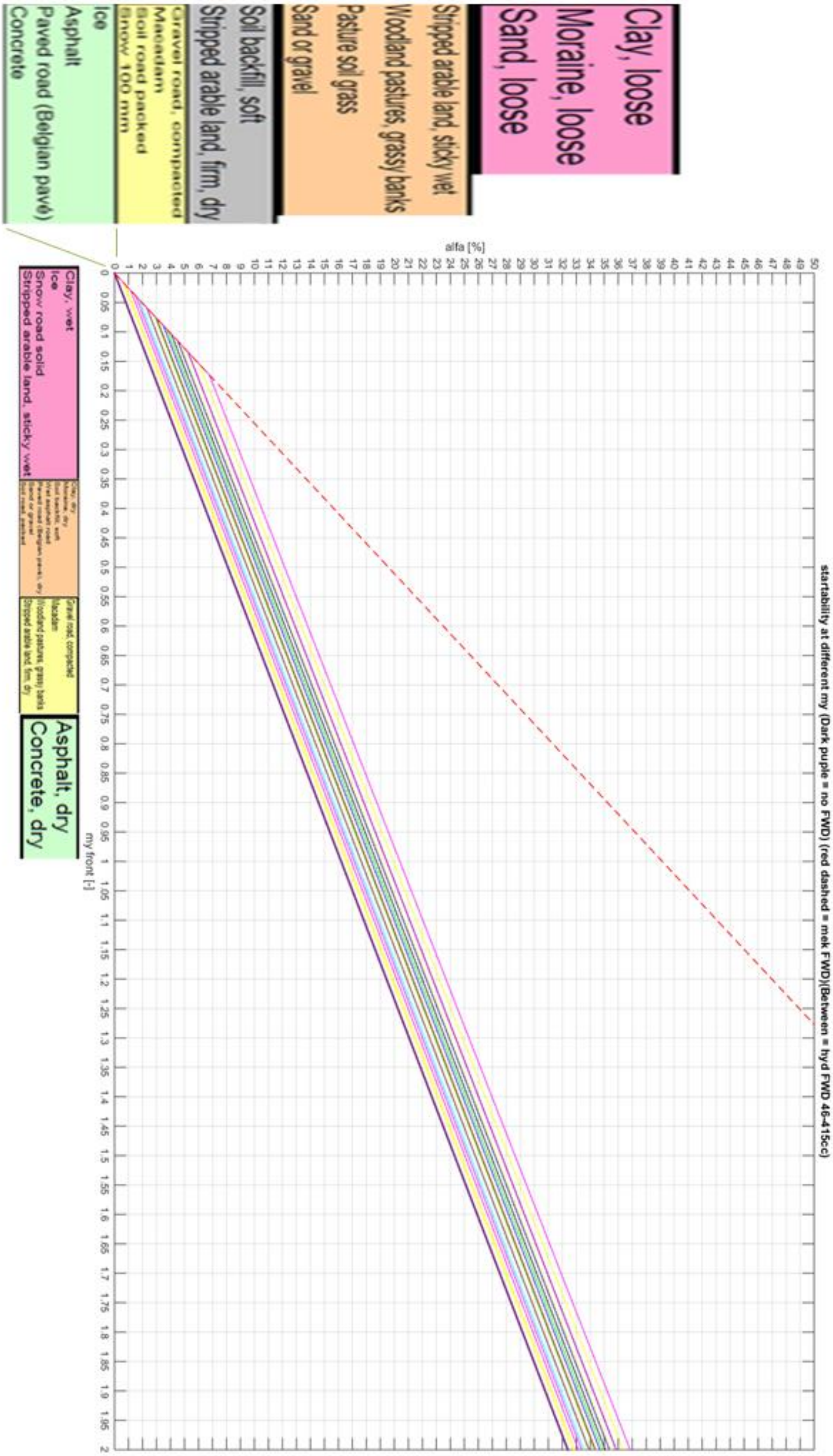
```
xlabel('my front [-]')
ylabel('alfa [%]')
xlim([0 2])
ylim([0 50])
grid on
set(gca,'XTick',0:0.05:2)
set(gca,'YTick',0:1:50)
title('startability at different my (Dark puple = no FWD) (red dashed = mek
FWD)(Between = hyd FWD 46-415cc)')
hold on
y2=[0 tan(a2)*100 tan(amax)*100 tan(amax)*100];
x2=[0 myhyd1 my3 2];
plot(x2,y2,colour(i),'linewidth',0.5)
y3=[0 tan(amax)*100 tan(amax)*100];
x3=[0 mymek my3];
plot(x3,y3,'r--','linewidth',1)
    end
end
```

Appendix X Calculations for RAPDD-GR friction and inclination loaded

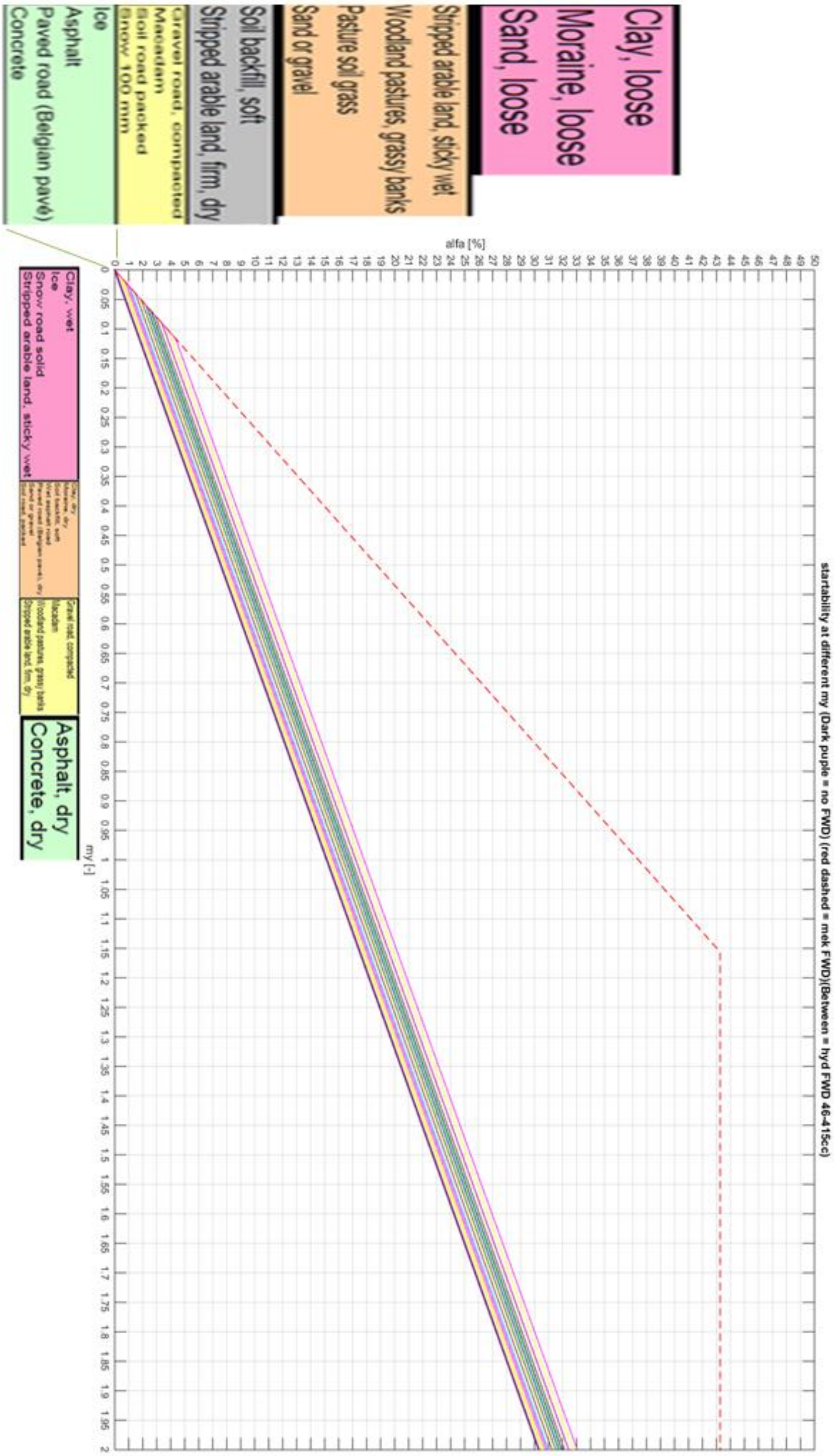
```
clc
clear all
i=0;
j=0;
Thyd=[795.4266894       1590.853379   2386.280068   3181.706758   3977.133447
4772.560137 5567.986826 6363.413515 7158.840205
1029.375716  2058.751431  3088.127147  4117.502863  5146.878579  6176.254294
7205.63001  8235.005726 9264.381442];
colour = ['y' 'm' 'c' 'r' 'g' 'b' 'k' 'y' 'm'];
for i=0:8
    i=i+1;
    for j=0:1
        j=j+1;
Tice=3550; %torque from combustion engine
ratio=11.73*3.61; %total ratio on first gear
mice=17280; %total axel load on the rear driven wheels. 36% weight on rear
axles of total 24 tons *2 axles
% 24000*0,36*2 = 17280
%mhyd=8000; %axel load on front wheel
mhydfront = 8832; % Is dimensioned for 9000 kg but set to 8000 kg
mhydrear = 24000*0.28;
mhyd = mhydfront+mhydrear;
mtot=90000; %total weight for the trailen and truck
radii=(((525-500)/(2500-8000))*mhydfront+536)/1000;   %rolling   radius   on
wheels. based on graf from EDB at mhyd
radiiice=(((525-500)/(2500-8000))*mice+536)/1000;%rolling radius on wheels.
based on graf from EDB at mice
g=9.81; %grofity constant
e=0.93; % efficency on ICE drivetrain
% 93 % verkningsgrad verifierad of Anders Hedman
rr=55*mtot/1000; %rolling resistance on asphalt (N/ton*ton)/1000???
Fice=((Tice*e*ratio)/radiiice); % Driving force produced by engine
amax=asin((Fice)/(mtot*g)); % max slope angle
myice=Fice/(mice*g*cos(amax)); % Limiting friction
Fhyd=(Thyd(j,i)/radii); % Driving force produced by hydraul motors
amaxhyd = asin((Fhyd)/(mtot*g));
myhyd1=(Fhyd)/(mhyd*g*cos(amaxhyd)); % Limiting friction
a2=asin(((mice+mhyd)*myhyd1*g*cos(amaxhyd))/(mtot*g));  %  Max  slope  angle
with C.H.D where the wheels spin
my3=(Fice-Fhyd)/(mice*g*cos(amax));
mymek=(Fice)/((mice+mhyd)*g*cos(amax));   %   Limiting   friction   mechanical
drive
y1=[0 tan(amax)*100 tan(amax)*100];
x1=[0 myice 2];
plot(x1,y1,'linewidth',2)
xlabel('my [-]')
ylabel('alfa [%]')
xlim([0 2])
ylim([0 50])
grid on
set(gca,'XTick',0:0.05:2)
set(gca,'YTick',0:1:50)
title('startability at different my (Dark puple = no FWD) (red dashed = mek
FWD)(Between = hyd FWD 46-415cc)')
hold on
y2=[0 tan(a2)*100 tan(amax)*100 tan(amax)*100];
x2=[0 myhyd1 my3 2];
plot(x2,y2,colour(i),'linewidth',0.5)
y3=[0 tan(amax)*100 tan(amax)*100];
```

```
x3=[0 mymek my3];
plot(x3,y3,'r--','linewidth',1)
    end
end
```

Appendix Y RADT-GR unloaded four motors



Appendix Z RADT-GR loaded four motors



Appendix AA Calculations for RADT-GR friction and inclination unloaded

```
clc
clear all
i=0;
j=0;
% Calculating weight distribution
A = 5347*9.81;
B = 3774*9.81;
C = 0*9.81;
D = 9121*9.81;
p=0;
yu = (D*4014.33-C*p*(4900+1370))/(B*4900);
xu = (D-C*p-B*yu)/A;
%--------------------------------
Thyd=[795.4266894       1590.853379   2386.280068   3181.706758   3977.133447
4772.560137 5567.986826 6363.413515 7158.840205
1029.375716  2058.751431  3088.127147  4117.502863  5146.878579  6176.254294
7205.63001  8235.005726 9264.381442];
Thyd = Thyd/2;
colour = ['y' 'm' 'c' 'r' 'g' 'b' 'k' 'y' 'm'];
for i=0:8
    i=i+1;
    for j=0:1
        j=j+1;
Tice=2000; %torque from combustion engine
ratio=11.73*3.61; %total ratio on first gear
mice = 3774; %
mhydfront = 5347; %
mhydrear = 0;
mhyd = mhydfront+mhydrear;
mtot = 23300; %total weight for the trailen and truck
radii=(((525-500)/(2500-8000))*mhydfront+536)/1000;   %rolling   radius   on
wheels. based on graf from EDB at mhyd
radiiice=(((525-500)/(2500-8000))*mice+536)/1000;%rolling radius on wheels.
based on graf from EDB at mice
g=9.81; %gravity constant
e=0.93; % efficency on ICE drivetrain
% 93 % verkningsgrad verifierad of Anders Hedman
rr=55*mtot/1000; %rolling resistance on asphalt (N/ton*ton)/1000???
Fice=((Tice*e*ratio)/radiiice); % Driving force produced by engine
amax=asin((Fice)/(mtot*g)); % max slope angle
myice=Fice/(mice*g*cos(amax)); % Limiting friction
%_________________________________________________________________

Fhyd=(Thyd(j,i)/radii); % Driving force produced by hydraul motors
amaxhyd = asin((Fhyd)/(mtot*g));
myhyd1=(Fhyd)/(mhyd*g*cos(amaxhyd)); % Limiting friction
a2=asin(((mice+mhyd)*myhyd1*g*cos(amaxhyd))/(mtot*g));  %  Max  slope  angle
with C.H.D where the wheels spin

%____________________________________________________________________
my3=(Fice-Fhyd)/(mice*g*cos(amax));
mymek=(Fice)/((mice+mhyd)*g*cos(amax));  %  Limiting  friction  mechanical
drive
y1=[0 tan(amax)*100 tan(amax)*100];
x1=[0 myice 2];
plot(x1,y1,'linewidth',2)
xlabel('my front [-]')
ylabel('alfa [%]')
```

```
xlim([0 2])
ylim([0 50])
grid on
set(gca,'XTick',0:0.05:2)
set(gca,'YTick',0:1:50)
title('startability at different my (Dark puple = no FWD) (red dashed = mek
FWD)(Between = hyd FWD 46-415cc)')
hold on
y2=[0 tan(a2)*100 tan(amax)*100 tan(amax)*100];
x2=[0 myhyd1 my3 2];
plot(x2,y2,colour(i),'linewidth',0.5)
y3=[0 tan(amax)*100 tan(amax)*100];
x3=[0 mymek my3];
plot(x3,y3,'r--','linewidth',1)
    end
end
```

Appendix BB Calculations for RADT-GR friction and inclination loaded

```
clc
clear all
i=0;
j=0;
Thyd=[795.4266894       1590.853379   2386.280068   3181.706758   3977.133447
4772.560137 5567.986826 6363.413515 7158.840205
1029.375716  2058.751431  3088.127147  4117.502863  5146.878579  6176.254294
7205.63001  8235.005726 9264.381442];
colour = ['y' 'm' 'c' 'r' 'g' 'b' 'k' 'y' 'm'];
for i=0:8
    i=i+1;
    for j=0:1
        j=j+1;
Tice=3550; %torque from combustion engine
ratio=11.73*3.61; %total ratio on first gear
mice=11213;  %total  axel  load  on  the  rear  driven  wheels.  60  %  of  total
weight on rear axles
%mhyd=8000; %axel load on front wheel
mhydfront = 9000; % Is dimensioned for 9000 kg but set to 8000 kg
% va ska vi ha på fronten för denna kombination??
mhydrear = 7475;
mhyd = mhydfront+mhydrear;
mtot=74000; %total weight for the trailen and truck
radii=(((525-500)/(2500-8000))*mhydfront+536)/1000;   %rolling   radius   on
wheels. based on graf from EDB at mhyd
radiiice=(((525-500)/(2500-8000))*mice+536)/1000;%rolling radius on wheels.
based on graf from EDB at mice
g=9.81; %grofity constant
e=0.93; % efficency on ICE drivetrain
% 93 % verkningsgrad verifierad of Anders Hedman
rr=55*mtot/1000; %rolling resistance on asphalt (N/ton*ton)/1000???
Fice=((Tice*e*ratio)/radiiice); % Driving force produced by engine
amax=asin((Fice)/(mtot*g)); % max slope angle
myice=Fice/(mice*g*cos(amax)); % Limiting friction
Fhyd=(Thyd(j,i)/radii); % Driving force produced by hydraul motors
amaxhyd = asin((Fhyd)/(mtot*g));
myhyd1=(Fhyd)/(mhyd*g*cos(amaxhyd)); % Limiting friction
a2=asin(((mice+mhyd)*myhyd1*g*cos(amaxhyd))/(mtot*g));  %  Max  slope  angle
with C.H.D where the wheels spin
my3=(Fice-Fhyd)/(mice*g*cos(amax));
mymek=(Fice)/((mice+mhyd)*g*cos(amax));   %   Limiting   friction   mechanical
drive
y1=[0 tan(amax)*100 tan(amax)*100];
x1=[0 myice 2];
plot(x1,y1,'linewidth',2)
xlabel('my [-]')
ylabel('alfa [%]')
xlim([0 2])
ylim([0 50])
grid on
set(gca,'XTick',0:0.05:2)
set(gca,'YTick',0:1:50)
title('startability at different my (Dark puple = no FWD) (red dashed = mek
FWD)(Between = hyd FWD 46-415cc)')
hold on
y2=[0 tan(a2)*100 tan(amax)*100 tan(amax)*100];
x2=[0 myhyd1 my3 2];
plot(x2,y2,colour(i),'linewidth',0.5)
y3=[0 tan(amax)*100 tan(amax)*100];
```

```
x3=[0 mymek my3];
plot(x3,y3,'r--','linewidth',1)
    end
end
```

Appendix CC Flow velocity using recomendation

```
clear all
Dp = 90; % displacment for pump in cm^3 (90cc)
Dpb = 19.6; % feeder pump displacement
nvolp = xx;
ne=2000;
ipto = 1.26;
Q = (Dp*ne*ipto*nvolp)/1000 % [l/min]
Qpb =(Dpb*ne*ipto*nvolp)/1000

dAB = 25/1000; % Diameter higher pressure A and B
dAB12 = 18/1000; % Diameter highpressure AM1-BM2
dFe1Fa = 20/1000; % BPR to pump ports Fe1 to Fa
dG = 15/1000; % Diameter feeder from pump to valve
dLm12 = 12/1000; % Feederline from valve block to motors
dTkT2 = 12/1000; % Feederline to pump house reservoir
dSFS = 35/1000; % suction line feeder pump port Sf - S
% High pressure
cAB = (Q*4)/(dAB^2*pi)/60000 % flow velocity pump - valve ports A and B
cAB12 = (Q/2*4)/(dAB12^2*pi)/60000 % Flow velocity valve - motors ports A1-
B2
%Feeder line. Flow calculated from motors to pump house
%cFe1Fa = (Qpb/2*4)/(dFe1Fa^2*pi)/60000 % Flow velocity feeder line pump -
valve port G
cG = (4.5*2*4)/(dG^2*pi)/60000 % Flow velocity feeder line pump - valve
port G
% Has reduced flow to half due to split???
cLm12 = (4.5*4)/(dLm12^2*pi)/60000 % Flow velocity feeder line valve -
motors port LM1 & LM2 is also a return line
% A quarter flow to here??
%cTkT2 = (Qpb/2*4)/(dTkT2^2*pi)/60000 % Flow velocity feeder line valve -
motors port LM1 & LM2
% Suction to feeder
cSFS = (Qpb*4)/(dSFS^2*pi)/60000 % Flow velocity feeder line valve - motors
port LM1 & LM2
```

Appendix DD Flow velocity FH-1825

```
clear all
Dp = 90; % displacment for pump in cm^3 (90cc)
Dpb = 19.6; % feeder pump displacement
nvolp = 0.93;
ne=2000;
ipto = 1.26;
d = (1*25.4)/1000; % I.D. 1" from pump to port A and B
d2 = (5/8*25.4)/1000; % I.D. 5/8" from AM1, AM2, BM1 and BM2
d3 = (1/2*25.4)/1000; % I.D. 1/2" From Lm1 and Lm2 to T
d4 = (1.25*25.4)/1000;

% d"* 25,4 gives mm 1/1000 => m
% d = I.D. for the hoses to the motors on fh-1825
Q = ((Dp*ne*ipto*nvolp)/1000)/60000
Q2=Q/2 % Q/2 since flow devided to two motors
Qbp =(Dpb*ne*ipto*nvolp)/1000/60000 % flow produced from feeder pump
Qb = 4.5/60000 % Flow directly at motors feeder

cmax = 8; % parker says recomended 5-6 and 8 as max [m/s]
IDp = 4.61 * sqrt(Q/cmax) % Minimum I.D. at the outlet of the pump [mm]

cm1 = (Q*4)/(d^2*pi) % flow velocity from port A and B pump
cm2 = (Q2*4)/(d2^2*pi) % flow velocity to motors
cm3 = (Qb*4)/(d3^2*pi) % flow velocity feeder lines valve block to motors
cm4 = (Qb*2*4)/(d3^2*pi) % flow velocity feeder lines feeder pump to valve
block
cm5 = (Qbp*4)/(d4^2*pi) % suction line to pump
```

Appendix EE If reduced velocities

```
clear all
Dp = 90; % displacment for pump in cm^3 (90cc)
Dpb = 19.6; % feeder pump displacement
nvolp = 0.93;
ne=2000;
ipto = 1.26;
d = (1*25.4)/1000; % I.D. 1" from pump to port A and B
d2 = (5/8*25.4)/1000; % I.D. 5/8" from AM1, AM2, BM1 and BM2
d3 = (1/2*25.4)/1000; % I.D. 1/2" From Lm1 and Lm2 to T
d4 = (1.25*25.4)/1000;

% d"* 25,4 gives mm 1/1000 => m
% d = I.D. for the hoses to the motors on fh-1825
Q = ((Dp*ne*ipto*nvolp)/1000)/60000
Q2=Q/2 % Q/2 since flow devided to two motors
%Qbp =(Dpb*ne*ipto*nvolp)/1000/60000 % flow produced from feeder pump
Qb = 4.5/60000 % Flow directly at motors feeder calculated by the hydraulics supplier

% If one pair is only used in the 4 wheel system
cmax = 7; % parker says recomended 5-6 and 8 as max [m/s]
cmax2 = 3;
IDm1 = 4.61 * sqrt(Q2*60000/2/cmax); % Minimum I.D. for high presure lines to motors [mm]
IDminch1 = IDm1/25.4 % converts to inch
IDm2 = 4.61 * sqrt(Qb*60000/2/cmax2); % Minimum I.D. for feeder lines to motors [mm]
IDminch2 = IDm2/25.4 % converts to inch

cm2 = (Q2/2*4)/(d2^2*pi) % flow velocity to motors
cm3 = (Qb*4)/(d3^2*pi) % flow velocity feeder lines valve block to motors, is not devided by two since this is the value given by the hydraulics supplier
```

Appendix FF Flow velocities using 140 cc pump

```
clear all
Dp = 140; % displacment for pump in cm^3 (90cc)
Dpb = 19.6; % feeder pump displacement
nvolp = 0.93;
ne=2000;
ipto = 1.26;
Q = (Dp*ne*ipto*nvolp)/1000 % [l/min]
Qpb =(Dpb*ne*ipto*nvolp)/1000

dAB = 25/1000; % Diameter higher pressure A and B
dAB12 = 18/1000; % Diameter highpressure AM1-BM2
dFe1Fa = 20/1000; % BPR to pump ports Fe1 to Fa
dG = 15/1000; % Diameter feeder from pump to valve
dLm12 = 12/1000; % Feederline from valve block to motors
dTkT2 = 12/1000; % Feederline to pump house reservoir
dSFS = 35/1000; % suction line feeder pump port Sf - S
% High pressure
cAB = (Q*4)/(dAB^2*pi)/60000 % flow velocity pump - valve ports A and B
cAB12 = (Q/2*4)/(dAB12^2*pi)/60000 % Flow velocity valve - motors ports A1-
B2
%Feeder line. Flow calculated from motors to pump house
%cFe1Fa = (Qpb/2*4)/(dFe1Fa^2*pi)/60000 % Flow velocity feeder line pump -
valve port G
cG = (4.5*2*4)/(dG^2*pi)/60000 % Flow velocity feeder line pump - valve
port G
% Has reduced flow to half due to split???
cLm12 = (4.5*4)/(dLm12^2*pi)/60000 % Flow velocity feeder line valve -
motors port LM1 & LM2 is also a return line
% A quarter flow to here??
%cTkT2 = (Qpb/2*4)/(dTkT2^2*pi)/60000 % Flow velocity feeder line valve -
motors port LM1 & LM2
% Suction to feeder
cSFS = (Qpb*4)/(dSFS^2*pi)/60000 % Flow velocity feeder line valve - motors
port LM1 & LM2
```

Appendix GG Description of compatible concepts

To increase understanding of the different solutions for the reader, the solutions indicated with green colour & “Yes” are hereby presented, where first concepts for RAPDD-GR are presented, then for RADT-GR.

1B1A

The hoses are routed between the AUXPark and the axle to a bracket which is fastened in the two holes used to secure the axle to the suspension, here it is important that the bracket is hard enough to not deform once the screws are tightened. At this bracket next section of hoses is attached and routed with a slack up to the chassis. The first section of this solution is subjected to the rotational movement which occurs when turning and the second section is subjected to vertical movement which occurs when the suspension is moving, see Figure 21.

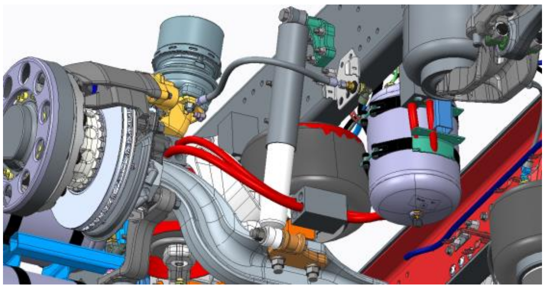


Figure 21 1B1A

1B2A

The hoses are directed between the AUXPark and the axle to the chassis side where the hoses are secured to a bracket attached to the chassis side at which a change to pipes also occurs. This solution does not divide the rotational and vertical movement over two sections, meaning each hose needs to handle both these movements, see Figure 22.

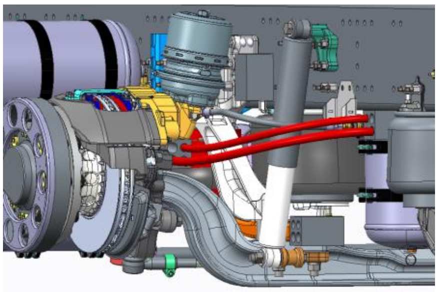


Figure 22 1B2A

1B3A1

The hoses are lead between the AUXPark and the axle and then do a 180 degree U-turn towards the bogie lift where a bracket is attached to the air bellows taps on the top. On the bracket nipples are attached and a new section of hoses are led to the chassis side. The bogie lift moves together with the remaining suspension leading to that the first hose section is subjected to the rotational movement and the second section vertical movement, see Figure 23.

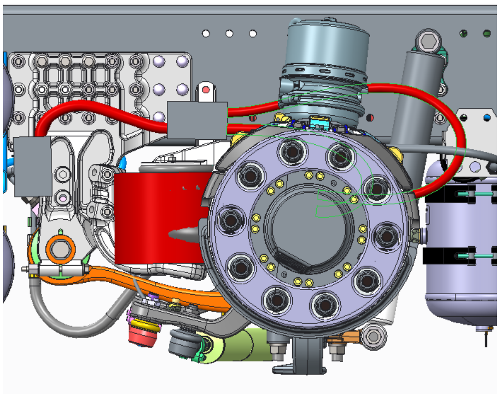


Figure 23 1B3A1

1B3A2

Same as 1B3A1 but with the difference that the hoses are lead between the AUXPark and axle from the rear side to the bracket located on the bogie lift and then a second section of hoses are lead to chassis. Just as the case for 1B3A1 the first hose section is subjected to the rotational movement and the second section vertical movement, Figure 24.

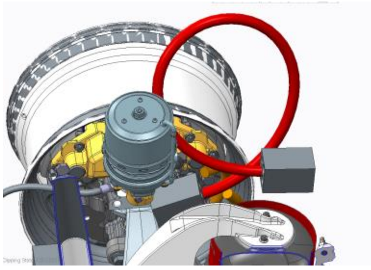


Figure 24 1B3A2

2B1A

The hoses are fitted to the pipes from the wheel spindle on the front side, routed around the AUXPark down to a bracket which is fitted to the holes used to secure the axle to the suspension, where the hoses then ascend vertically to the chassis side. The first section of hoses will be subjected to the turning motion and the second to the vertical movement, see Figure 25.

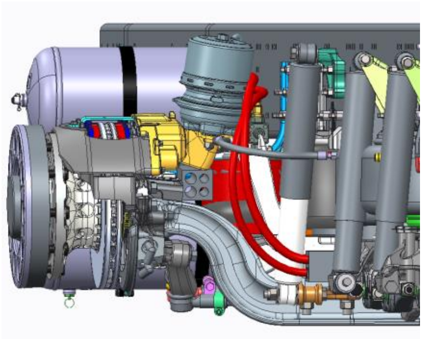


Figure 25 2B1A

2B2A

The hoses are fitted to the pipes from the wheel spindle on the front side, routed around the AUXPark to a bracket attached to the chassis where the hoses are switched to pipes. All the hoses will be subjected to both rotational and vertical movement see Figure 26.

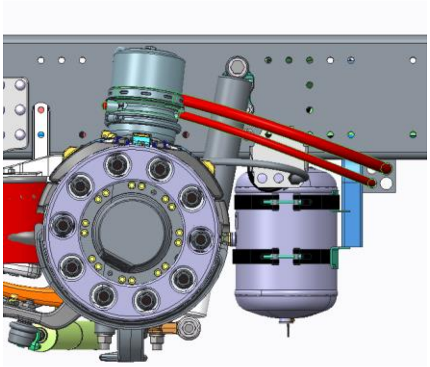


Figure 26 2B2A

2B3A

The pipes from the wheel spindle are lead on the rear side of the AUXPark, interchanged to hoses which are routed around the AUXPark to a bracket attached to the bogie lift where a second section of hoses are attached and lead towards a bracket attached to the chassis. The bogie lift moves together with the remaining suspension leading to that the first hose section is subjected to the rotational movement and the second section vertical movement, see Figure 27.

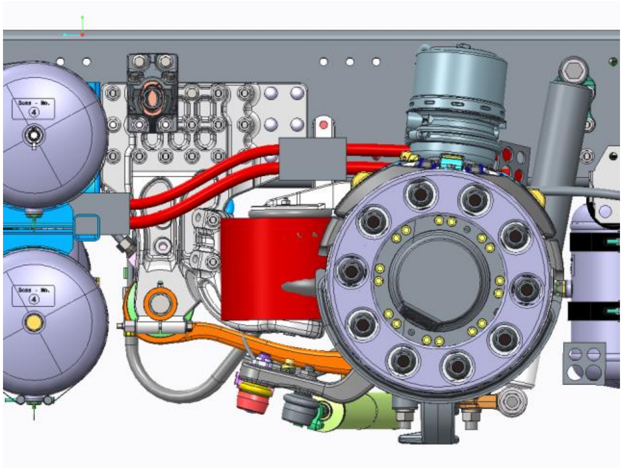


Figure 27 2B3A

1B1AA

The hoses are lead between the wheel brake and the axle to a bracket which is attached to the axle's rear side in the already existing M10 holes. The hoses are then interchanged to pipes which are routed to a bracket attached to the bogie lift bracket, where hoses are attached and routed to chassis. The first hose section is subjected to the rotational movement, and the second from the vertical movement, see Figure 28.

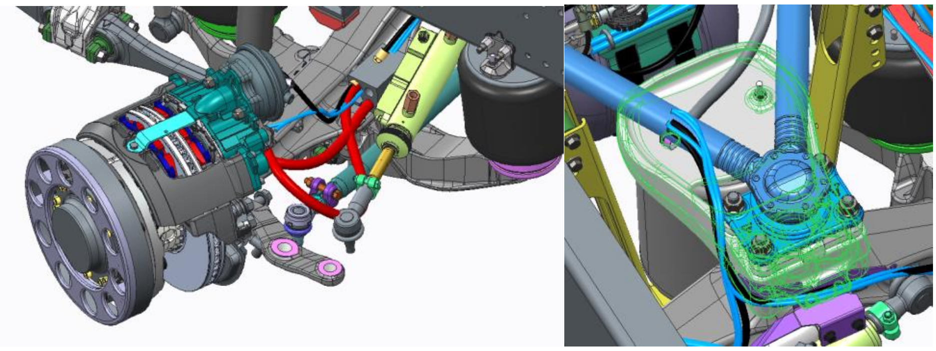


Figure 28 1B1AA

1B2AA

The hoses are lead between the wheel brake and the axle to a bracket located between the shock absorber and air spring which is attached to the axle's front side in the already existing M10 holes on the rear side or the M8 holes on the front side if possible. The hoses are then interchanged to pipes which are routed to a bracket attached to the bogie lift bracket, where hoses are attached and routed to chassis as in 1B1AA. The first hose section is subjected to the rotational movement, and the second from the vertical movement, see Figure 29.

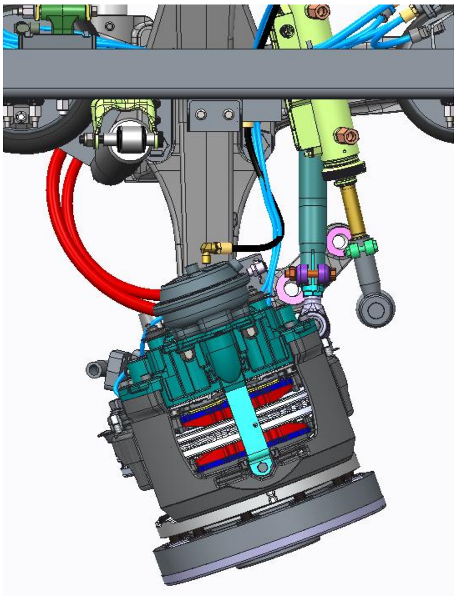


Figure 29 1B2AA

1B3AA

The hoses are lead between the wheel brake and axle from the rear side of the axle towards a bracket attached to the chassis. All the hoses will be subjected to both rotational and vertical movement see Figure 30.

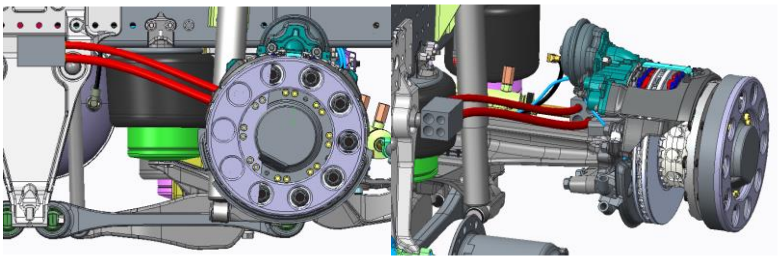


Figure 30 1B3AA

1B4AA

The hoses are lead between the wheel brake and axle from the front side of the axle towards a bracket attached to the chassis. All the hoses will be subjected to both rotational and vertical movement see Figure 31.

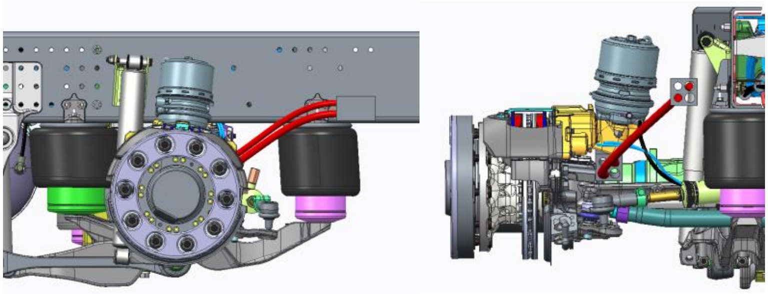


Figure 31 1B4AA

2B3AA

The hoses are lead around the wheel brake from the rear side of the axle towards a bracket attached to the chassis. All the hoses will be subjected to both rotational and vertical movement see Figure 32.

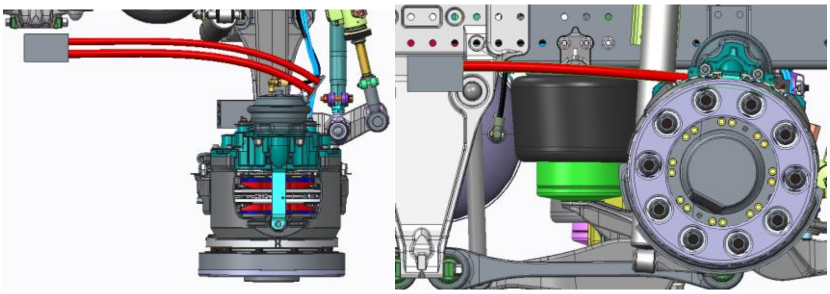


Figure 32 2B3AA

2B4AA

The hoses are lead around the wheel brake from the front side of the axle towards a bracket attached to the chassis. All the hoses will be subjected to both rotational and vertical movement see Figure 33.

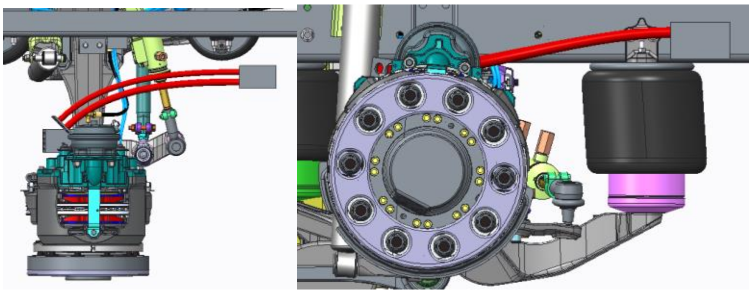


Figure 33 2B4AA

7B1AA

The hoses are routed above the wheel brake down to a bracket attached to the axle's rear side in the already existing M10 holes. The hoses are then interchanged to pipes which are routed to a bracket attached to the bogie lift bracket, where hoses are attached and routed to chassis as in 1B1AA. The first hose section is subjected to the rotational movement, and the second from the vertical movement, see Figure 34.

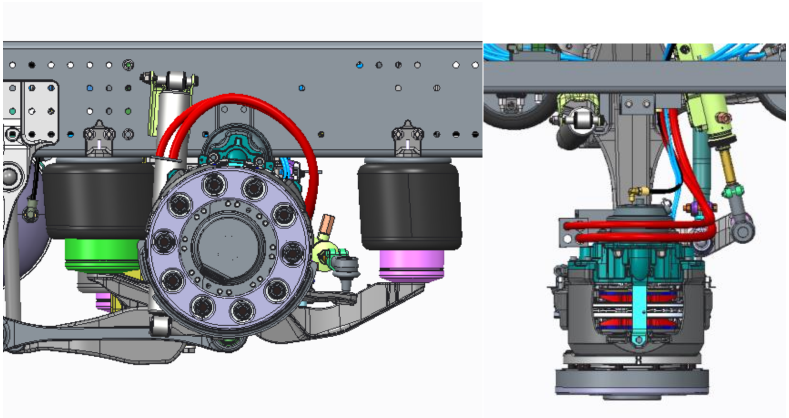


Figure 34 7B1AA

7B3AA

The hoses are led over the wheel brake from the front side of the axle back between the wheel brake and axle towards the bracket attached to the chassis. Each hose will be subjected to rotational movement and vertical movement, see Figure 35.

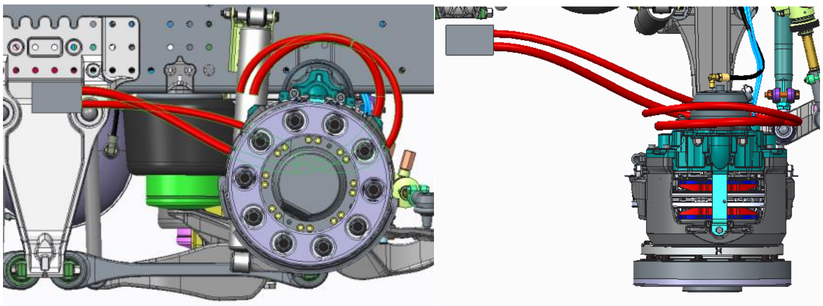


Figure 35 7B3AA

7B4AA

The hoses are lead above the wheel brake from the front side of the axle towards a bracket attached to the chassis. All the hoses will be subjected to both rotational and vertical movement see Figure 36.

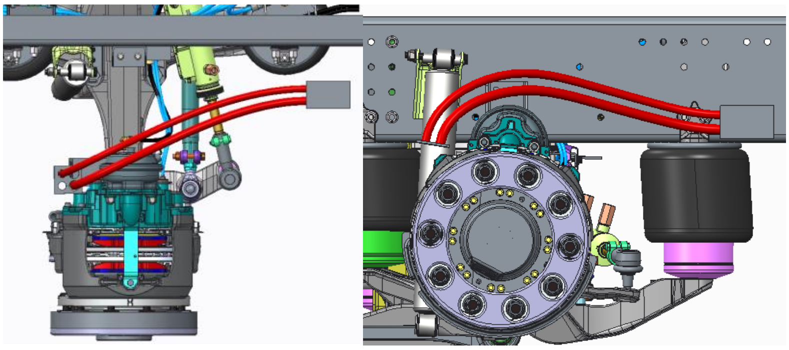


Figure 36 7B4AA

8B2AA

The hoses are routed above the wheel brake from the rear side down to a bracket attached to the axle’s front side in the already existing M10 holes on the rear side, or in the M8 holes on the front side if possible. The hoses are then interchanged to pipes which are routed to a bracket attached to the bogie lift bracket, where hoses are attached and routed to chassis as in 1B1AA. The first hose section is subjected to the rotational movement, and the second from the vertical movement, see Figure 37.

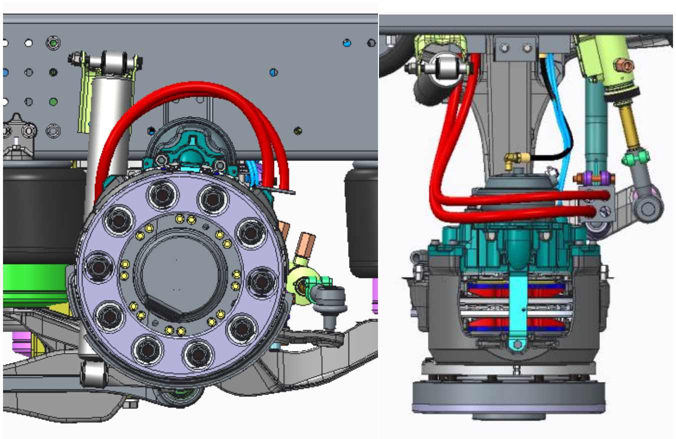


Figure 37 8B2AA

8B3AA

The hoses are led above the wheel brake from the rear side of the axle towards a bracket attached to the chassis. All the hoses will be subjected to both rotational and vertical movement see Figure 38.

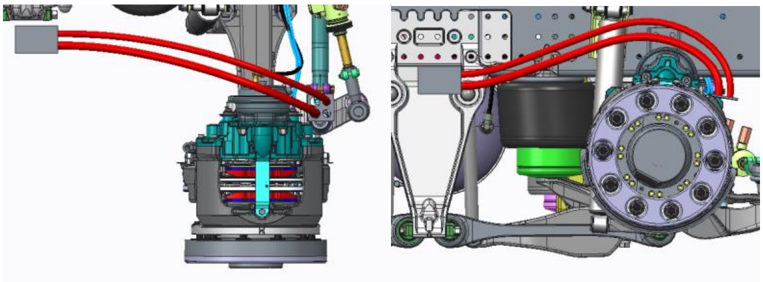


Figure 38 8BAA

8B4AA

The hoses are led over the wheel brake from the rear side of the axle back between the wheel brake and axle towards the bracket attached to the chassis. Each hose will be subjected to rotational movement and vertical movement, see Figure 39.

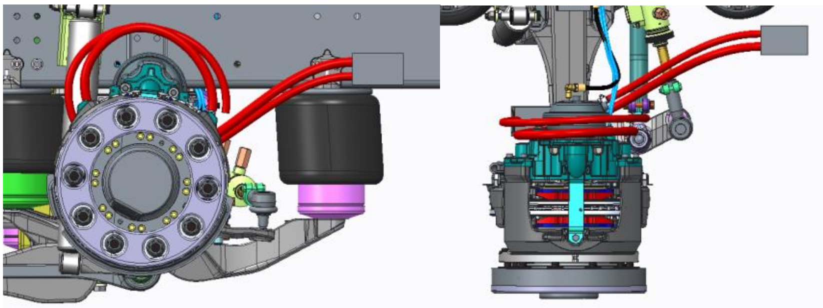


Figure 39 8B4AA

Appendix HH Pros and cons

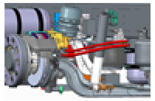
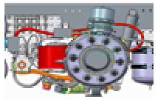
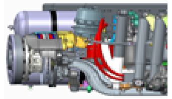
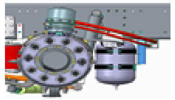
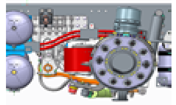
B - Concepts	A - Concepts	Pros	Cons	AA - Concepts	Pros	Cons
1B	1A	+ Seperates vertical movement and rotational movement	- Will still result in some twisting when turning since the two connections are not in the same planes. (could be improved or - Would probably require change of placment for air tank between pusher and driven axle	1AA	+ Seperates rotation and vertical movement over to hose sections + "Natural" routing of hose which does not take space from other parts + Several routing possibilities to chassis + Does not occupy space for wheel brake	- Steering rod imposes packaging difficulties - Minimum bending radius requires an angled connection on the axle to enable a larger loop if to be
	2A	+ Fewer parts needs to be used	- Every hose needs to handle both rotational movement and vertical movement - The need of longer hoses can implement some problems with bending radius			
	3A1 (entering from front side)	+ Largge amount of space to allow bending radius	- Tight fit between AUXPark and damping cylinder since the hoses here will be freely moving from the start of the bend - Will still be twisting when turning since the hose ends are not in the twisting plane - New screw taps for the air bellow to attach bracket will be needed	2AA	+ "Natural" routing of hose which does not take space from other parts + Several routing possibilities to chassis + Does not occupy space for wheel brake	- Minimum bending radius requirement requires hoses to be routet between air bellow and air damper, small amount of available space - Hoses more vulnerable since on the front side
	3A2 (entering from rear side)	+ Transition between hose section 1 and 2 is in line with each other	- The distance between the two attachemnt sides for the hoses subjected to rotational movement will be to short - New screw taps for the air bellow to attach bracket will be needed	3AA	+ Maybe be possible to implement without moving or modifying existing parts + More available free space + Total system can be made shorter than the other concepts and therefore reduce expected losses	- Deformation in several planes will occur, thus also twisting of hoses - Less attachment possibilities on rear side of
				4AA	+ Maybe be possible to implement without moving or modifying existing parts + More available free space + Total system can be made shorter than the other concepts and therefore reduce expected losses	- Deformation in several planes will occur, thus also twisting of hoses - Total system will be longer and therefore increased losses can be expected

B - Concepts	A - Conc	Pros	Cons	AA - Concepts	Pros	Cons
2B	1A	+ Allows for longer hose length + "good" space for fittings at change from pipes to hoses	- Not a natural bend for the hose, bend goes in several planes	3AA	+ The hose ends can more easily be leveled in driving mode + Shorter routing distance, lower pressure	- Deformation will occur in several planes, meaning twisting of hoses will occur - Modification in mudlfaps will be needed - Risk of contact with damping cylinder - Less attachment possibilities on rear side of spindle
	2A	+ Hose ends can be in the same horisontal plane when in driving height	- Each hose needs to handle both the steering rotationa and vertical movement			
	3A	+ Each section only handles rotation or vertical movement + Hose ends can be set to move in the same plane	- Air bellows infront of pusher axle needs to be moved to enable possibility to route hoses - New taps to air bellow is needed => special order of bellow is needed	4AA	+ The hose ends can more easily be leveled in driving mode + Better attachment possibilities on the front side of the spindle + Less objects occupies area of interest	- Deformation will occur in several planes, meaning twisting of hoses will occur - Modification in mudlfaps will be needed

B - Conc	AA - Concept	Pros	Cons
7B	1AA	+ Relatively large amount of space between wheel brake and mud flaps	- Can be difficult to allow hose ends to be in the same plane - Occupies lare amount of space of wheel end area - Long brackets will be needed
	3AA	+ Long hose length to absorb twisting motion	- Hoses will occupy large area - Solution will need hoses to be held in place between wheel brake and axle but still able to freely rotate. Slide bearings might stop working
	4AA	+ Longer hoses which handles twisting better + Possible to have hose ends leveled in driving mode	- Each hose handles both rotational movement and vertical movement, twisting is enevitable - Mud flaps will probably be needed to be modified - Routing backwards will lead to longer flow system and higher pressure losses

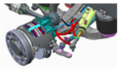
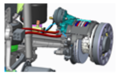
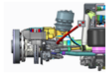
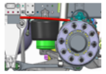
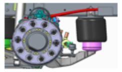
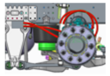
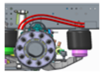
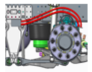
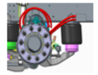
B - Concepts	AA - Concepts	Pros	Cons
8B	2AA	+ Relatively large amount of space between wheel brake and mud flaps	- Can be difficult to allow hose ends to be in the same plane - Occupies lare amount of space of wheel end area - Vulnerable location on the front side of axle
	3AA	+ Routing towards the valve block + Longer hoses which handles twisting better + Possible to have hose ends leveled in driving mode	- Each hose handles both rotational movement and vertical movement, twisting is enevitable - Mud flaps will probably be needed to be modified
	4AA	+ Long hose length to absorb twisting motion	- Hoses will occupy large area - Solution will need hoses to be held in place between wheel brake and axle but still able to freely rotate. Slide bearings might stop working - Less good fastening possibilities on the rear side

Appendix II Kesselring RAPDD-GR

RAPDD-GR		Ideal		1B1A2		1B2A		1B&3A1		1B3A2		2B1A2		2B2A		2B3A	
Criteria	Weight	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score
Seperates vertical movement and rotational movement	5	5	25	5	25	2	10	5	25	5	25	5	25	3	15	5	25
Number of used parts	3	5	15	2	6	3	9	2	6	2	6	1	3	3	9	2	6
Possibility to give hose required slack	5	5	25	3	15	4	20	2	10	1	5	4	20	4	20	2	10
Required amount of space	4	5	20	2	8	3	12	2	8	2	8	3	12	2	8	2	8
Complexity of used brackets	3	5	15	2	6	4	12	2	6	2	6	3	9	4	12	2	6
Degree of twisting (hose ends in the same plane as movement)	5	5	25	3	15	2	10	3	15	2	10	2	10	3	15	5	25
Possibility to realize (available space, fastening possibility,	4	5	20	4	16	5	20	3	12	2	8	2	8	5	20	3	12
Required number of changes	3	5	15	3	9	3	9	2	6	2	6	3	9	3	9	2	6
Occupied space at wheel end	3	5	15	4	12	4	12	3	9	1	3	3	9	3	9	3	9
Vulnerability to damage	4	5	20	3	12	4	16	2	8	2	8	3	12	4	16	4	16
Possibility to add swivel- or quick fittings	2	5	10	3	6	4	8	2	4	1	2	3	6	3	6	2	4
Manufacturing	2	5	10	4	8	4	8	1	2	1	2	1	2	2	4	2	4
Aftermarket	2	5	10	4	8	4	8	2	4	2	4	2	4	2	4	2	4
Total score			225		146		154		115		93		129		147		135
Relative score to ideal			100%		65%		68%		51%		41%		57%		65%		60%
Rank					2		1		6		7		5		2		4
Perceived feeling	5	5	25	3	15	5	25	2	10	1	5	2	10	4	20	3	15
Adjusted total score			250		161		179		125		98		139		167		150
Adjusted relative score			100%		64%		72%		50%		39%		56%		67%		60%
Adjusted rank					3		1		6		7		5		2		4

Appendix JJ Kesselring RADT-GR

RADT-GR		Ideal		1B1AA		1B2AA		1B3AA		1B4AA		2B3AA		2B4AA		7B1AA		7B3AA		7B4AA		8B2AA		8B3AA		8B4AA	
Criteria	Weight	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score
Seperates vertical movement and rotational movement	5	5	25	5	25	5	25	1	5	1	5	1	5	1	5	5	25	1	5	1	5	5	25	1	5	1	5
Number of used parts	3	5	15	2	6	2	6	3	9	3	9	3	9	3	9	2	6	3	9	3	9	2	6	3	9	3	9
Possibility to give hose required slack	5	5	25	1	5	2	10	4	20	4	20	3	15	4	20	4	20	5	25	4	20	3	15	2	10	5	25
Required amount of space	4	5	20	3	12	3	12	3	12	3	12	3	12	2	8	2	8	1	4	3	12	2	8	2	8	1	4
Complexity of used brackets	3	5	15	2	6	1	3	4	12	3	9	4	12	3	9	2	6	3	9	3	9	3	9	3	9	3	9
Degree of twisting (hose ends in the same plane as movement)	5	5	25	3	15	3	15	2	10	2	10	3	15	3	15	3	15	3	15	3	15	3	15	3	15	3	15
Possibility to realize (available space, fastening possibility,	4	5	20	1	4	1	4	0	0	3	12	4	16	4	16	2	8	2	8	3	12	1	4	2	8	2	8
Required number of changes	3	5	15	5	15	5	15	3	9	3	9	3	9	3	9	5	15	3	9	3	9	5	15	3	9	3	9
Occupied space at wheel end	3	5	15	4	12	3	9	4	12	4	12	3	9	3	9	1	3	1	3	3	9	2	6	3	9	1	3
Vulnerability to damage	4	5	20	2	8	2	8	4	16	4	16	3	12	4	16	3	12	3	12	3	12	2	8	2	8	3	12
Possibility to add swivel- or quick fittings	2	5	10	1	2	1	2	4	8	4	8	4	8	4	8	3	6	4	8	3	6	2	4	3	6	4	8
Manufacturing	2	5	10	3	6	3	6	4	8	3	6	3	6	3	6	2	4	1	2	3	6	2	4	3	6	1	2
Aftermarket	2	5	10	3	6	3	6	4	8	3	6	3	6	3	6	2	4	2	4	3	6	3	6	3	6	2	4
Total score			225		122		121		129		134		134		136		132		113		130		125		108		113
Relative score to ideal			100%		54%		54%		57%		60%		60%		60%		59%		50%		58%		56%		48%		50%
Rank					8		8		6		1		1		1		4		10		5		7		12		10
Perceived feeling	5	5	25	1	5	1	5	2	10	4	20	2	10	4	20	2	5	1	5	4	20	2	10	4	20	1	5
Adjusted total score			250		127		126		139		154		144		156		137		118		150		135		128		118
Adjusted relative score			100%		51%		50%		56%		62%		58%		62%		55%		47%		60%		54%		51%		47%
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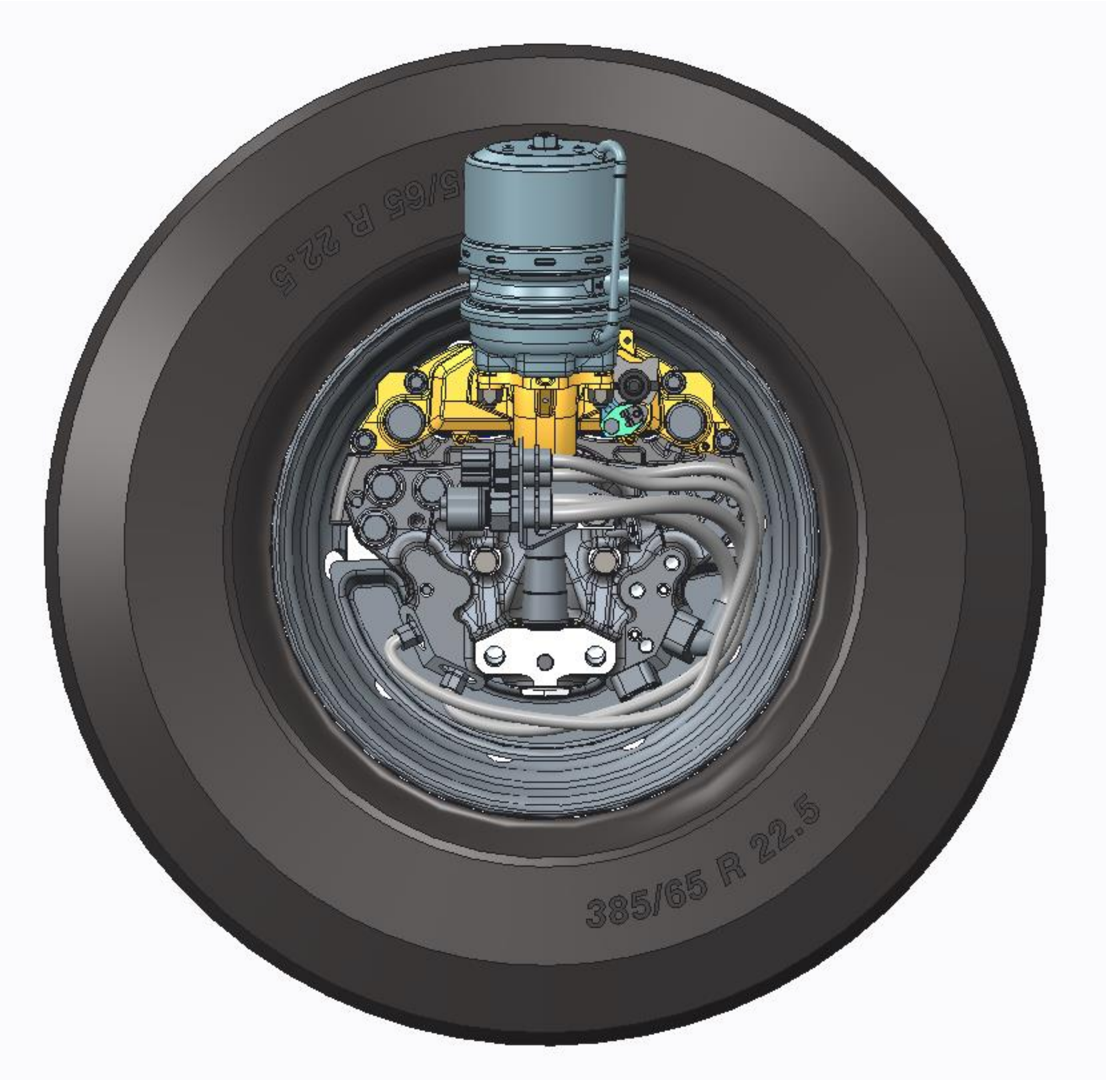
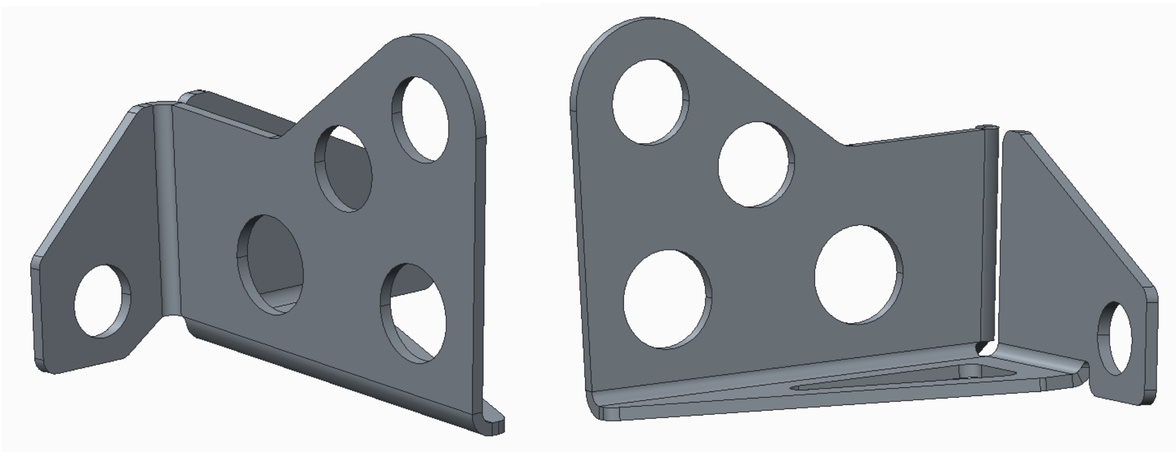
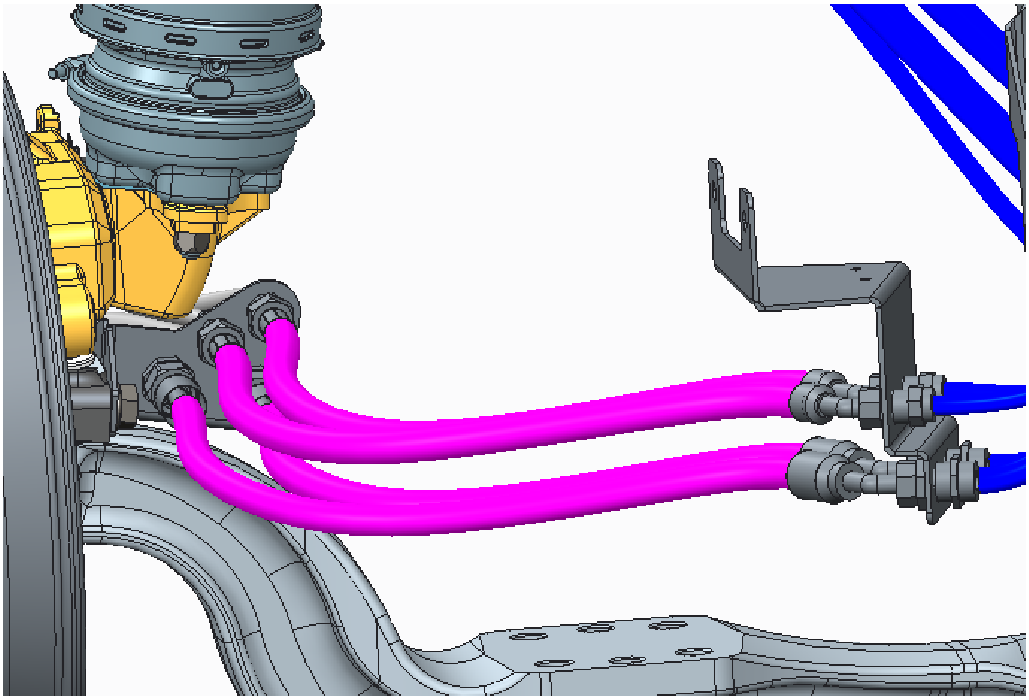
Appendix KK Pipes RAPDD-GR wheel end

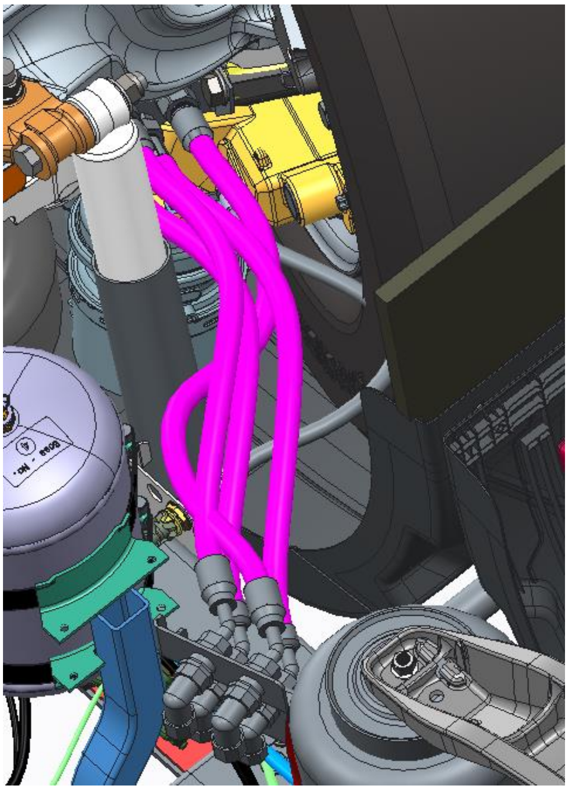


Appendix LL Bracket RAPDD-GR wheel end

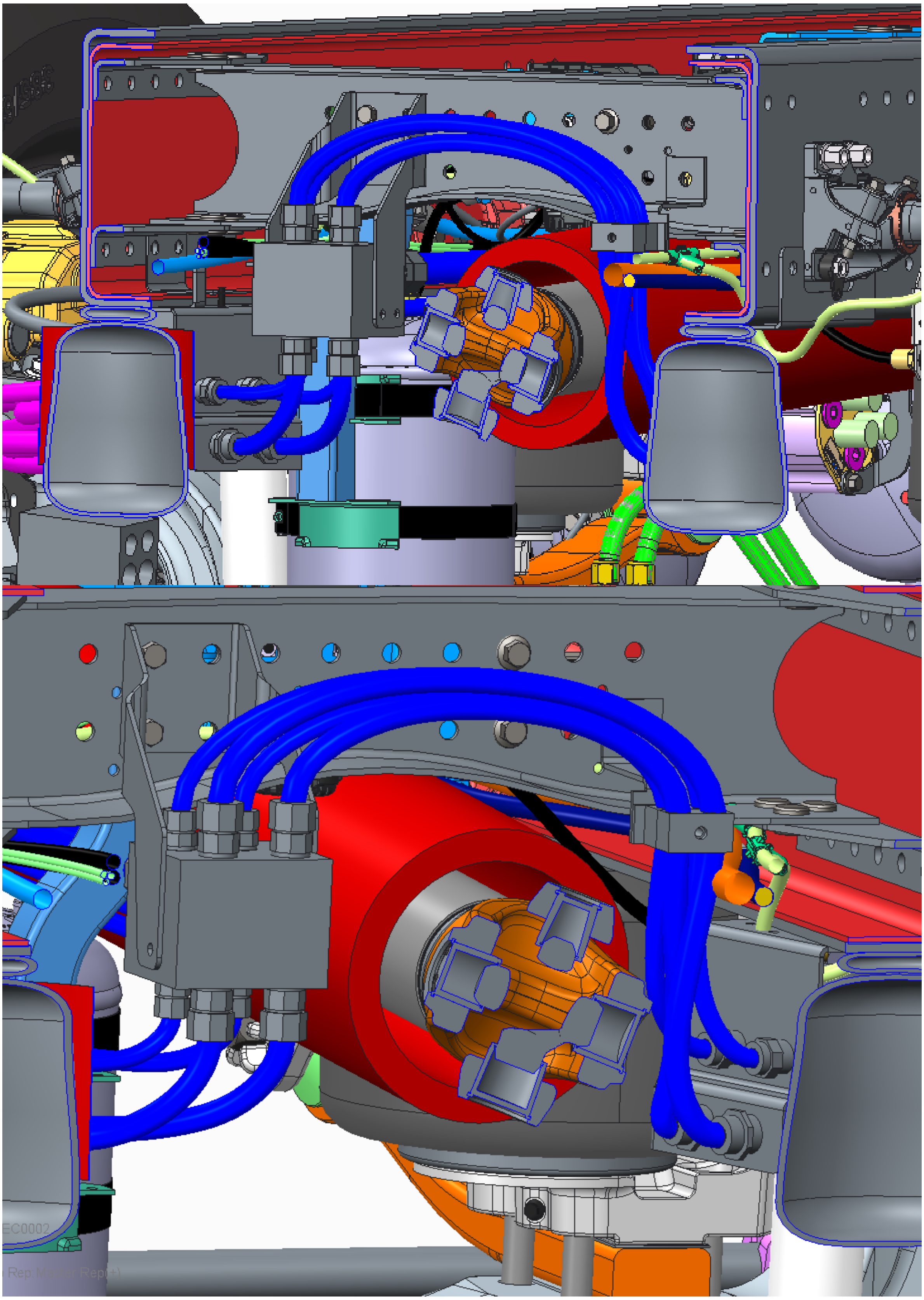


Appendix MM Hoses RAPDD-GR wheel end



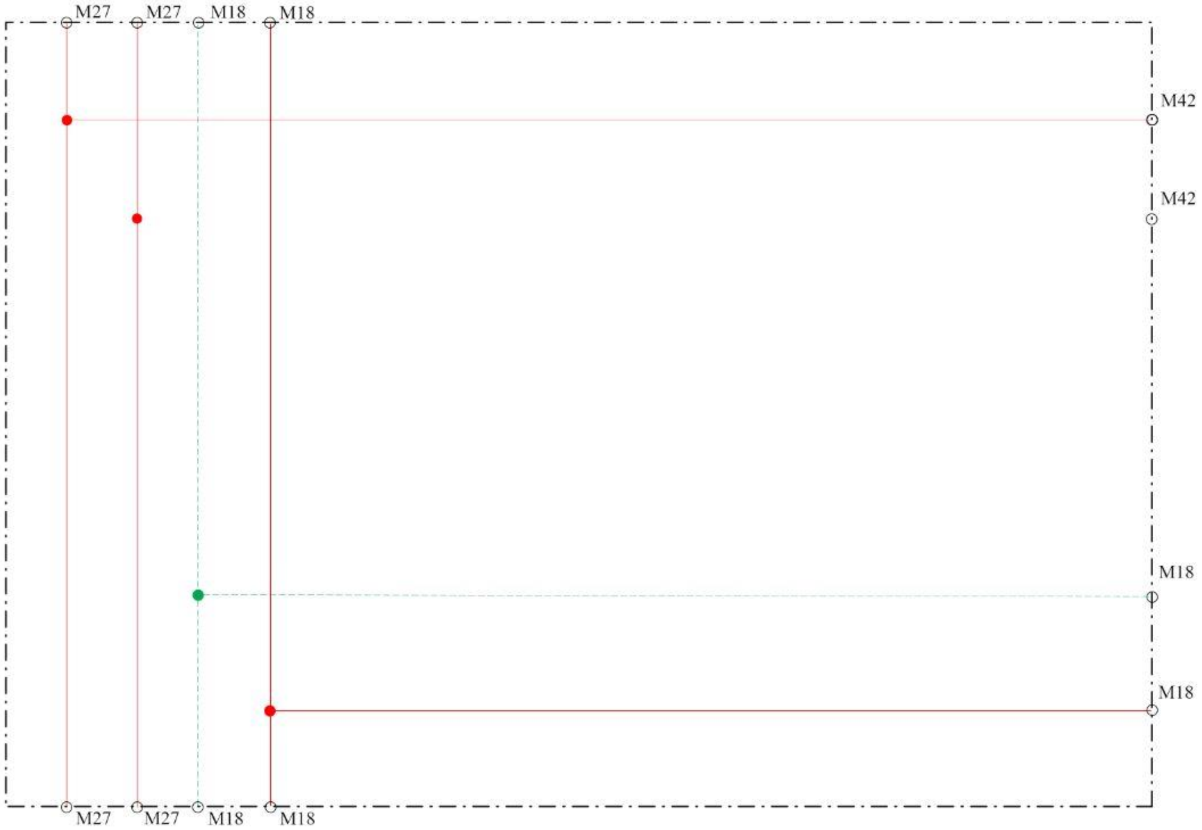


Appendix NN Pipes to connection block RAPDD-GR

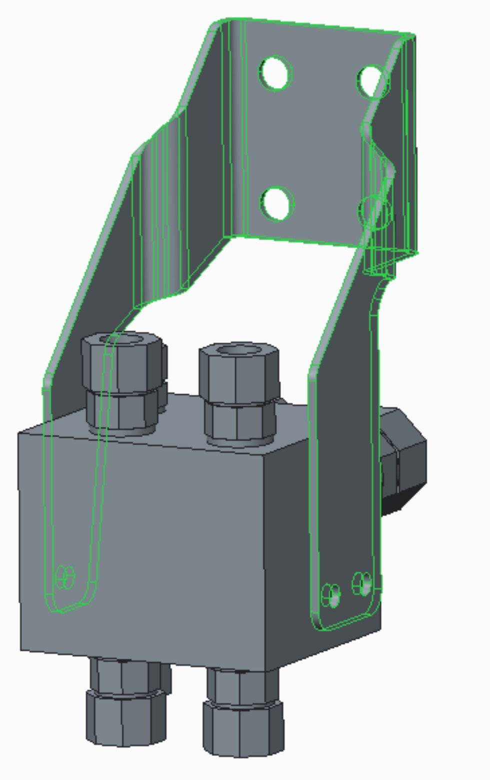


Appendix OO Connection block system disengaged

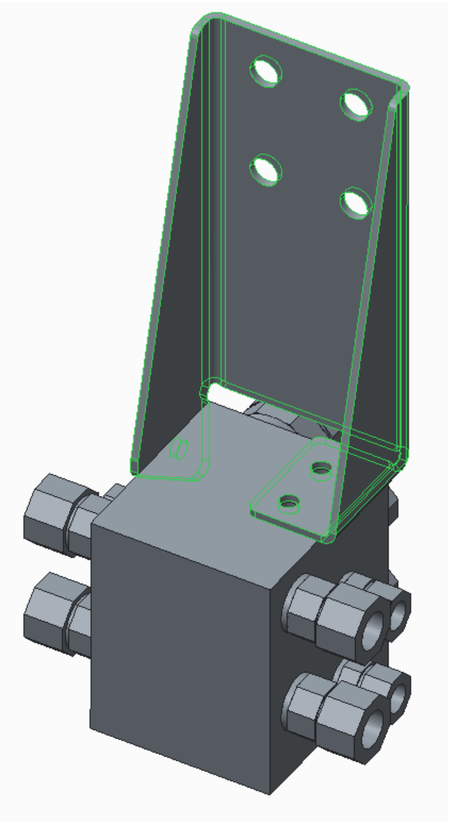
Appendix PP Connection block system engaged



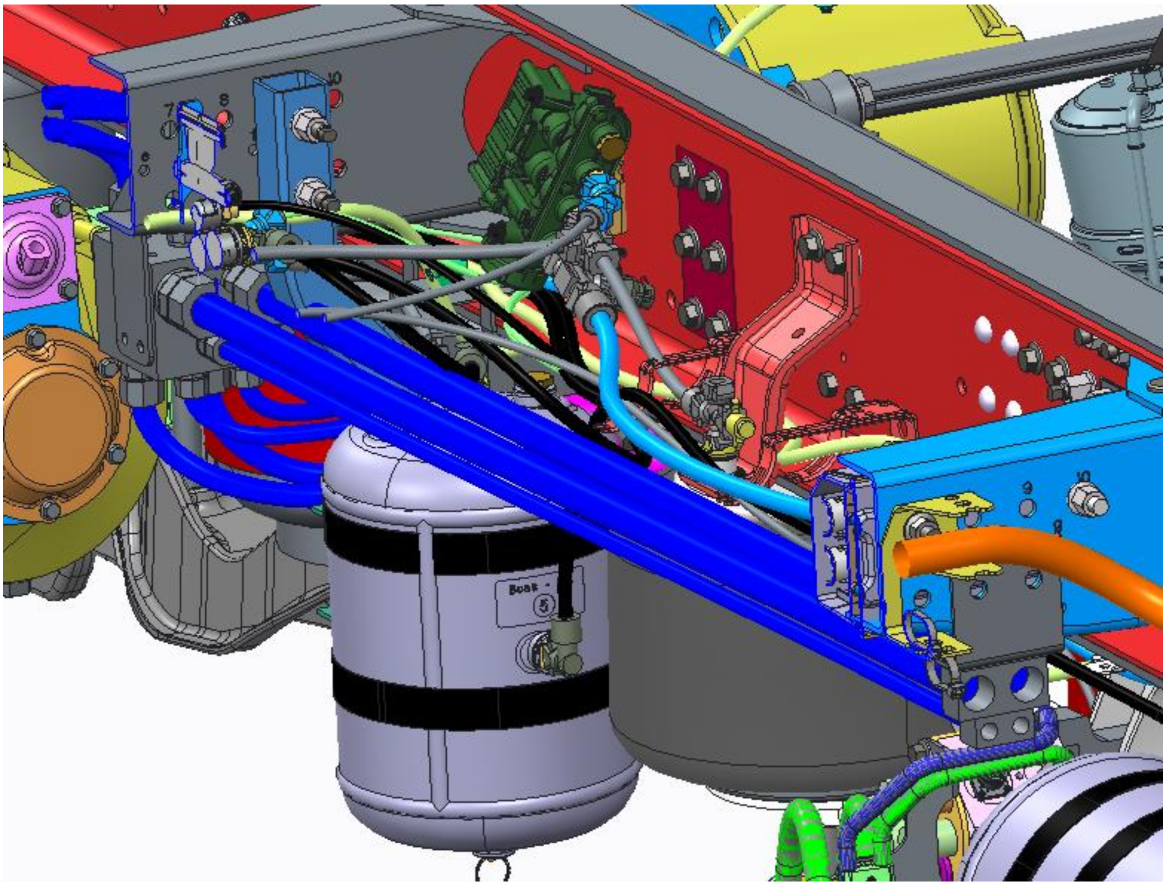
Appendix QQ Connection block RAPDD-GR



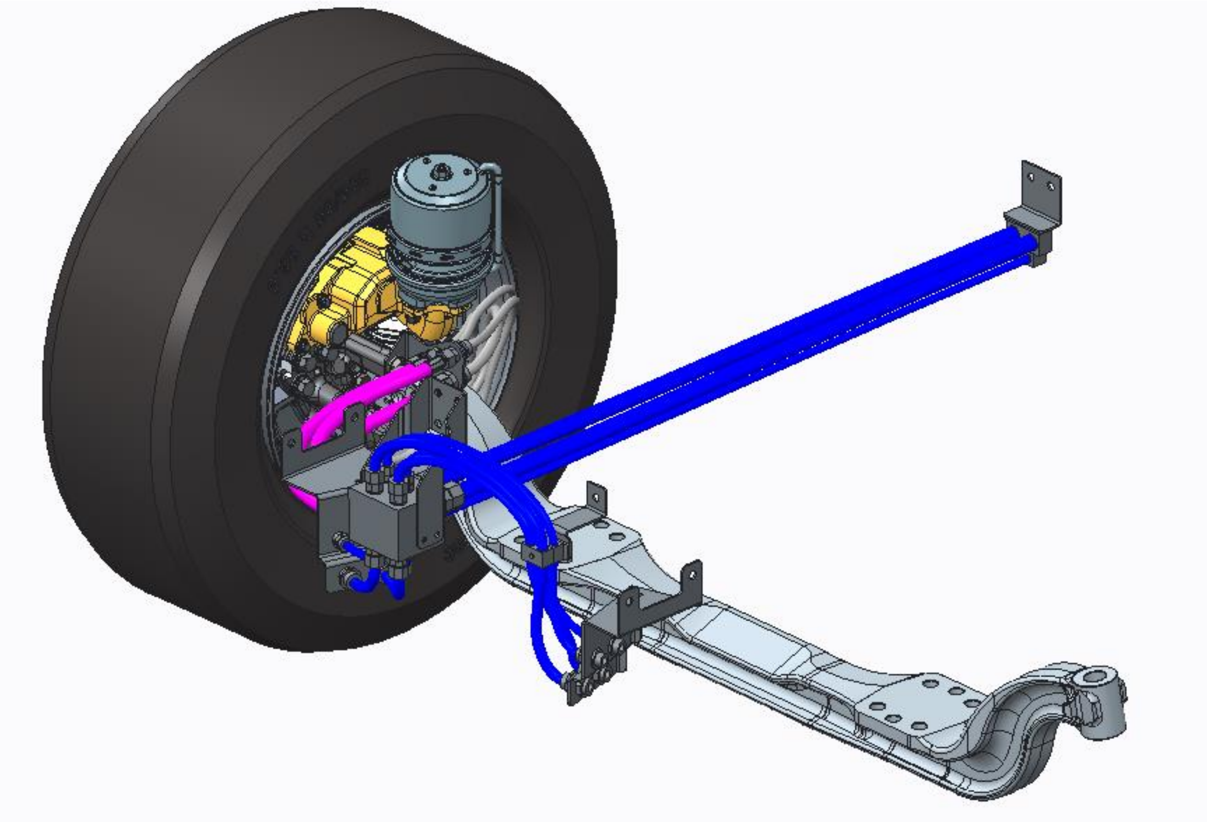
Appendix RR Connection block RADT-GR

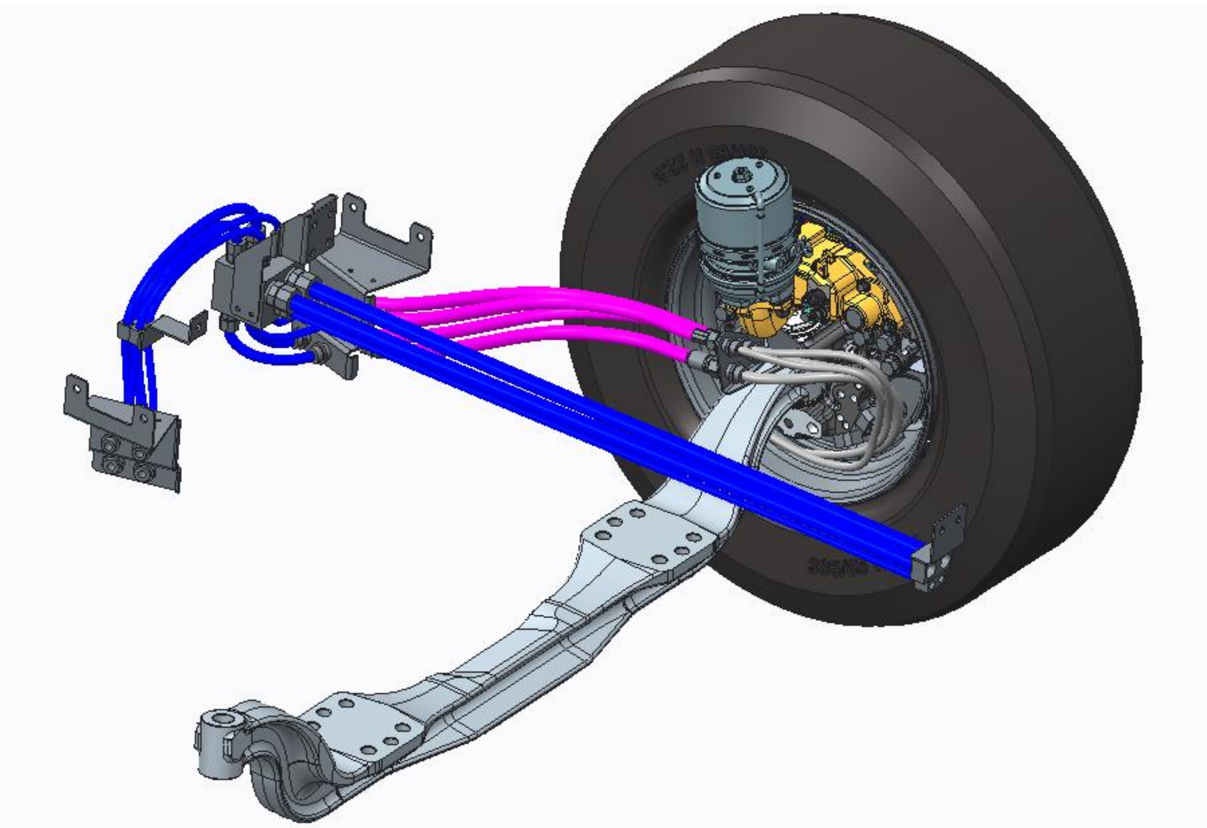


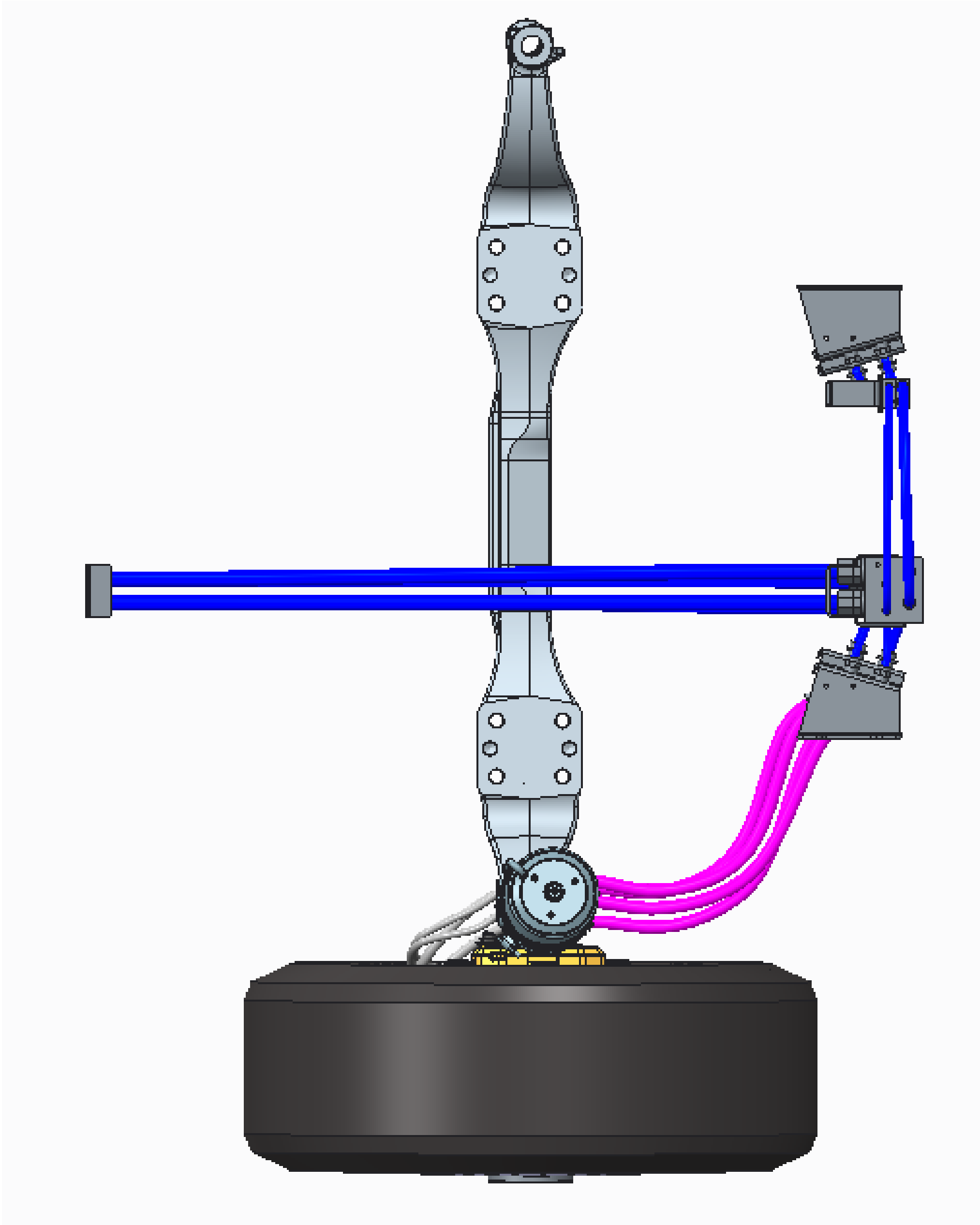
Appendix SS Tank 21944726



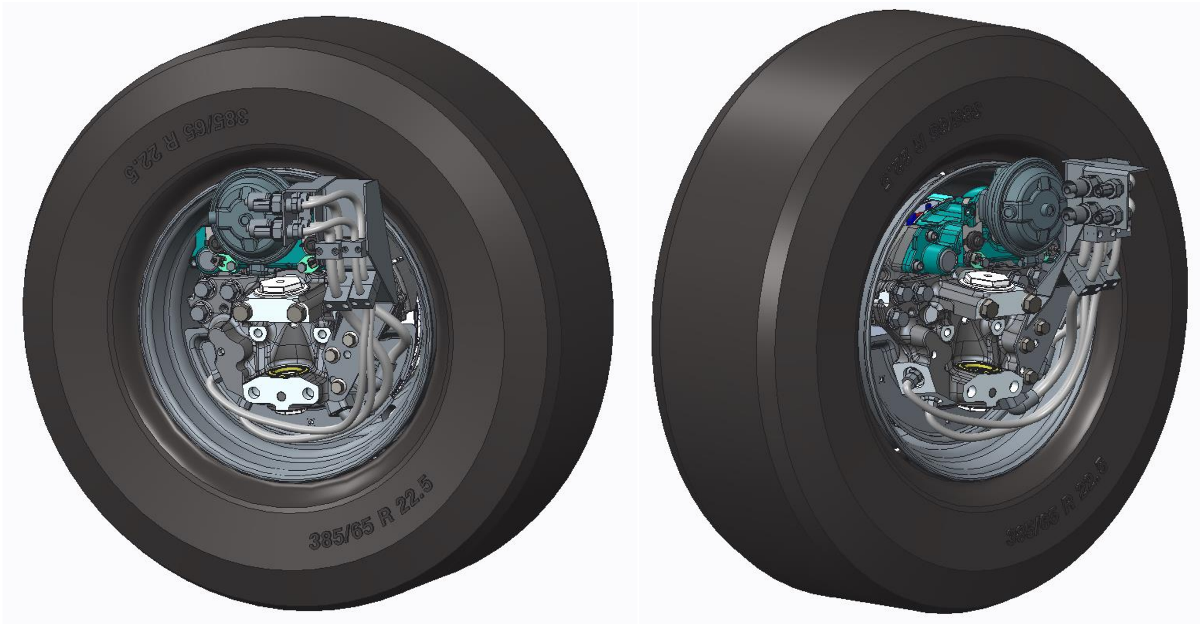
Appendix TT Routing RAPDD-GR



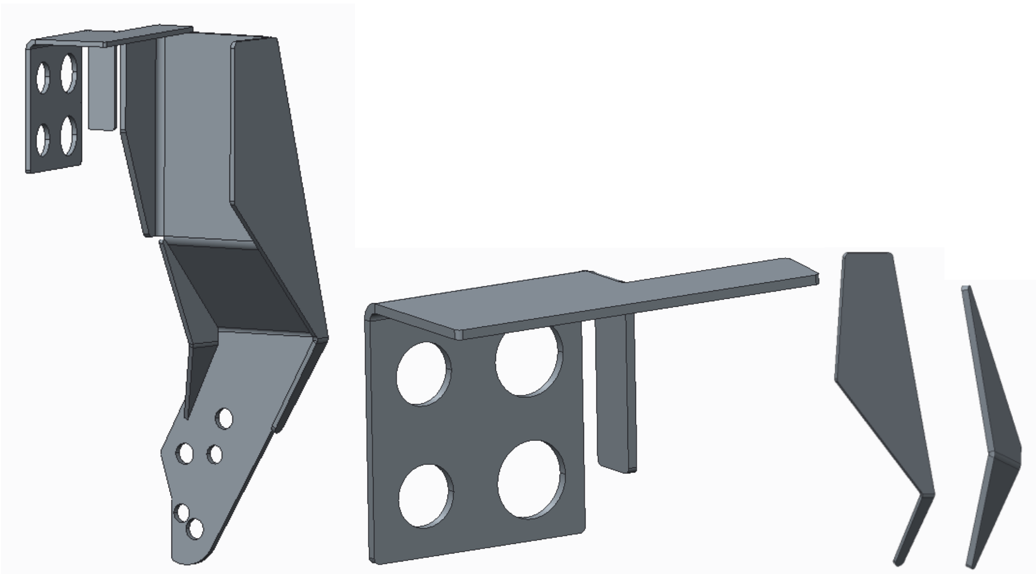




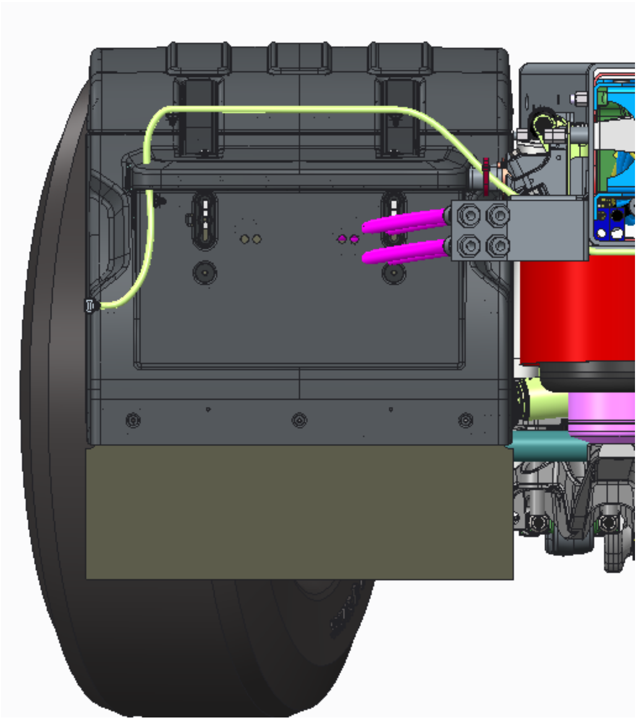
Appendix UU Routing RADT-GR wheel end



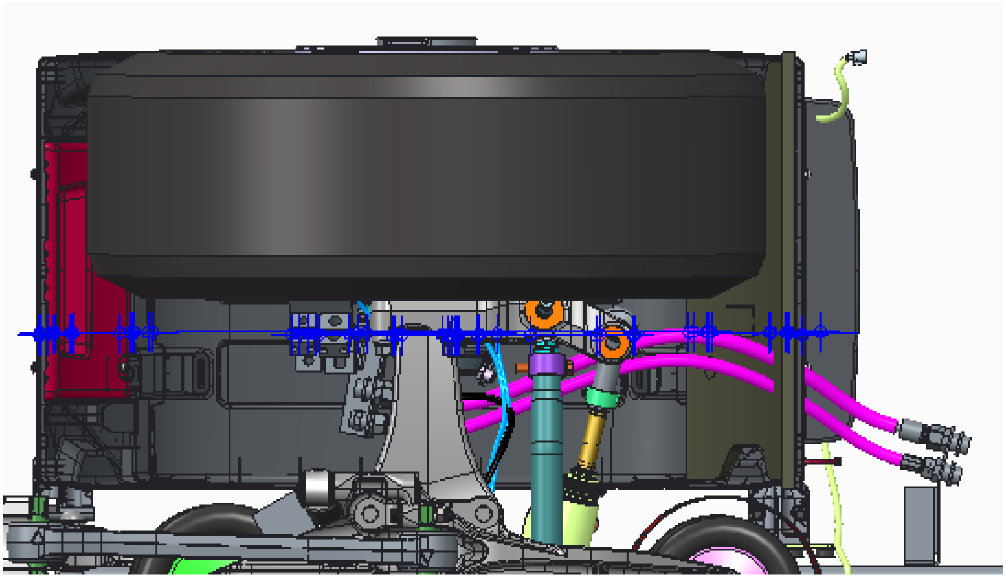
Appendix VV Bracket RADT-GR wheel end



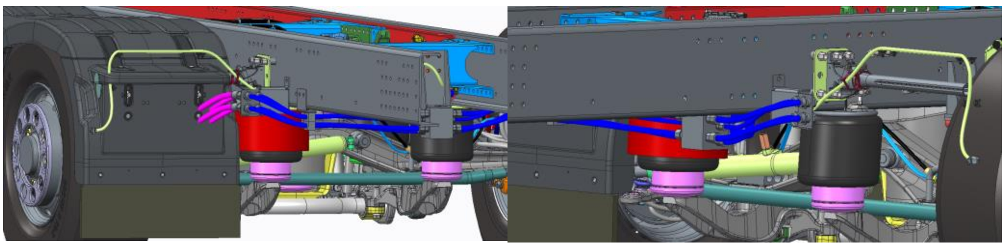
Appendix WW Mudflap clash RADT-GR



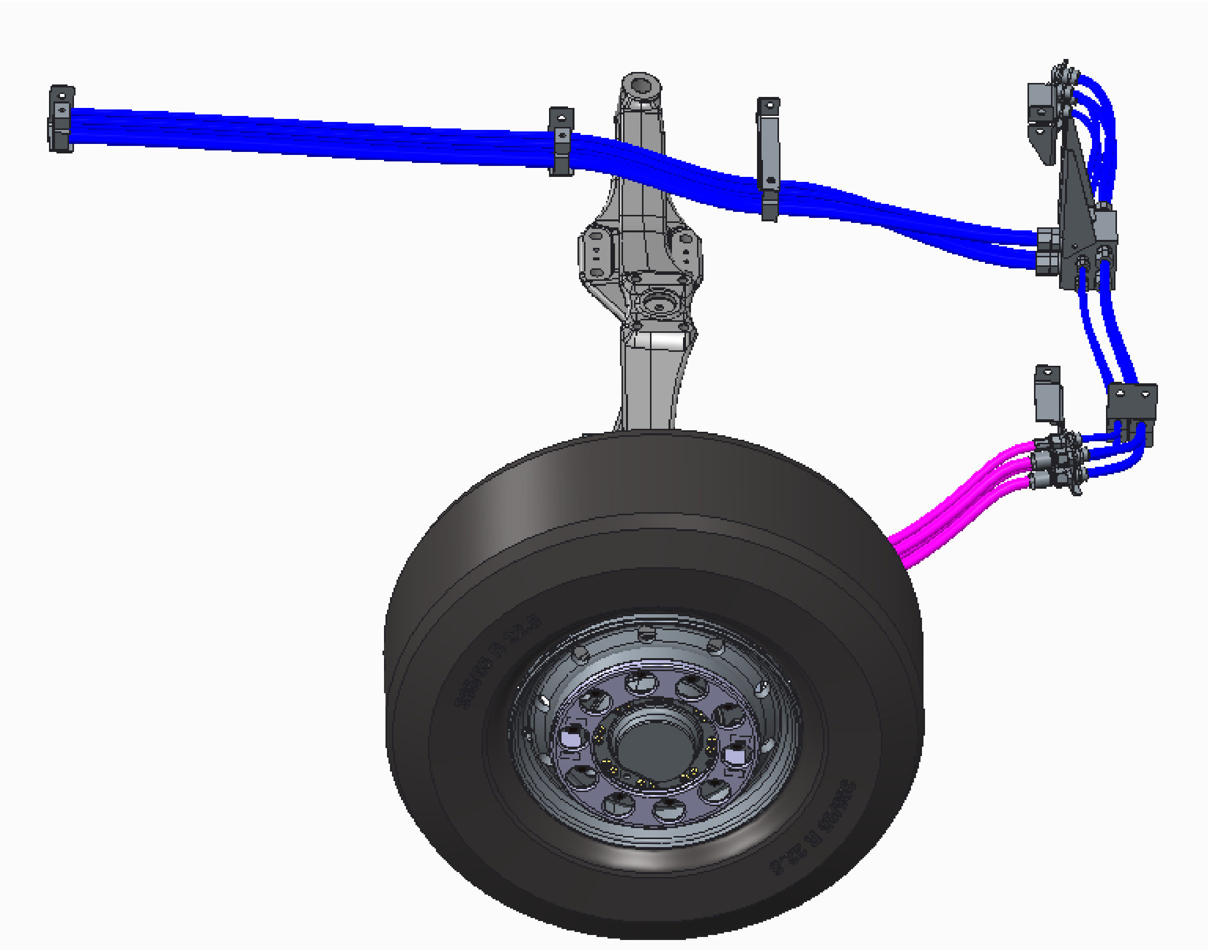
Appendix XX 60 mm free distance

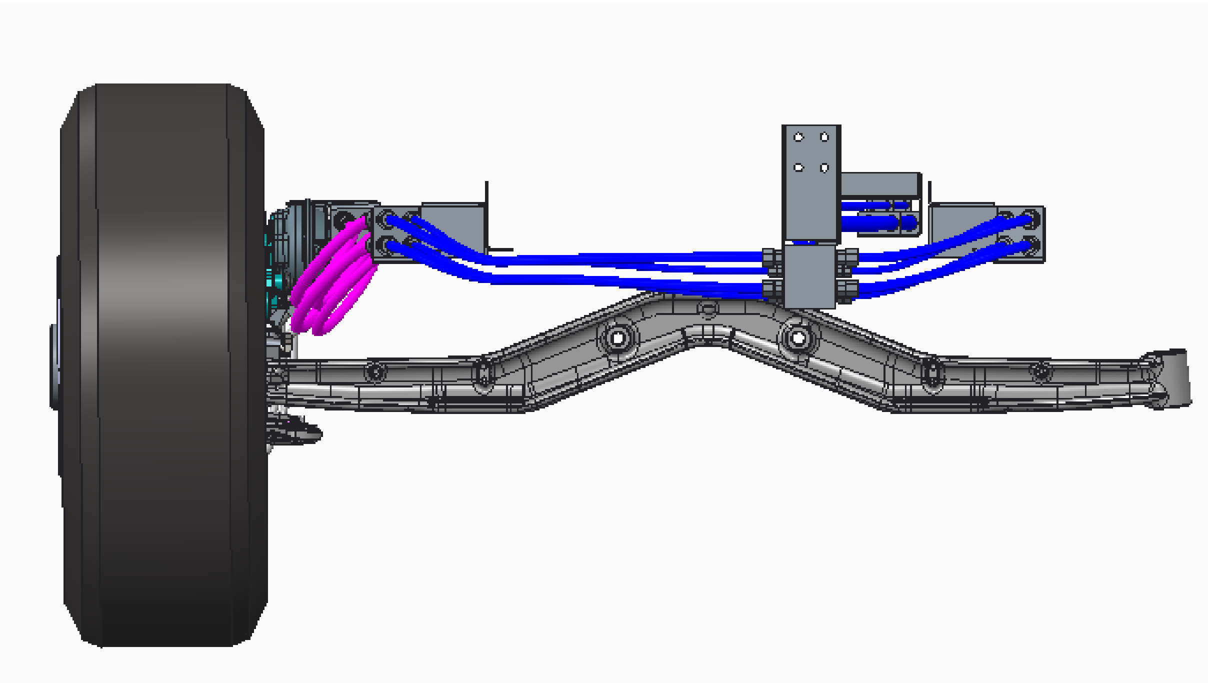


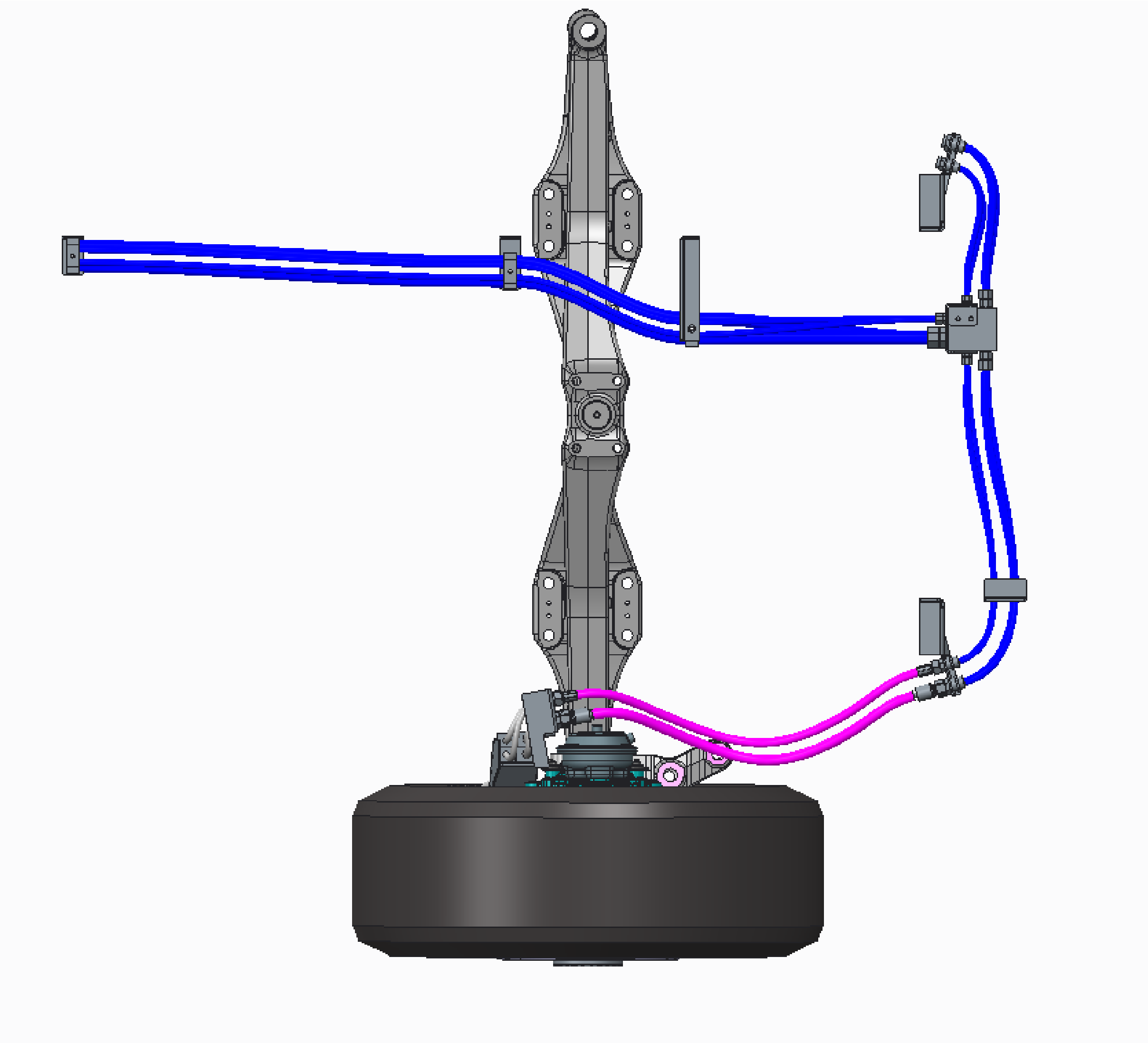
Appendix YY Left and right brackets RADT-GR



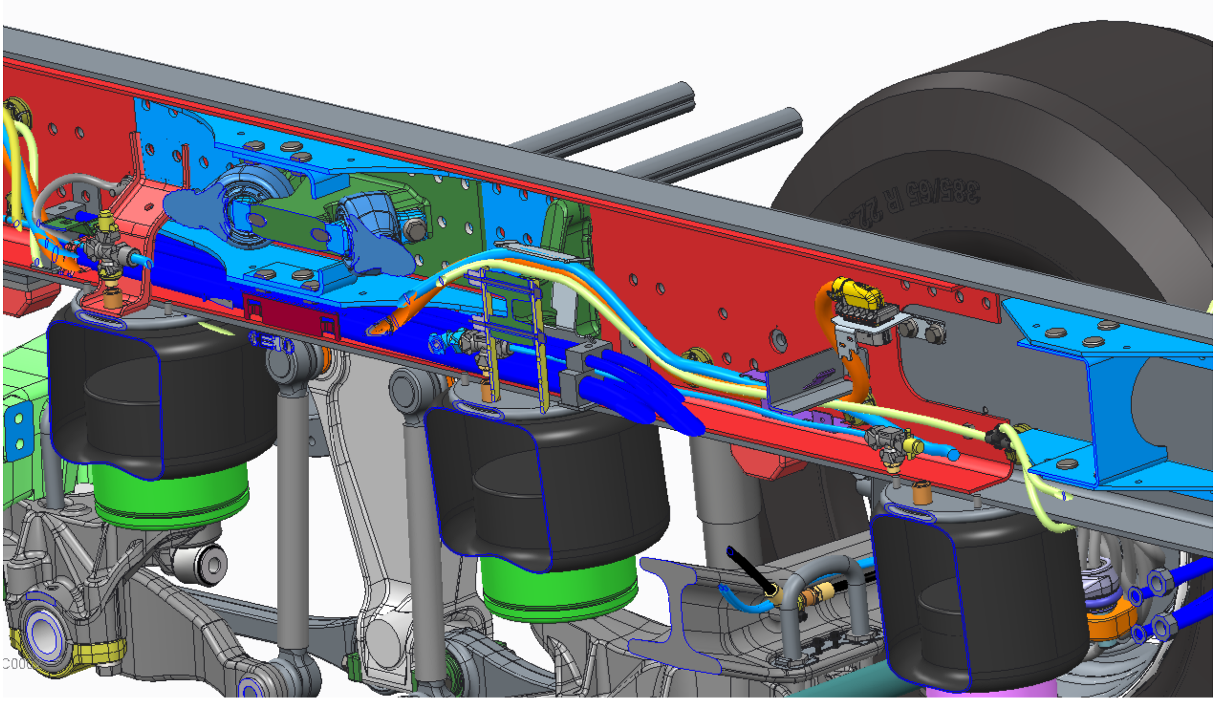
Appendix ZZ RADT-GR routing overview



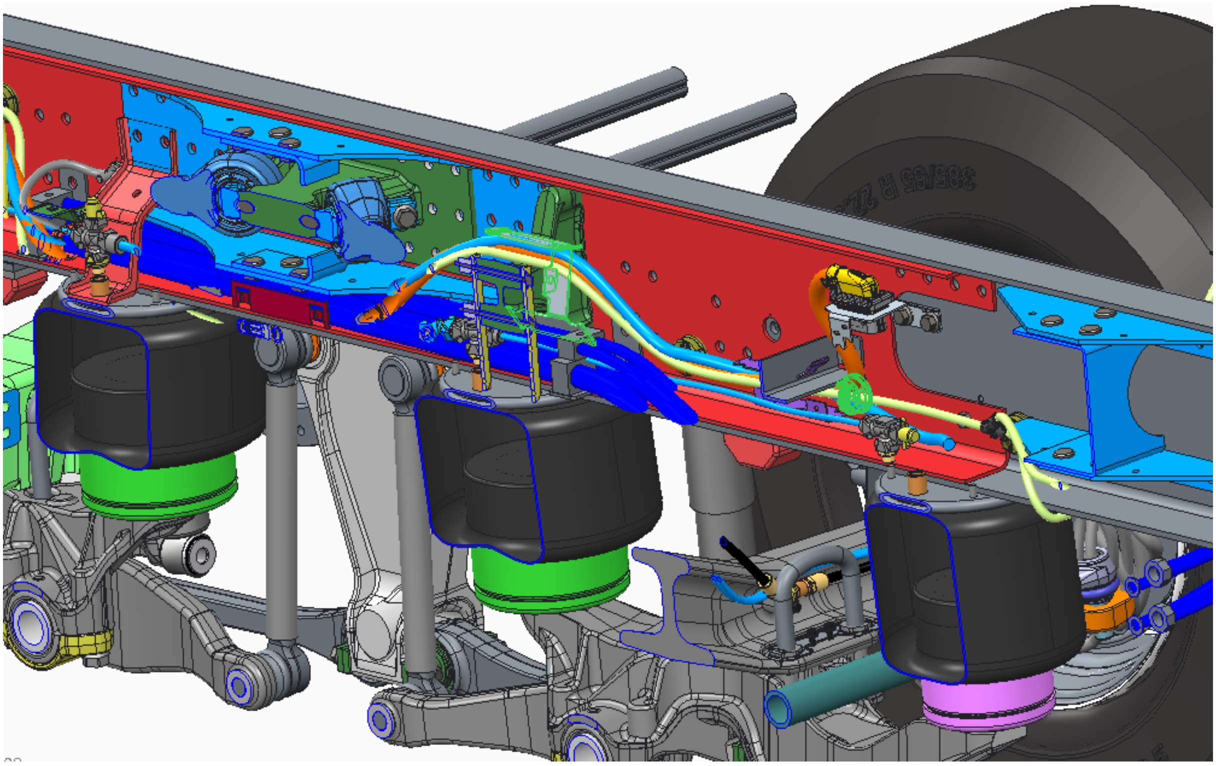




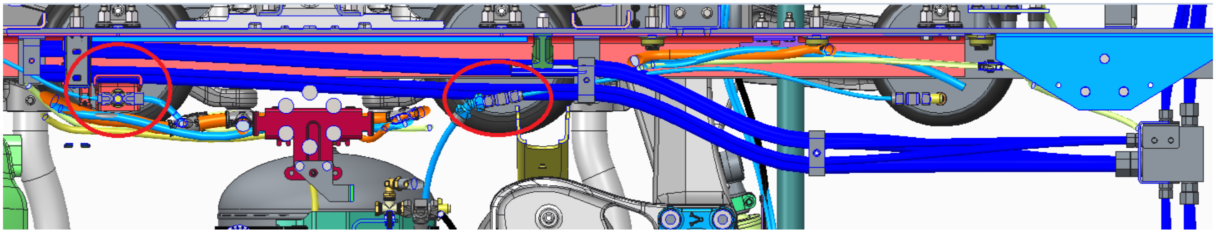
Appendix AAA Section view 1



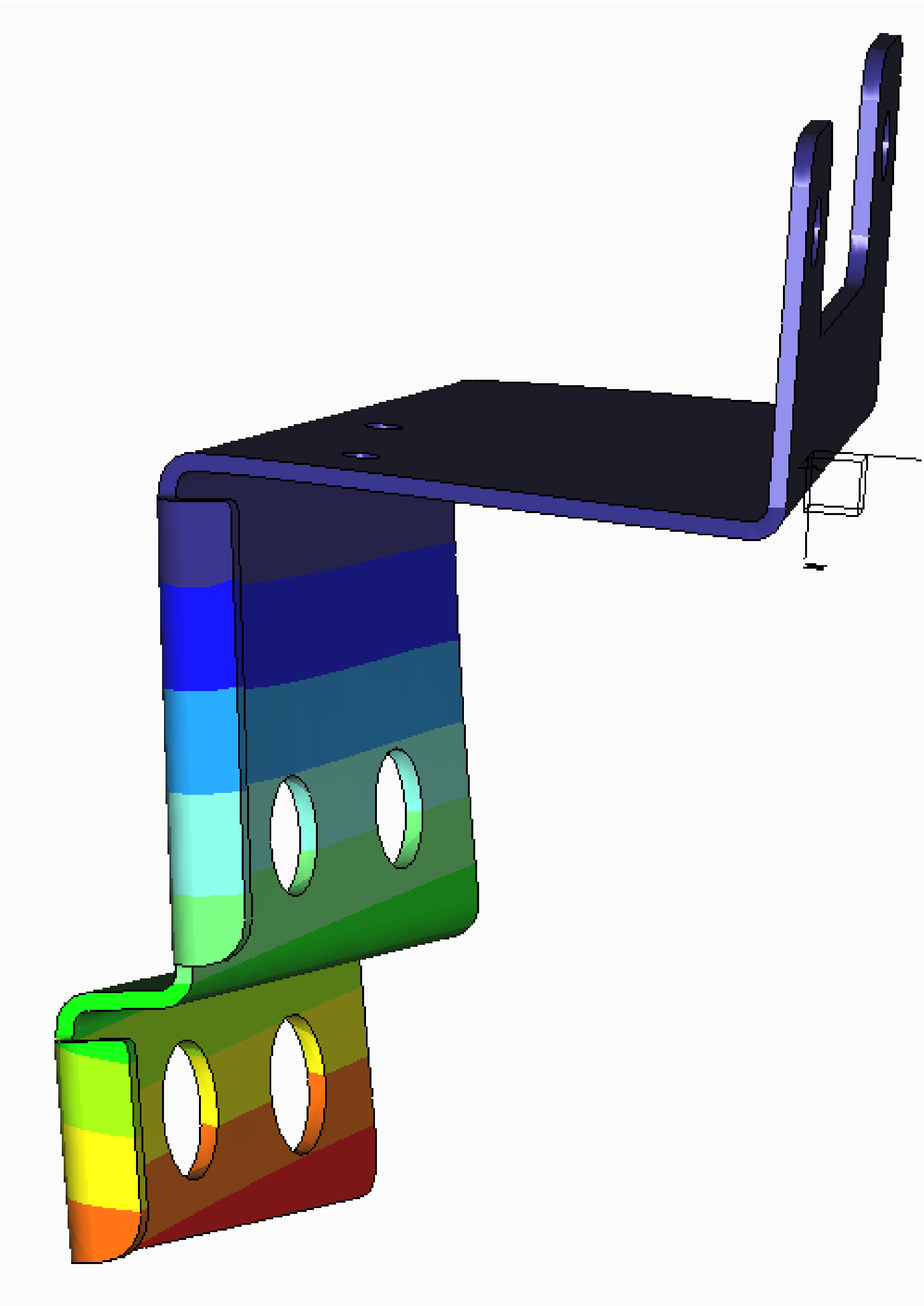
Appendix BBB Section view 2



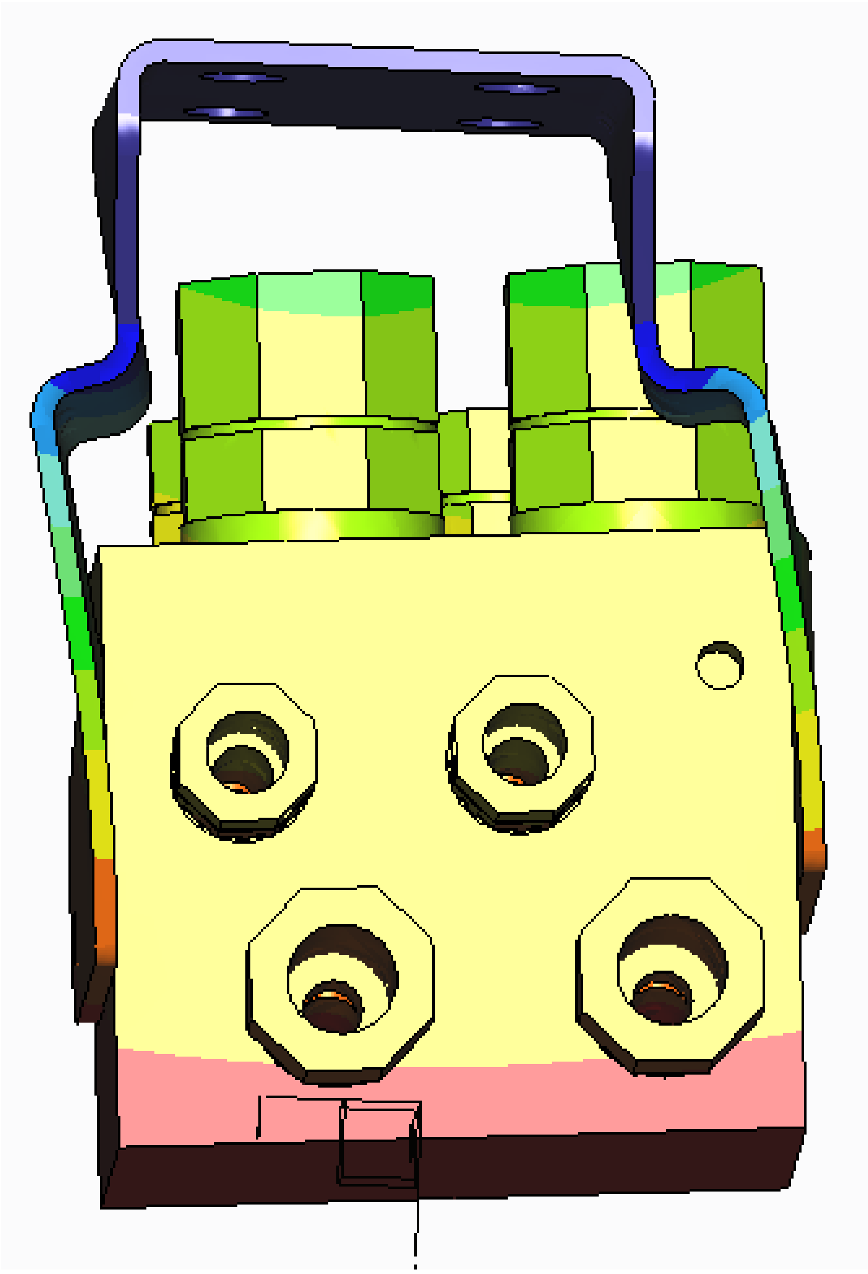
Appendix CCC Needed bends



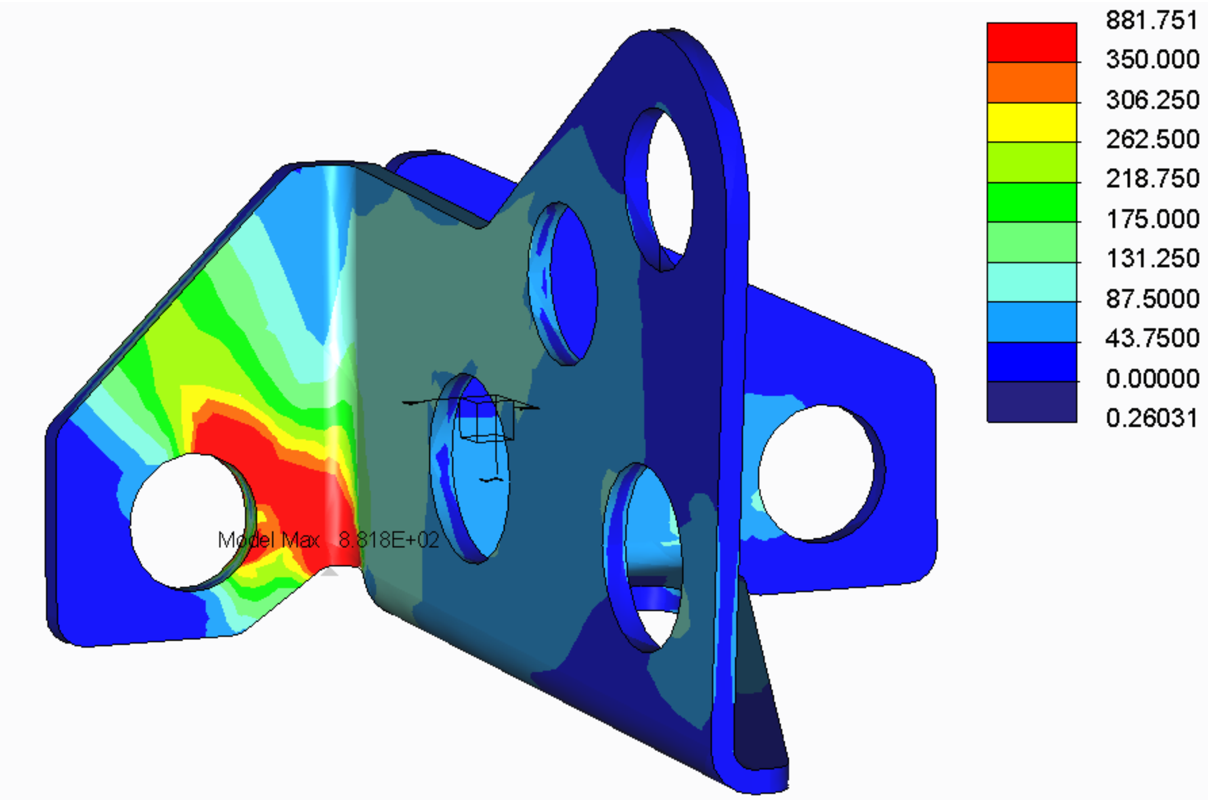
Appendix DDD A208649-bracketchassirapdd2_1



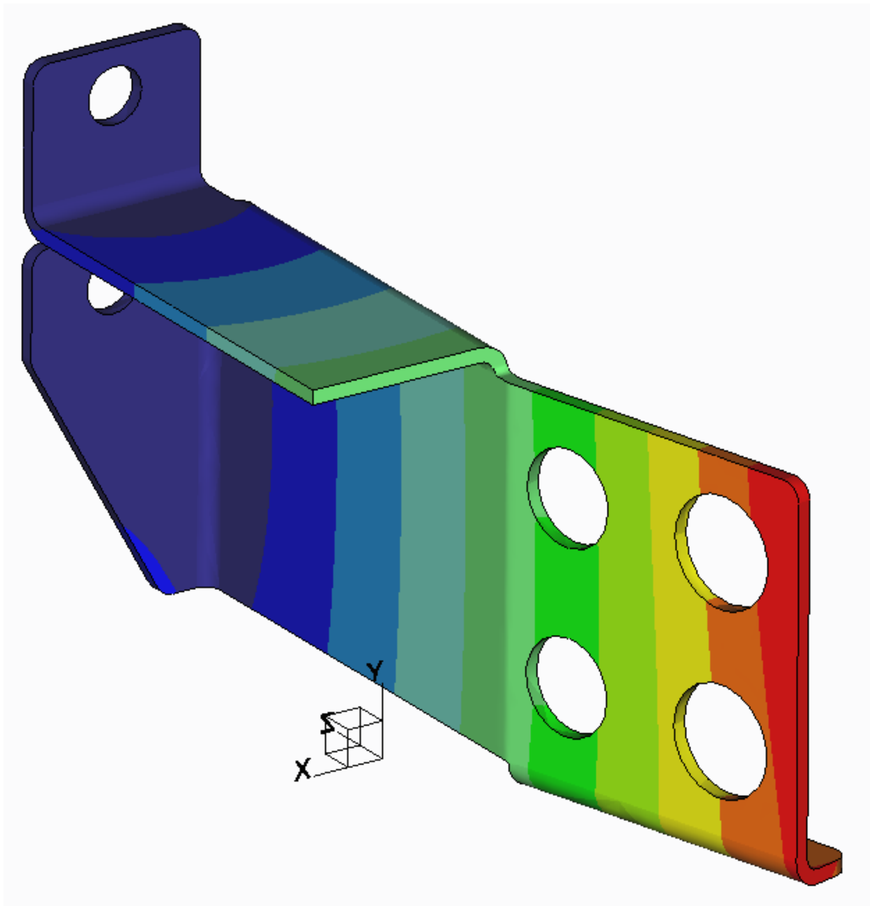
Appendix EEE A208649_bracketcrossbeamrapdd2



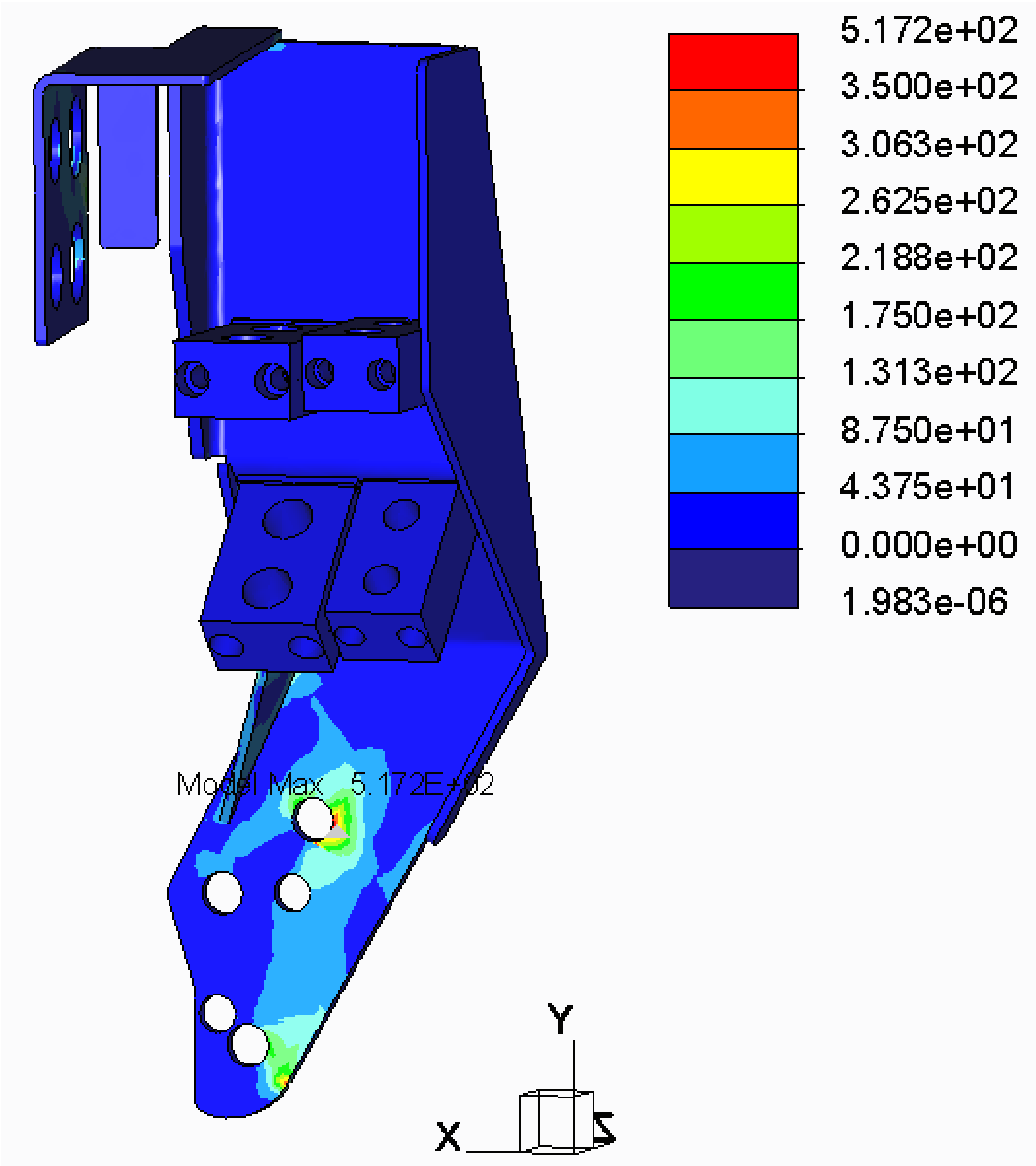
Appendix FFF A208649_bracketauxspindlerapdd2 z-direction



Appendix GGG A208649_bracketchassiradt3_2



Appendix HHH A208649_bracketspindleradt x-direction



Appendix III A208649_bracketspindleradt y-direction