Design and Development of a Multifunctional Test Rig

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Foreword

This Master thesis is the final part of our degree, Master of Science in Mechanical Engineering, carried out at The University of Lund. The thesis describes a product development process executed in cooperation with Faiveley Transport Nordic AB in Landskrona, Sweden.

A lot of people have helped us throughout the project and without them the result would not have been as satisfactory. First of all we would like to thank our supervisor at Faiveley, Fredrik Blennow, whom has guided us with high interest and given us great support. Furthermore we would like to thank Per Persson, Magnus Carlsson and Fredrik Nilsson at Faiveley's lab department. They have answered questions of various natures with never ending enthusiasm and been very helpful.

We would also like to give recognition to Andreas Arnell, Tobias Persson and their colleagues at Product Development at Faiveley. They have given us valuable information and inputs throughout the project.

Finally we would like to thank Kenneth Jeppsson at Ingenjörsfirma Jeppsson AB, Ystad for sharing his vast knowledge in manufacturing.

Lund, January 2010

Mikael Lindström & Johan Stridh

Abstract

In this project, the several steps in a product development process can be followed, from the first brainstorming of basic concepts to the final implementation of the manufactured product in the factory.

The project was assigned by Faiveley Transport Nordic AB and its aim was to design a well functioning test rig for testing of their train brake units. The new rig's advantages compared to old existing test rigs at Faiveley, is that it should be compact, flexible and able to test multiple train brake units at the same time.

Throughout the project the methodology of Ullrich and Eppinger's "Product Design and Development" was used at a large extent. As a first step in this methodology, target specifications were set and thereafter the concept generation could start. The designing of the test rig was divided into sub problems to be solved separately and after several iterations a final design was found. To make sure the test rig was dimensioned in a satisfying way comprehensive calculations were carried out, e.g. ANSYS calculations.

After the supervisors at Faiveley approved the design it was manufactured by the company Ingenjörsfirma Jeppsson AB. When the test rig was delivered careful testing took place. The results were very positive, all components functioned as wished and the test rig responded well when applied to forces.

As Faiveley wanted a new pneumatic system to drive the train brakes, this was ordered by Festo. It consisted of one control unit and ten valve units in a terminal making the device very compact. A casing was designed and manufactured to protect the sensitive equipment.

Finally the target specifications were compared to those of the actual test rig. All specifications were found satisfactory and the project was considered successful.

KEYWORDS

Test rig
Product design and development
Faiveley Transport Nordic AB
Pro Engineer
Structural analysis
Pneumatics

Sammanfattning

I de flesta företag där man tillverkar och säljer fysiska produkter är innovation och produktutveckling av största vikt. För att bli framgångsrik måste man ha spetskunskaper inom sitt område men även förfoga över bra utrustning och utrymmen för att utföra tester av befintliga samt nya produkter. Det är oerhört viktigt att dessa verktyg har en väldefinierad uppdragsformulering samt att de kan utföra dessa uppdrag med ett bra resultat. I det här examensarbetet, utfärdat av Faiveley Transport Nordic AB, har huvuduppgiften varit att tillverka en testrigg för utmattningstest av tågbromsar. Anledningen till att man vill ha en ny rigg är att befintliga riggar inte klarar testa ett större antal bromsar åt gången och kräver, trots denna brist, ett stort utrymme i testlabbet. Den nya testriggen vill man göra så kompakt som möjlig med möjlighet att testa ett flertal tågbromsar åt gången.

Examensarbetet utfördes av Mikael Lindström och Johan Stridh som en avslutande del i civilingenjörsutbildningen på LTH inom maskinteknik med inriktning mot produktutveckling.

Faiveley Transport Nordic AB har sitt kontor och även sin fabrik i Landskrona. Deras huvudprodukt är BFC-bromsar (Brake Friction Concept) som används i tåg men de tillverkar även andra relaterade produkter. Företaget hette tidigare SAB Wabco men blev 2004 uppköpt av det franska företaget Faiveley Transport. Faiveley Transport är ett globalt företag som har många olika tågrelaterade produkter i sin portfölj. Enheten i Landskrona hade år 2008 133 medarbetare samt en omsättning på 278 miljoner kronor.

Handledare på Faiveley var Product Engineering Manager Fredrik Blennow som tillsammans med sina medarbetare hade utformat en uppdragsformulering som beskrev vilka egenskaper testriggen skulle uppfylla. Denna uppdragsformulering användes sedan för att fastställa restriktioner samt en målsättning med projektet. Exempel på dessa specifikationer från Faiveley var:

- Antal bromsar som ska testas samtidigt
- Maximal deformation vid belastning
- Möjlighet att välja en viss elasticitet
- Lista på bromsar med olika egenskaper som skulle kunna testas
- Kostnad

Det bestämdes att Ullrich & Eppingers metodik för produktutveckling, som går att finna i boken ”Product Design and Development”, skulle följas i största möjliga mån. Under hela projektet användes dessutom kunskaper och kursmaterial som erhållits under fyra år på LTH för att lösa uppkomna problem. Efter en diskussion med

Faiveley bestämdes det att ProEngineer Wildfire 3 skulle användas för att skapa 3D modeller och ritningar. Ett annat program som användes mycket under projektet var ANSYS Workbench, detta användes för att göra FEM-analyserna på riggen.

Det första steget i arbetet var att bekanta sig med de olika bromsar som ingick i projektet. Mått, tyngd, bromskraft och obstruktioner för alla bromsar noterades. Dessutom studerades i detalj hur dagens tester genomfördes.

När grundläggande förståelse för den önskade produkten var uppnådd började identifieringen av kundbehov. Uppdragsformuleringen fick av naturliga skäl stor påverkan då denna identifierade många krav på riggen. Men utöver denna fördes utförliga diskussioner med labbteknikerna om fördelar och nackdelar med de gamla testriggarna. Data om utrymmen som krävdes, vilka hjälpverktyg de använde och förbättringsförslag antecknades också. Med hjälp av den insamlade informationen samt en undersökning av de befintliga testriggarna kunde slutligen målspecifikationer för den nya testriggen fastställas.

När målspecifikationerna var bestämda började konstruktionsfasen av projektet. Den kan bäst beskrivas som en iterativ process där riggens olika komponenter delades upp i delproblem. Efter att ha genererat ett antal olika förslag på hur testriggens grundstruktur skulle se ut, dvs. antal och typ av stationer samt deras inbördes förhållande, valdes slutligen ett förslag genom utvärderingar, så kallade *Concept screening* och *Concept scoring*. Under denna process hölls upprepade möten med F. Blennow där förslag diskuterades, men där även vissa krav på testriggen förändrades. Det bestämdes efterhand att testriggen bara behövde innehålla fyra stationer istället för fem samt att det inte skulle finnas en station för enbart bromsar med långa spindlar.

När grundstrukturen var fastställd kunde övriga delproblem lösas. Utöver testriggens benställning skapades eller justerades även mer eller mindre komplexa saker så som avståndsmått, lösning för hur mätutrustningen skulle fixeras, fixturer för att fästa bromsar, stödbitar och monteringshål. Efterhand som projektet fortskred ritades komponenterna upp och sammanställdes i ProEngineer så att en tydlig överblick över testriggen kunde erhållas.

När hela konstruktionen till sist var färdig påbörjades beräkningsdelen. I uppdragsformuleringen var det fastställt att testriggen skulle klara utmattningstester av bromsar med en bromskraft på 70 kN. Då detta examensarbete inte var inriktat på beräkning var det tvunget att göra vissa begränsningar och det bestämdes att fokusera på de delar av testriggen som bedömdes vara mest utsatta. Det som undersöktes var deformationer samt spänningar med hjälp av ANSYS, risk för utmattningsbrott och slutligen krafternas storlek i skruvförbanden.

I ANSYS-beräkningarna kunde det fastställas att deformationerna samt spänningarna klarade de uppsatta säkerhetsmarginalerna. Men då det blev ännu bättre resultat med tjockare plåtar samtidigt som kostnadsskillnaden var försumbar bestämdes det att byta till de tjockare plåtarna. Skruvförbandsberäkningarna visade också bra resultat men vid handberäkningarna för utmattningsbrott blev spänningen lite för hög vid ett av lastfallen. Då handberäkningarna ej tog hänsyn till alla förstyvningar av konstruktionen som skulle motverka just detta ansågs det inte vara ett problem. Dessutom visade samma lastfall i ANSYS, där förstyvningarna var med, en spänning mycket lägre än den högst tillåtna.

Efter att de ansvariga på Faiveley godkänt den slutliga konstruktionen av testriggen skickades en beställning till Ingenjörsfirma Jeppsson, som brukar utföra tillverkningsarbeten av den här typen åt Faiveley. När testriggen var tillverkad och levererad påbörjades en rad tester. Till en början testades generella saker för att säkerställa att alla komponenter var korrekt tillverkade och att alla konstruktions-lösningar fungerade som önskat. Därefter testades riggens utböjning och faktiska spänningar vid belastning. Samtliga tester gav bra resultat men det fanns naturligtvis anmärkningar. Det viktigaste som kom fram var att en korrekt montering av bromsarna är av yttersta vikt. För att inte få felaktiga värden vid mätning samt onödigt slitage av riggen måste bromskraften angripa helt i centrum på lastcellspaketets axel.

Innan projektet var avslutat ville Faiveley ha en ny pneumatisk lösning samt möjlighet att styra denna med befintliga LabView-program. En kompakt lösning bestående av en styrenhet och tio ventiler köptes av Festo. Dessa gick att styra utan problem efter lite programmering. Slutligen tillverkades en skyddsplåt så att pneumatiken och dess strömförsörjning kunde monteras på riggen utan risk för att skadas.

När samtliga delar av projektet var avslutade jämfördes testriggens mål-specifikationer med de slutliga specifikationerna. Resultatet var mycket till-fredställande och testriggen var därmed redo att tas i drift.

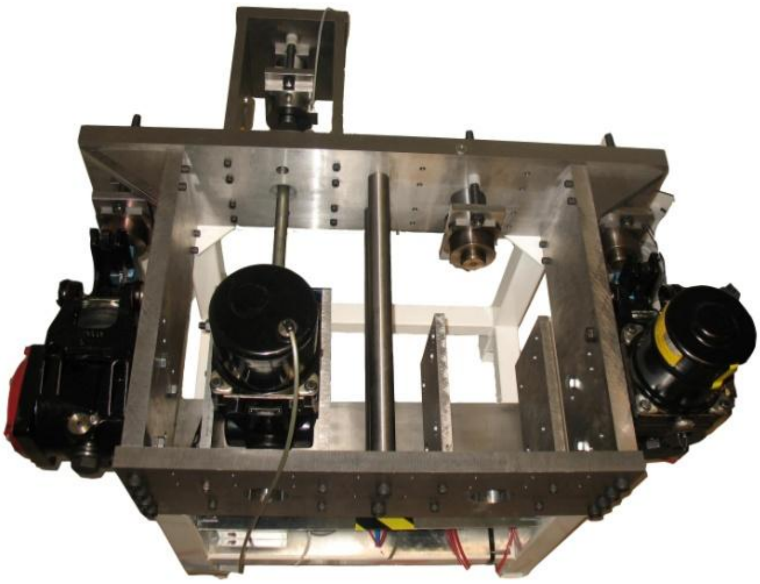

Bild 1 Testriggen med tre bromsar monterade.

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1 Introduction

1.1 Background

In most industries whose main business is to manufacture and sell products, development of these products is of great importance. To be successful in this, companies not only need experienced engineers but also a test department where it is possible to make diverse but still accurate analyzes of new and existing products. Depending on what kind of tests the companies need to perform they need different tools and machines. When investing in new equipment it is important that its purpose is determined beforehand and that it in the end fulfills this purpose.

At Faiveley Transport Nordic AB's (Faiveley's) test department, a variety of test rigs are used for different types of measurements. Some train brakes are tested to see how they can withstand vibration, some tests evaluate certain parts of a brake unit such as gaskets, springs and spindles. This project will show the development process of a test rig, which will be used in Faiveley Transport Nordic AB's test lab for endurance tests of train brakes, i.e. how a brake unit will function over time. Currently no test rig at Faiveley can handle more than two train brake units (TBUs) at once.

1.2 Problem description

When developing a new product there are several aspects to take into consideration. In this project the final product has to satisfy the specifications set by Faiveley. The test rig that is to be developed in this project is supposed to perform endurance tests of up to five train brakes simultaneously, which existing rigs cannot carry out. When designing, the demand of flexibility has to be thought of throughout the process simultaneously as cost, performance, and ease of use have to be considered. To be sure no failure will occur due to fatigue or nominal stresses, comprehensive calculations have to be made. Finally a pneumatic system has to be implemented into the test rig.

1.3 About the company

Faiveley Transport Nordic AB is located in Landskrona. It was former known as SAB Wabco and the company was acquired by Faiveley Transport as late as 2004. Faiveley Transport is a worldwide supplier of systems and equipments for the railway industry and in their portfolio they for example have a large spectrum of different types of brakes. In Landskrona the main focus is on developing and manufacturing BFC (Brake Friction Concept) Tread Brake and Bogie Brake units. In 2008 Faiveley Transport Nordic AB had a turnover of 277.5 million SEK and 133 employees.

1.4 Project participants

The following persons were involved in completing this master thesis project:

Master thesis authors	**Supervisors at Faiveley**
Johan Stridh	M. Sc. Fredrik Blennow
Mikael Lindström	M. Sc. Andreas Arnell
	M. Sc. Tobias Persson
Supervisor at the University of Lund Division of Machine Design	**Examiner at the University of Lund Division of Machine Design**
Lecturer Per-Erik Andersson	Professor Robert Bjärnemo

2 Objectives

The aim of this project is to design and develop a test rig that fulfills Faiveley's demands. The test rig will be able to perform tests on multiple train brakes with full insurance that the results are accurate. The project will use a well proven methodology to carry out the product design and development process. This project will be carried out in an efficient way using knowledge acquired at LTH and the various expertises that exists at Faiveley. Where new unknown areas are run across, knowledge will be sought and studied until it can be applied to the issue at hand.

The designing and dimensioning of the test rig will be done considering common existing designing rules and guidelines. Also calculations and material analyses will be done accordingly in order to generate a low cost, multifunctional test rig.

The aim is that when the project comes to an end a fully working test rig that satisfies Faiveley's own objectives will be operating in the company's test lab. This includes the pneumatic controlling of the brakes and the necessary education of the personnel.

A few restrictions delimit the project, such as the project should be completed in early 2010 and the total cost of a new rig. These and a few more are mentioned in Appendix A, which contains all the objectives and restrictions set up by Faiveley prior to the start of this project.

3 Methodology

The project is a complete product design and development project which will span from early ideas and needs to a fully functional and tested product. The design and developing phase will be based on the methodology presented by K. Ullrich and S. Eppinger in their Product Design and Development [1] which is lectured at the Division of Machine Design at LTH.

The project will use many of the tools and tips provided by Ullrich & Eppinger as well as what has been taught during various courses at the faculty.

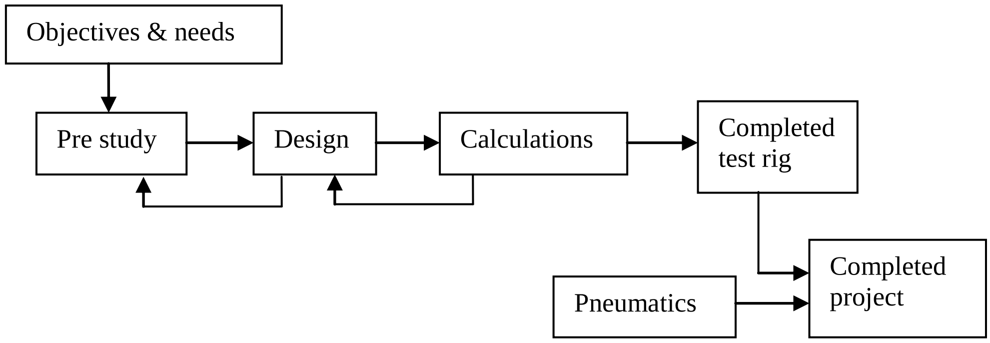
It is important that a good understanding of what Faiveley is expecting from this master thesis work and thus the actual test rig. Therefore sufficient time will be spent on truly identifying and understanding the needs stated by Faiveley. This will be done by studying how their test activities are functioning today.

When a solid foundation of information has been gathered the actual designing and developing according to the specified methodology will start. As steps of the process are completed commentaries will be provided to analyze and clarify what has been done.

The designing part of the project will be concluded with drawings of the complete test rig being sent to a manufacturer. A testing part will take effect as soon as the rig is delivered and then a full evaluation of both the test rig and the project will be made.

Parallel to the designing of the test rig a pneumatic system for running and controlling the brakes will be developed. This system will comprise entirely of already existing solutions and no new designing is needed. The work will focus on finding the suitable equipment by analyzing the various manufacturers’ products.

The procedure of this project is shown below where the iterative process of product development is noticeable.



4 Pre study

As a part of getting the project running a number of things had to be done. The first step was to get familiar with Faiveley, its products and test facilities. It was considered vital to fully analyze and understand the design of the products to be tested as well as the structure of the testing principles in use today.

4.1 The Tread Brake Unit

The most common type of train brake manufactured by Faiveley is the Tread Brake Unit (TBU). These come in a variety of sizes and versions mainly because each new brake is custom designed for the buyer's specific needs. A few things however, are alike for the different versions. Figure 4.1 below shows an overview of the inbound parts of a generalized TBU. The housing is the central structure that contains all the regulatory parts as well as the spindle that pushes the service brake and its brake shoes forward. On the housing there are a number of fixture holes to which the brake is joined to the train's structure. The service brake is pneumatically driven.

Normally a TBU also includes a parking brake (PB) that makes sure that the train does not move when it is parked. The PB is also pneumatically driven but in contrary to the service brake the braking will be applied even if the air pressure is lost. The only way to loosen the brake then is to pull the emergency release cable, either by hand or some manually controlled device.

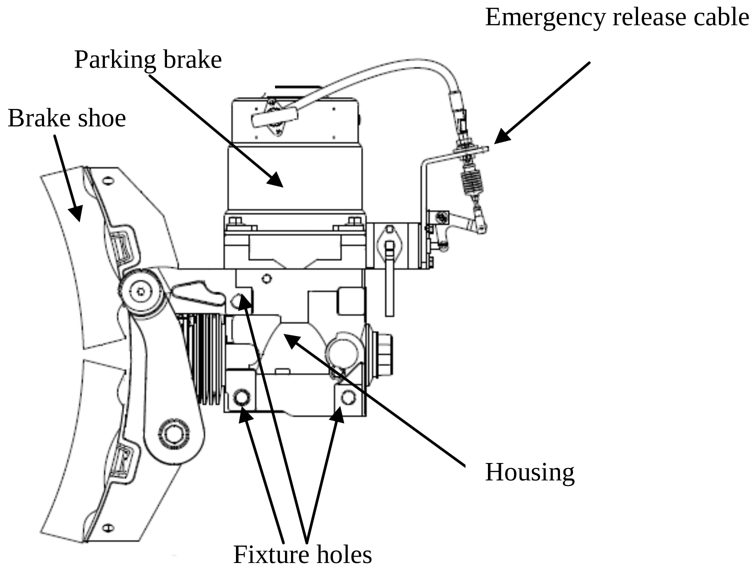
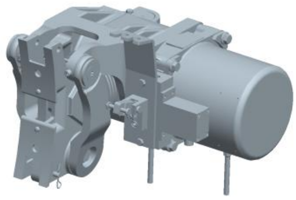


Figure 4.1 Components of a TBU.

The TBU in figure 4.1 shown on the previous side is one of the most common types. The parking brake extends upwards which makes the width for this TBU small. TBUs with PBs extending to the left or the right are considered bulky as they get a lot wider.

In addition to these there are long TBUs that do not have brake shoes or its mounting. These only have a long spindle running outwards from the housing. This spindle is attached to a structure on the trains that in its turn has a brake shoe attached to it which brakes the train. These spindles can get up to 1000 mm long. Figures of the different types of TBUs are shown in Table 4.1.

Table 4.1 Different types of TBUs.

TBU	TBU with bulky design	Long TBU
		

The brake force for all three types of TBUs is achieved by the wedge principle. The air pressure pushes the piston head down. The angle of the wedge determines how much the force amplifies in the brake direction as shown in Figure 4.2 below. On normal TBUs and the ones with bulky designs the rounded brake shoe is pushed directly against the trains' steel wheels slowing the trains down.

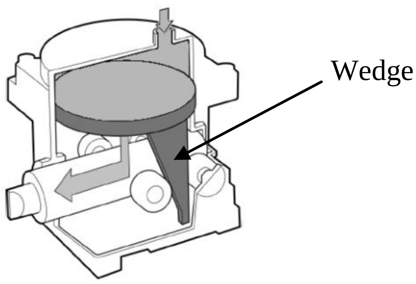


Figure 4.2 The TBU braking principle.

Prior to the start of the project nine different drawings of TBUs that Faiveley wanted to be tested on the new rig, were delivered. The actual drawings of the specific brakes will not be disclosed in this report as is the will of Faiveley. These nine TBUs were the ones that the new test rig primarily had to be able to handle. The first step in the chain of the developing process was to analyze these. The main factors that were to be determined from the drawings were the following;

- Weight of the TBUs.
- Number of holes for fixture.

- Position of holes.
- Positioning – is it mounted from the right, left or from below?
- Dimensions.
- Maximum and average spindle lengths.
- Output force from brake at service pressure.
- Does the TBU have a parking brake feature?
- Does it have any visual obstructions such as arms, air connections or any bulky parts, which will be in the way when mounting a TBU?

The results of this compilation can be seen in Appendix B. This helped to get a good view of how the brakes operate and how they are mounted. The next step was to analyze how tests on the TBUs are done today.

4.1.1 How today's tests are conducted

A standard endurance test is carried out as follows, see Figure 4.3 for a schematic layout as seen from above. The TBU (1) is fixed to the rig's structure (2). The test sensor (3) is mounted to the rig also but it can in general be moved lengthwise to its sought position. The spindle of the TBU (4) moves back and forth when the brake is applied. On any TBU the length of the spindle can be altered. Some brakes operate in its minimal length and some in its maximum. However most TBUs and subsequently most tests will be done with the spindles extended to its normal length. If required the length of the spindle can be adjusted by the adjustment screw (5) on the back of the TBU. The pneumatic cabling is connected to the valves (A – D) depending on if it is the service brake or the parking brake that is to be tested.

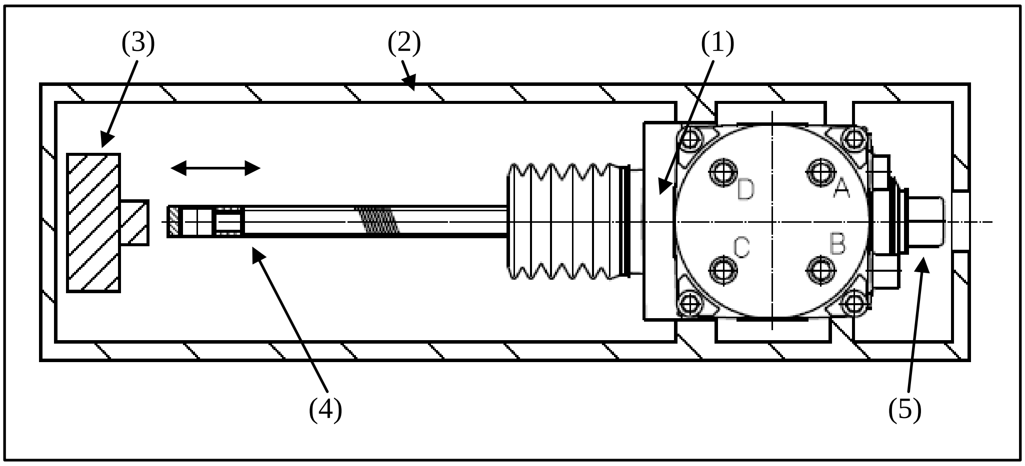


Figure 4.3 Schematic figure of a test of a long TBU.

At the beginning of a test a real test sensor is used to get values of the TBUs braking force. In order to spare the sensors they are replaced by dummies when measurements are not taken. Then in the end of a test when new values are gathered the real sensors are mounted to the rig again. In an endurance test the brake is applied and loosened in about 500 000 cycles.

Other tests that are to be carried out on the new rig are tests on the parking brakes and their emergency release mechanisms and also tests of hysteresis. In the hysteresis tests the braking force is measure throughout one braking cycle to determine how much the force varies within a cycle during loading and unloading.

The next step was to begin the actual product development phase as according to Ullrich & Eppinger.

4.2 Identifying needs for the new test rig

Before any designing of a new product can be done it is vital that everything is crystal clear regarding the various aspects and functions of the new test rig; what it should do, what it would look like etc. A list of objectives and restrictions (see Appendix A) regarding the test rig was set up by the company as a basis for designing the rig. These objectives and restrictions were interpreted as needs, see Table 4.2, which will later be translated into specifications with a certain metric as according to the methodology.

Table 4.2 Description of needs.

Need
The test rig is capable of running five simultaneous TBU tests.
The test rig handles TBUs with various shapes, e.g. with parking brakes.
The test rig includes one station with hydraulic connection.
The test rig is capable of testing nine different predefined TBUs.
The test rig is rigid.
The test rig provides an option to simulate elasticity.
The test rig is designed to resist fatigue.
The test rig allows space for test sensors.
The test rig provides space for additional air cylinders used for test of emergency release of parking brakes.
The test rig is able to test future TBU models.
The test rig is designed in a cost efficient way.
The test rig is movable.
The test rig fits within and can be moved about in the company’s lab facility.
The test rig (without TBUs) is light enough to be lifted by the existing forklift truck.

Commentary

Some of the needs presented in Table 4.2 must be further explained while others such as "*The test rig is movable*" speak for them self.

In reality the brakes, brake shoes and its mounting flex a little when the braking is applied to the wheel. Hence the company wants to be able to simulate these real conditions in the test rig. Therefore the rig is to be capable of measuring the function of the brakes in tests with either elastic or non elastic stops.

Some TBUs include a parking brake. The parking brakes must also function intentionally even after a long period of usage. Therefore there must be enough room on and around the test rig for the required air cylinders to be mounted on. These cylinders are relatively small and its connections flexible but nonetheless it must be considered when designing the rig.

As described in section 4.1.1 today's tests are conducted with the load sensors being in front of the brakes in the braking direction. One of the needs simply state that there has to be enough room for these sensors on the new rig.

Because the test rig is to be used in a laboratory environment it felt important to get the lab technicians opinions on a new test rig. It was felt that their experience in matters such as mounting TBUs, safety and accessibility were of great importance. Therefore a meeting was held on September 2nd 2009[1] to discuss the various aspects of the test rig.

The statements made by the lab technicians were interpreted into needs and are presented in Table 4.3. The statements are divided into groups in order to get a clear view of what was experienced as poor with today's test rig and what can be improved in a new one.

Table 4.3 Statements and needs from the lab technicians.

Question	Statement	Interpreted need
Typical uses	It is very important that there is plenty of space for connecting the pneumatics.	The test rig allows access for pneumatic connections.
	There must be enough space underneath the test rig for the movable crane which is used to load/unload TBUs.	The test rig allows access for a movable crane when loading/unloading TBUs.
Likes – current rig	Today we have fixtures that can handle different types of brakes.	The test rig uses flexible fixtures that permit mounting of various TBUs.
	If the TBU has a rear air connection an extra set of plates is used to mount the TBUs.	The test rig has a set of different plates.

Table 4.3 Continuing from previous page.

[1] Participating in this meeting were J. Stridh and M. Lindström and from the lab M. Carlsson and P. Persson.

	~~There is enough space around~~ the rig so we can easily and ergonomically work with the assembly.	~~The test rig allows an easy and~~ ergonomic mounting of TBUs.
	The existing rigs all have trays for storage of bolts, nuts and tools.	The test rig provides storage for tools, bolts and such.
Dislikes – current rig	We don't have enough space for screws and tools when assembling a TBU.	The test rig provides sufficient space between TBU and the rig for tool access.
	When testing TBUs with long spindles it normally takes about half a day to assemble due to changing of test rig components.	The test rig allows a easy and quick changing of TBUs with various spindle lengths.
Suggested improvements for the new rig	Alumec is preferred as a construction material due to its good properties in strength and density.	The test rig is constructed of a material with good characteristics regarding weight and strength.
	If a forklift truck is used for loading/unloading it must have enough clearance under the test rig.	The test rig allows access for a forklift truck when loading/unloading of TBUs.
	The new rig should have wheels of steel instead of today's plastic ones because of wear and instability.	The test rig uses components that are rigid and resilient to wear.
	Safety is important; we don't want to be at risk of being pinched by running TBUs while assembling another TBU.	The test rig provides a safe work environment.
	We want to be able to reach all areas of the rig easily.	The test rig allows access to all stations.
	We want to be able to quickly load/unload a new TBU.	The test rig allows for a quick change of TBUs when loading/unloading.

To determine which needs were of higher importance than others a survey was done in which the participants were asked to rank the different needs on a scale from one to five. On September 4th 2009 both R&D Manager Andreas Arnell and lab technician Per Persson were asked to fill out a form, see Appendix C. As a complement to these two another form was filled out, this one by Johan Stridh and Mikael Lindström in order to get an unbiased view of the different needs.

The importance of the needs was calculated as the mean value of the three different answers. The result is presented in Table 4.4.

Table 4.4 Complete list of needs and their importance.

No	Need	Importance
1	The test rig is capable of running five simultaneous TBU tests.	2.3
2	The test rig handles TBUs with various shapes, e.g. with parking brakes.	3.7
3	The test rig will include one station with hydraulic connection.	3.0
4	The test rig is capable of testing nine different predefined TBUs.	4.0
5	The test rig is rigid.	4.0
6	The test rig provides an option to simulate elasticity.	5.0
7	The test rig is designed for fatigue.	5.0
8	The test rig allows space for test sensors.	3.7
9	The test rig provides space for additional air cylinders used for test of emergency release of parking brakes.	3.7
10	The test rig is able to test future TBU models.	2.7
11	The test rig is constructed in a cost efficient way.	3.0
12	The test rig is movable.	4.0
13	The test rig fits within and can be moved about in the company's lab facility.	4.0
14	The test rig (without TBUs) is light enough to be lifted by the existing forklift truck.	4.0
15	The test rig allows access for pneumatic connections.	4.3
16	The test rig allows access for a movable crane when loading/unloading TBUs.	3.0
17	The test rig uses flexible fixtures that permit mounting of various TBUs.	3.7
18	The test rig has a set of different plates.	2.7
19	The test rig allows an easy and ergonomic mounting of TBUs.	3.3
20	The test rig provides storage for tools, bolts and such.	1.0
21	The test rig provides sufficient space between TBU and the rig for tool access.	3.0
22	The test rig allows an easy and quick changing of TBUs with various spindle lengths.	2.7
23	The test rig is constructed of a material with good characteristics regarding weight and strength.	2.7
24	The test rig allows access for a forklift truck when loading/unloading of TBUs.	2.7
25	The test rig uses components that are rigid and resilient to wear.	4.3
26	The test rig provides a safe work environment.	4.0
27	The test rig allows access to all stations.	4.0
28	The test rig allows for a quick change of TBUs when loading/unloading.	2.3

Commentary

The result shows that most of the needs are of high importance which hardly is surprising. Some of the needs are ones that the new test rig must be able to do, and as such they get a very high importance. A few, such as the ones regarding storage for tools, is ranked lower as there will undoubtedly be enough room for storing tools in the vicinity of the test rig.

As an example of how the importance differs, it is more significant that the test rig can be run in a safe way than the mounting of TBUs is done in an ergonomically good way. Especially since some of the test are carried out for months and the mounting only takes about an hour.

4.3 Product specifications

When the customer needs have been identified the work to establish product specifications starts. The goal is to achieve target specifications that will be considered throughout the project when developing the product. It is important to remember that the achieved target specifications might change in a later stage because of tradeoffs and unexpected events. The final specifications will be set in a later state of the project.

The needs attained in previous section are first combined into specifications after how they are related. Different needs can occur in more than one specification and as can be seen in Table 4.5 that occurs frequently. Each specification is given a unit and a final importance, which is obtained from the average importance calculated from the needs used in defining the specification. Some of the specifications are not measureable and are therefore graded subjectively. Table 4.6 gives a brief description of the specifications and its intended purpose.

Table 4.5 Specifications and their related needs.

Spec. No.	Need Nos.	Specification	Unit	Average importance	Final importance
1	5, 7, 23	Maximum braking force of TBUs	N	3.9	4
2	1, 5, 7, 10	Maximum weight of TBUs	kg	3.5	4
3	11	Cost	SEK	3.0	3
4	5, 6	Elasticity	mm/kN	4.5	5
5	1, 2, 5, 9, 12, 13, 19	Size	m^3	3.6	4
6	1, 5, 11, 14, 20, 23	Mass	kg	2.8	3
7	5, 7, 23, 25	Stiffness	mm/kN	4.0	4
8	2, 9, 15, 18, 21	Clearance for mounting TBUs	mm	3.5	4
9	12, 16, 24	Clearance beneath the rig	mm	3.2	3
10	5, 7, 23, 25	Service life	Years	4.0	4
11	6, 8, 9	Clearance for bulky TBUs	mm	4.0	4
12	1, 2, 4, 6, 10, 12, 13, 17, 22	Flexibility	Subjective	3.5	4
13	16, 17, 19, 21, 22, 28	Changing time of TBU	Hours	3.0	3
14	5, 19, 21, 26	Safety	Subjective	3.6	4
15	16, 17, 19, 20, 21, 22, 24, 26, 28	User friendly	Subjective	2.9	3
16	1, 2, 4, 10, 17	Number of various types of TBUs at each station	Amount	3.3	3

Table 4.6 Description of specifications.

Spec. No	Description of specification
1	Maximum force from each TBU the test rig should be able to handle both statically and in aspect of fatigue.
2	Maximum weight of each TBU.
3	The total cost of manufacturing and assembling the test rig and its additional components.
4	The interval of possible embedded elasticity.
5	The final size in all directions of the test rig and its components.
6	The total mass of the test rig.
7	The maximum allowed strain of the test rig when in use.
8	The clearance, for use of tools, between the test rig and the fixture holding the TBU.
9	The clearance, for access for the forklift/crane used in the lab, between the test rig and the floor.
10	Number of years the test rig should be useable.
11	Clearance needed so that TBUs with additional obstructions/or differently placed air connections can be mounted.
12	How many various TBUs with different lengths, sizes, obstructions and so on that can be mounted at each station.
13	How long time it takes to remove a TBU and mount another.
14	Is there any risk of the operator getting injured?
15	Is the test rig designed so that the operator can mount/dismount TBUs in an ergonomic way and with as little effort as possible?
16	Amount of TBUs that can be mounted at each station.

Commentary

As can be seen in Table 4.5 most of the specifications had about the same importance, which implies that the final product should have quite balanced tradeoffs between specifications. However there are some differences, and examples of less vital attributes are cost, mass and changing time whereas elasticity and stiffness are more important.

4.4 Benchmarking

Due to the definition of the product, test rigs for TBUs are not that common or documented around the world, it was decided that no external benchmarking was needed. Analyzing Faiveley's existing equipment is more than enough to get adequate information about similar products. Of course an external benchmarking will be made further on in the project when minor sub problems have to be solved.

The test lab at Faiveley has used a number of different rigs and equipment throughout the years for various tests of the TBUs. When going through these a total of three different test rigs where found which were considered useful for this project. The three existing types of rigs are presented below with a short description of its function. The total size has been measured and the weight has been estimated.

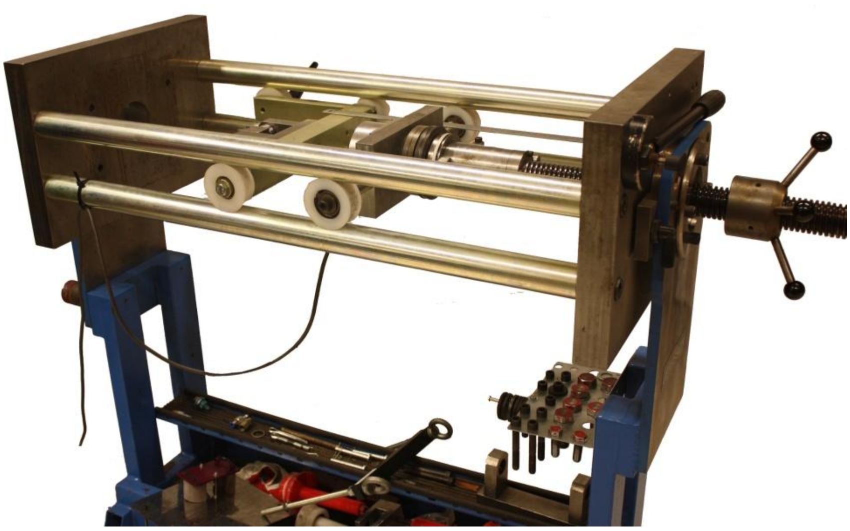
4.4.1 Model I

On the currently available rigs of this type (there are two identical), tests on only one TBU can be performed per rig. Given the great variation of TBU designs and various dimensions the amount of work for mounting a TBU on the rig is sometimes large. For example if extra long TBUs are to be tested the bearing round bars on the sides of the rig have to be replaced with longer ones. These are very heavy and working with such heavy masses is not convenient from an ergonomic perspective.

As can be seen in Picture 4.1 the overall design is quite simple. Each of the shortest sides consists of 50 mm thick plates. These have to be thick enough to carry the weight and load of the TBU when the brake is applied. The TBU is mounted to the rig by bolting it to two fixture plates (one on each side). These plates are then bolted to the rig by two M12 bolts on each plate.

The position of the load cell, which might also include the elastic stops, is altered by turning the threaded axel on which it is set. The axle functions the same way as a screw. The axle is fixed to the rig via another axel mounted perpendicular to it which has plastic wheels mounted on its ends which run between the bearing cylinders of the rig.

- **Size:** length 1000, width 510, height (from floor) 1200 mm.
- **Mass:** approximately 200 kg.
- **No. of TBUs:** One at a time.



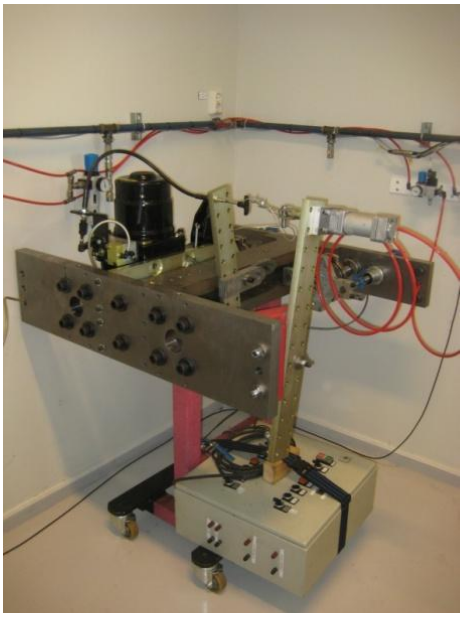
Picture 4.1 Model I.

4.4.2 Model II

This test rig is used for today's fatigue tests. It is more robust than Model I which can be seen in Picture 4.2. Comparing to the other models this rig has 35 to 40 mm thick plates throughout its design. Like the previous rig this rig also has a very flexible axle which controls the position of the load cell. But for this rig the axle is driven by an electric motor instead of being hand driven. The two middle plates are well dimensioned and have lots of drilled holes to simplify mounting of different TBUs and equipments that might be needed to perform the test.

The amount of work for mounting the TBUs is relatively high as they are mounted from one side only. Another down side compared to Model I is that this rig cannot handle long TBUs as its construction is fixed and there is no way to extend it lengthwise.

- **Size:** length 1030, width 975, height (from floor) 1050 mm.
- **Mass:** approximately 230 kg.
- **No. of TBUs:** Two at most.



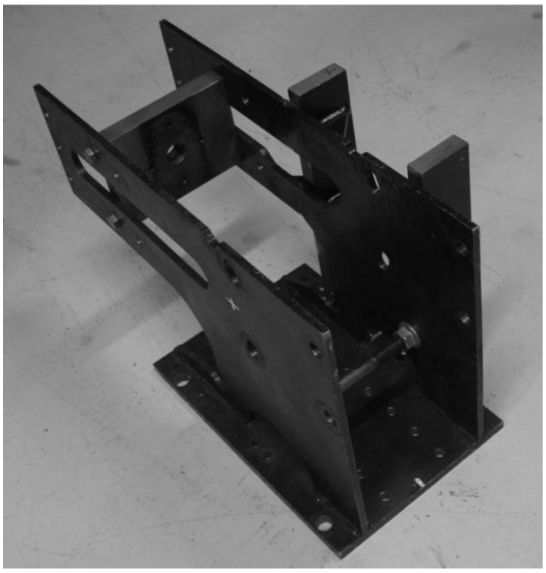
Picture 4.2 Model II.

4.4.3 Model III

This model is the oldest of the three and consists of two 15 mm plates welded together by a bottom plate. The arrangement of the test sensor also holds the two plates together and makes it more robust, see Picture 4.3. The perks of this rig is its low weight thanks to the TBUs being mounted directly to the rig and not on specially made fixture plates. It is very compact and only 230 mm wide. The down side of this is that not all TBUs can be mounted as the bearing plates cannot have an unlimited number of fixture holes.

This rig is primarily used in vibration tests where the loads are not particularly high on the rig itself. Should this model be used in a new rig the plates must be made thicker than the existing 15 mm ones.

- **Size:** length 630, width 230, height 500 mm.
- **Mass:** approximately 50 kg.
- **No. of TBUs:** One at a time.



Picture 4.3 Model III.

4.5 Target specifications

Once all specifications are obtained from the needs, the target values are to be chosen. One marginal and one ideal value are determined for each specification. Some marginal values were taken straight from the R&D Manager's objectives and restrictions (Appendix A) whereas other marginal values were obtained from measuring the already existing test rigs. For the specifications where the unit is subjective the values were chosen arbitrarily as can be seen in Table 4.7.

Most of the ideal values were obtained through the marginal values and the old test rigs as references while the values for specifications such as size and service life were estimated.

Table 4.7 Target specifications.

Spec. No.	Specification	Unit	Final import-ance	Marginal value	Ideal value
1	Maximum braking force of TBUs	N	4	70 000	80 000
2	Maximum weight of TBUs	kg	4	65	75
3	Cost	SEK	3	100 000	50 000
4	Elasticity	mm/kN	5	0 – 0.30	0 – 0.30
5	Size (W x D x H)	m^3	4	2,5 x 2 x 1,3	1,5 x 1,3 x 1,2
6	Mass	kg	3	1000	250
7	Stiffness	mm/kN	4	0.020	0.015
8	Clearance for mounting TBUs	mm	4	80	100
9	Clearance beneath the rig	mm	3	160	170
10	Service life	Years	4	10	25
11	Clearance for bulky TBUs	Subjective	4	Good	Very good
12	Flexibility	Subjective	4	Good	Very good
13	Changing time of TBU	Hours	3	2	1
14	Safety	Subjective	4	Medium	High
15	User friendly	Subjective	3	Low	High
16	Number of various types of TBUs at each station	Amount	3	2	5

Commentary

The total cost of the project is hard to estimate as no exact numbers have been mentioned from Faiveley's side. The marginal value is estimated and not set as the cost limit, hence it can be exceeded.

The maximum allowed weight of the rig cannot surpass 1000 kg as that is the lifting capacity of the forklifts at Faiveley.

It should be noted that the values for specification no. 16 are average values. Some stations might only be adaptable to one specific TBU whereas other stations might be more flexible and hold a greater number of different TBUs.

5 Pre design

The next step in the chain of product development is to start sketching the actual product. By various means a number of simple drafts or concepts are created. The different product concepts are then evaluated based on how well they fulfill the previously compiled specifications. A few of the better suited concepts will proceed to be further developed and finally a second evaluation with weighted criteria, called *Concept Scoring* [1 p.130 – 134], is performed from which a final concept is chosen.

5.1 Concept Generation

A test rig of the complexity of which is to be designed in this project must be thoroughly analyzed beforehand. How many different parts does it consist of? Which specifications must be considered primarily? Can it be divided into smaller, less complex, sub problems to make the designing and evaluations easier?

The most basic and fundamental part of the new rig is its base which must be able to provide space for five TBUs simultaneously while being compact and rigid. Hence it was decided that the overall design of the test rig's base was to be generated before additional parts such as TBU fixtures, sensor fixtures and pneumatic cabling could be designed. This had to be done in a way which optimized both size and weight and thereby also cost of the rig. The goal was also to make it as flexible as possible to allow for a great number of different TBUs to fit onto the rig.

Before any concepts for the general layout were generated the existing test rigs, as described in section 4.4, were analyzed to see if a combination of the three, called Models I – III, could be useful. The actual concepts were thought up using the "brainstorming" technique.

It was the will of Faiveley to build a test rig with exactly five stations of which two were assigned to the important and common TBUs, two for TBUs with bulky designs and one long station for the long TBUs. This resulted in an agreement that the concepts for the new rig all had to involve these five different stations. Hence the seven drafts generated look fairly alike and mostly differs on the aspect of station arrangement. To clarify the drafts a general explanation is given below.

The rig is the complete bearing structure which the TBU tests are to be conducted on. A station is a part of the rig to which the TBUs are mounted, it will have a different design depending on which type of TBU it will hold, see Figure 5.1. The TBU is the brake unit which is to be tested, it has many various designs. The concepts also show where the test sensors (load cells) are to be placed.

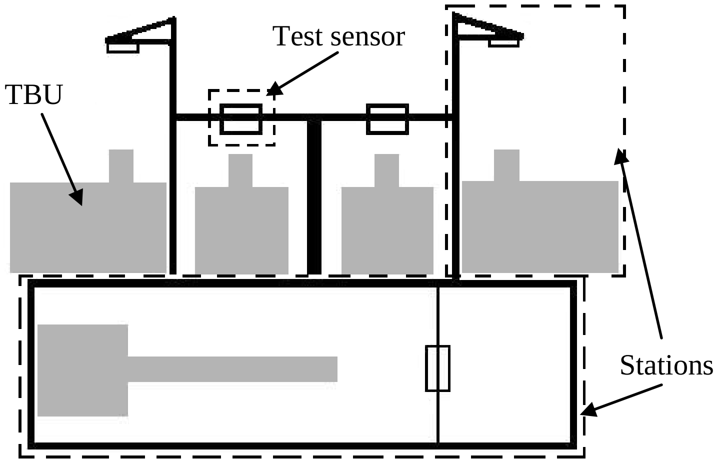


Figure 5.1 Description of test rig.

On the next few pages the seven drafts of the test rig's layout are presented with a sketch and a short description of its function.

Concept A

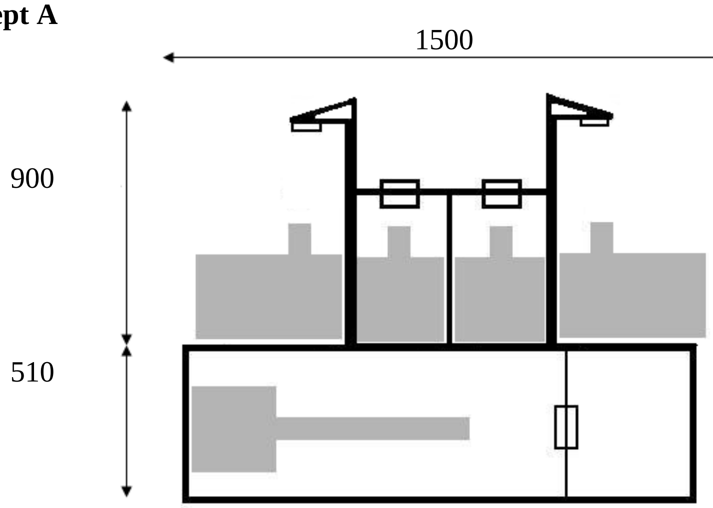


Figure 5.2 Layout A.

The layout of this concept is made out of one long station (Model I) at the bottom, two small stations (Model III) mounted to the long side of the long station and two stations on each side. These two stations will primarily handle TBUs with only a right or left mounting. Therefore the plates that hold these have to be bigger than in the rest of the structure.

Concept B

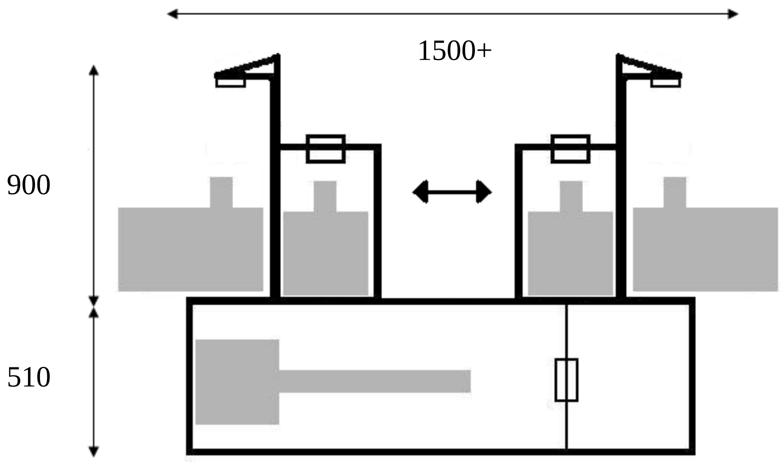


Figure 5.3 Layout B.

This concept is comprised of the same different types of stations as concept A. The only difference is that upper stations can be slid apart to make room for the technicians when they mount/dismount TBUs.

Concept C

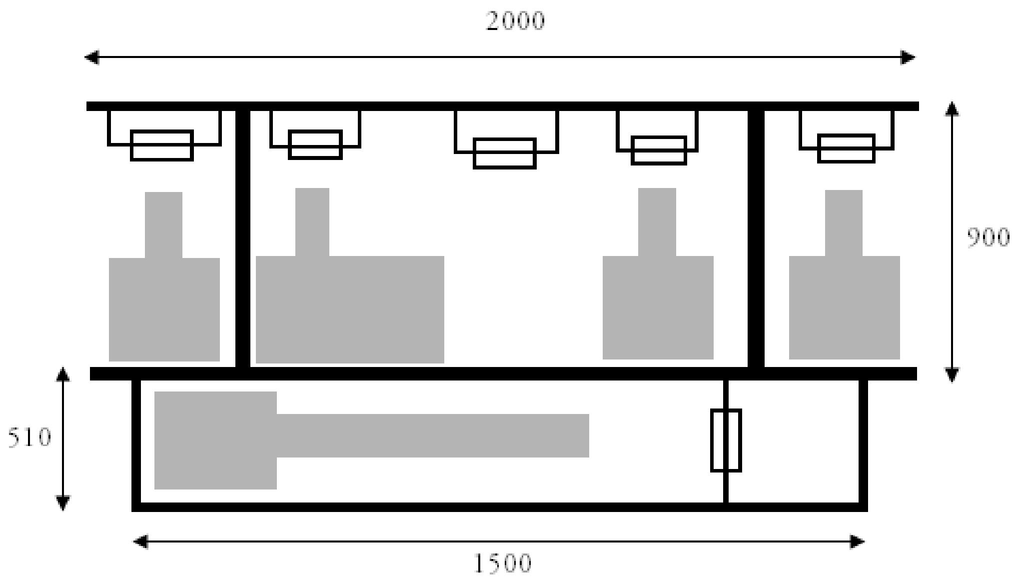


Figure 5.4 Layout C.

The general idea with concept C is to make the test rig as flexible as possible and make it able to hold TBUs with many different designs. A long station is needed and it is of the same size as before. The rest of the rig is not divided into stations as such but between the two bearing beams run a number of fixtures for the test sensors. This allows the TBUs to be mounted with a greater variation.

Concept D

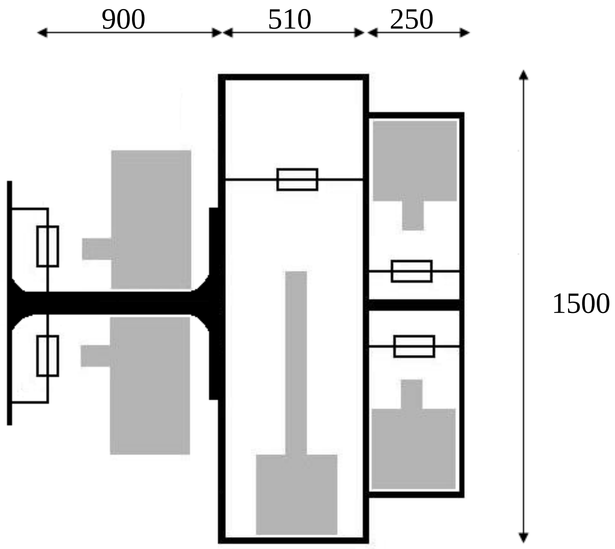


Figure 5.5 Layout D.

This layout holds a long station of Model I in the middle and on its right side two stations of Model III are mounted. To the left a two-sided station of type Model II is mounted which is able to handle TBUs with bulky design.

Concept E

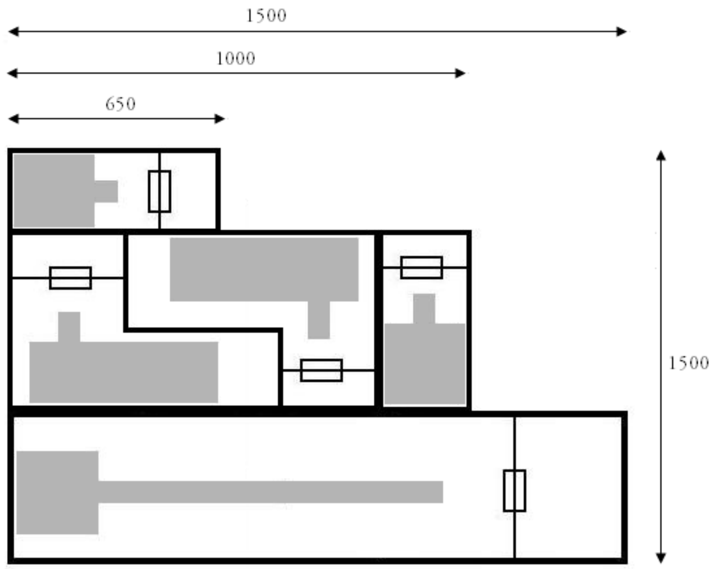


Figure 5.6 Layout E.

The idea of this layout is to use up space as efficiently as possible. Two stations in the middle are designed in such a way that they permit mounting of one-sided mounted TBUs with a very compact structure. Furthermore it consists of one station of type Model I and two of type Model III.

Concept F

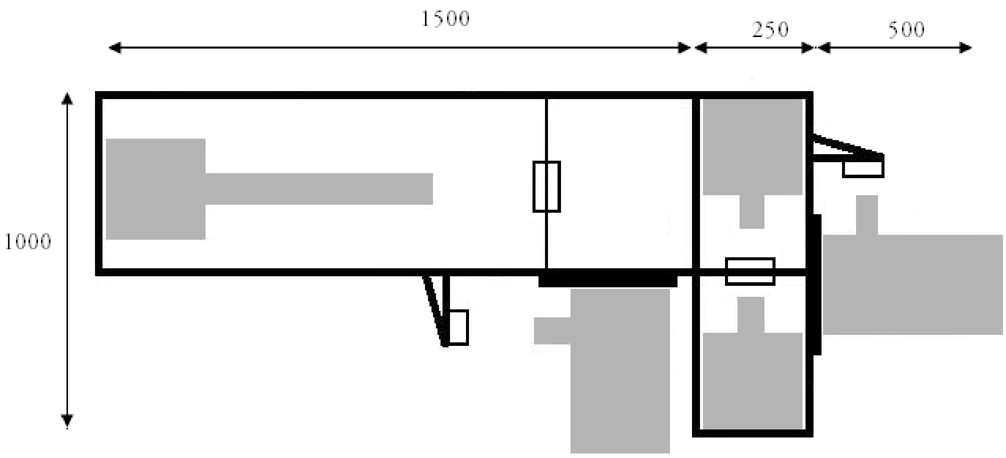


Figure 5.7 Layout F.

This layout for the test rig has the long station and the two small ones (Model III). The rig’s structure is enhanced on two locations where two additional TBUs are to be mounted. These can be of various designs but are primarily thought to be TBUs with bulky designs.

Concept G

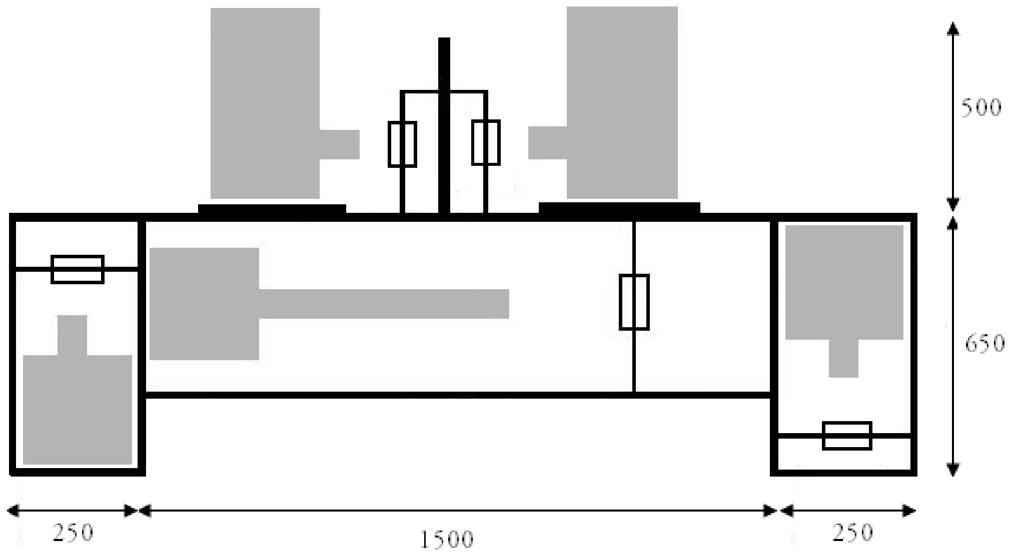


Figure 5.8 Layout G.

Here the two small stations of type Model III are located on the short ends of the long station. This is to make them as easy as possible to access while mounting TBUs. The structure is enhanced on one of the long side of the long station in order to fit two more TBUs.

5.2 Primary evaluation

When a satisfying number of concepts of the basic construction were generated the process of evaluating and ranking these started. The method of doing this is to choose important criteria and then compare and rank the different concepts.

5.2.1 Concept screening

The criteria used in this process were generated from a combination of the product specifications and a discussion with the managers at the company. Whereas criteria such as *Cost*, *Size* and *Weight* are quite straight forward the others might need a small explanation.

With *Stability* the test rig’s response in use is considered. How much will different components strain under high loads and how will the whole rig itself correspond to the forces? *Accessibility* means how easily the technicians can reach each station when mounting the TBUs and connecting the corresponding pneumatic devices. *Clearance for tools* could have been included in the prior criteria but sometimes a station can be easily reached but still cause problems when mounting because of lack of space for the tools needed. With *Flexibility* the number of different TBUs possible to mount on the rig is regarded. The last criteria considers how easily future developed TBUs can be adapted to the rig.

Concept A was chosen as reference because of its straight forward design and that the other concepts more or less were adjusted from this one. The result of the first evaluation can be seen below in Table 5.1.

Table 5.1 Primary evaluation of the design suggestions.

Criteria	A	B	C	D	E	F	G
Cost	0	-	-	0	-	0	0
Size	0	0	-	0	0	0	0
Mass	0	-	-	0	-	0	+
Stability	0	-	+	0	+	-	0
Accessibility	0	+	0	+	0	+	+
Clearance for tools	0	0	+	0	0	0	0
Flexibility	0	0	+	0	0	0	0
Simplifying for future designs	0	0	+	0	0	-	0
Sum "-"	0	3	3	0	2	2	0
Sum "+"	0	1	4	1	1	1	2
Sum "0"	8	4	1	7	5	5	6
Total	0	-2	+1	+1	-1	-1	+2
Proceeds	X		X	X			X

Commentary

B: Compared to concept A concept B will be more expensive and potentially heavier due to its sliding ability. The size is the same but gets a little larger when the upper modules are slid fully to the sides. The sliding ability also makes it harder to use welds and mechanical joints which prevent a robust structure. On the plus side the sliding permits a greater accessibility to the stations in the middle whereas the clearance for tools is the same as for concept A. Concept B allows for exactly the same configuration of different TBUs as A, hence it get a “0” on *Flexibility*. On the count of *Simplifying for future designs* it is deemed the same as A because of the similarities in station layout.

C: The size of C is a bit larger than A and therefore the weight is also higher while the stability is improved. More material and more fixtures for test sensors equal a higher cost. It has the same accessibility to the various test stations as concept A. The clearance for the tools needed for mounting TBUs is greater because C’s structure is not divided into stations to the same extent as A’s. Because of the large space in the middle it will provide a high flexibility and adaptability for many different TBUs, even those not included in this project.

D: This concept uses the same configuration of stations as A and because of that the size, weight, cost and stability is estimated to be the same. It has a better accessibility to the stations because the two small stations are located in front of each other and not side by side as in concept A. The clearance for tools is neither greater nor lesser and because it consists of the same stations as A it has the same flexibility. On the count of future TBU designs this concept and layout gives the same variation as concept A.

E: The idea of E was to optimize the size used by the test rig. However the length of the long station (1500 mm) resulted in a total size that did not differ that much from concept A. In fact the more complicated configuration of the stations caused the weight and cost to get higher than that of A. The more compact layout of concept E is thought to give a better stability while the accessibility to the test rig when mounting TBUs and using the necessary tools is about the same as A. Lastly the flexibility and its adaptability for future TBUs is the same because the same stations as concept A are used.

F: The layout of concept F is more stretched out than A and as such it is considered to be less stable. The same amount of material will be used which gives concept F the same rating in criteria concerning weight, cost and size. The arrangement of the stations provides a greater accessibility for the users when mounting the TBUs while the clearance for tools is the same. The arrangement also makes the flexibility the same as for concept A. The adapting of future TBUs to this concept is considered hard as the down-right area of the rig can get very cramped with three TBUs mounted in a small area of the rig.

G: This concept consists of fewer plates than concept A and will therefore weigh less while its size and cost stays the same. The symmetric arrangement of the five stations makes it just as stable as A and also makes it more accessible when mounting brakes. However the size of each station is the same as that of the stations of concept A and thus giving G the same clearance for tools as A. This concept is thought to manage the same amount of various TBUs as concept A because of the similarity of the stations.

5.2.2 Presentation of concepts to Faiveley

On September 17^{th} 2009 a discussion[2] was held about the winning concepts. Concepts D and G were accepted as they were for further development. However as concepts A and C were very much alike it was settled that a combination of the two also was suitable solution to proceed with.

It was also agreed upon that from now on the test rig only had to be capable to test long TBUs with a spindle length no longer than 500 mm. Hence the long stations do not need to be 1500 mm but can be about 1000 mm instead. This was done as very long TBUs can be tested with a shorter spindle without it affecting the braking function too much.

5.3 Further development

From the first evaluation concepts D, G and a combination of A and C were selected for further development. That is, some new dimensions were set and others were changed. Furthermore it was discussed how to actually mount the TBUs and finally simple 3D models were made for better understanding. From the 3D models estimated weights of the rigs were obtained (excluding the TBUs) where a general density of 7.8 g/cm^3 was used. It should be stated there was no intention to make complete drawings, they were only made to better visualize the size of the concepts.

Concept AC

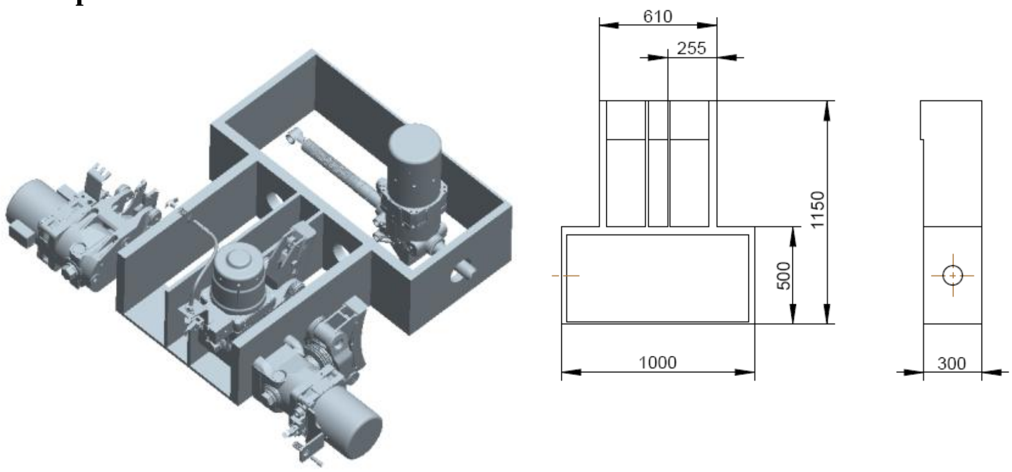


Figure 5.9 Concept AC: 3D-view (left) and dimensions (right).

This concept is a further development of concepts A and C. The most significant changes are that the two stations in the middle are smaller and supposed to resemble Model III and that the four middle TBUs' mounting positions have switched direction

[2] Presentation of the winning concepts. Participating were J. Stridh, M. Lindström, F. Blennow and A. Arnell.

by 180 degrees. In this way material can be used in a more efficient way which results in reduced weight and cost.

As can be seen in Figure 5.9 the dimensions have decreased quite dramatically. For the long station it was possible to decrease the length from 1500 mm to 1000 mm due to the restrictions set by the company were changed. The weight was calculated to 355 kg.

Concept D

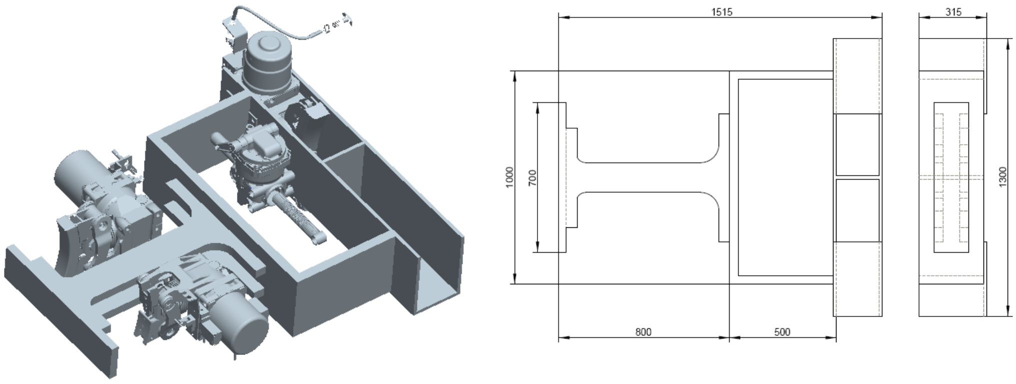


Figure 5.10 Concept D: 3D-view (left) and dimensions (right).

The basic features are the same as before for this concept. In Figure 5.10 it can be seen that the two stations on the left are similar to Model II. As in concept AC the long station's dimensions have decreased but because of the two stations of type Model III to the right, the overall dimensions are practically the same. The weight of the new concept D is calculated to 418 kg.

Concept G

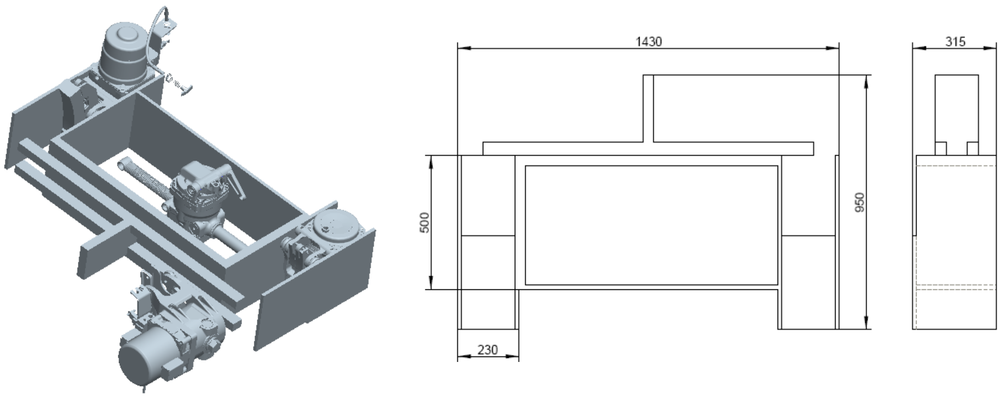


Figure 5.11 Concept G: 3D-view (left) and dimensions (right).

This concept has not changed that much either except that two bars, see Figure 5.11, have been attached to the long station for fixation of the two TBUs to the left. The test rig's width has decreased for the same reason mentioned earlier. It can be noted that this concept has the lowest weight of the three. The weight is estimated to 312 kg.

5.3.1 Concept scoring

To distinguish between the remaining concepts concept scoring was used. To give the more important criteria more influence of the outcome a weighted scoring was applied. As can be seen in Table 5.2 the reference (bold number) is not the same throughout the scoring but changes for each criterion. This is because the rating system in this way becomes more justice and easier to apply.

The weight factors are based on the specifications' weighting but *Cost* has increased in importance.

Table 5.2 Results of concept scoring.

Criteria		**Concept**					
		AC		**D**		**G**	
	Weight	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score
Cost	0.25	**3**	0.75	1	0.25	3	0.75
Size	0.15	4	0.60	2	0.30	**3**	0.45
Weight	0.05	**3**	0.15	2	0.10	4	0.20
Stability	0.15	2	0.30	**3**	0.45	1	0.15
Accessibility	0.15	4	0.60	5	0.75	**3**	0.45
Clearance for tools	0.05	2	0.10	**3**	0.15	3	0.15
Flexibility	0.15	**3**	0.45	3	0.45	3	0.45
Simplifying for future designs	0.05	3	0.15	3	0.15	**3**	0.15
	Total score		3.10		2.60		2.75
	Rank		1		3		2

Commentary

Cost: AC and G get the same score because they consist of approximately the same number and size of plates and beams. None of them have any geometry that is difficult to manufacture. D however has two plates which are rounded at the ends. These require a few more processing steps when manufactured and the cost will therefore get higher.

Size: G takes up 1500 x 650 mm^2 and D 1300 x 1500 mm^2 and AC 1000 x 1150 mm^2 of space. Hence G is of a smaller size but has an oblong layout. AC is much more square which will make it fit easier into the lab.

Weight: The calculations of the weights were performed on the 3D models in Pro Engineer. The density was set to 7 800 kg/m^3 to correspond to any given construction steel. D weighs more than AC and G less than AC.

Stability: D as reference is assumed to have a good stability because of its weight and symmetry. The increased weight towards the other designs depends primarily on the thicker plates used. G gets a low rank because it lacks a steady base frame. AC get a lower rank because it lacks a rear beam or plate that would have given it more of a ”box shape” and thus stability.

Accessibility: AC allows for good accessibility to all the stations except the two small ones in the middle, especially in the area where the load cell will be placed. Layout G is deemed less accessible because of its small stations being in the way of the TBUs with parking brakes to the sides. Also the areas on the short sides of the long station will get cramped. D however allows for good access to all stations throughout the rig.

Clearance for tools: All three suggestions have areas where it will be hard to get good access with the tools required for mounting the TBUs. For AC it is primarily the plates on the outsides where two TBUs are to be mounted to the same plate. D and G have better opportunities for tools to access the rig but they must be reviewed in order for the mounting to go smoothly.

Flexibility: All three suggestions possess the same possibility to test brakes of different types because they have the same number and similar design of the test stations. No consideration regarding the test rigs’ connection abilities to pneumatic systems have been taken. This is because these applications easily can be adjusted afterwards.

Simplifying for future designs: Again they all get the same rank as they hold the same opportunities to drill new holes for mounting future types of TBUs.

When all criteria were summarized concept AC had the highest score. The other two suggestions received scores that were quite high but still significantly lesser than that of AC.

6 Detailed design

Concept AC is the chosen concept that will be even further developed in the next phase called detailed design. This phase consists of an iterative process in which all the different parts of the test rig will be put together and obtain its final design. It is only natural that this process is time consuming as the inbound parts are not easily joined together which require the base frame of the test rig to change many times before a last solution is presented. Furthermore as time goes by in the development process wishes and requirements on the test rig set up by the company might change, leading to even more changes on the original design.

In this stage the parts that were previously left out, as the overall layout was considered the primary and most vital part, will now be focused on. These parts will solve issues such as:

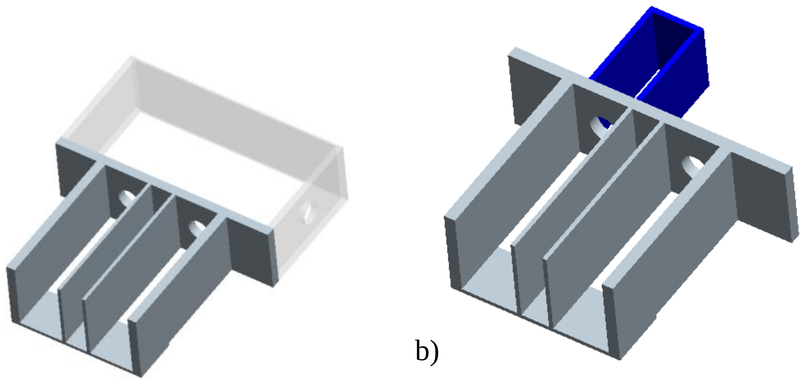
- How the TBUs are mounted to the test rig.
- How the test sensor and its mounting will function.
- How the requirement of an elasticity of 0 – 0.3 mm/kN will be met.
- How the parts of the rig are joined together.

6.1 Major changes

Initially the chosen concept was presented to Faiveley and its design was discussed with F. Blennow[3]. One of the main conclusions was that the large box of type Model I, used to house the long TBUs, uses up too much space and material. It stands for about 50 % of the total material and thus a big part of the cost. According to him it is not justifiable anymore to have a station assigned only for long TBUs. However Faiveley still wanted to be able to test long TBUs in the new test rig. The conclusion led to the removal of the long station and the three plates that made up the box. The fourth plate remained as it is part of the other stations.

To adjust the test rig to still being able to handle long TBUs it was decided that in the two stations in the middle a hole should be made so that the long spindles can be mounted through the front plate. In order to use the load cell and the elasticity an additional construction must be mounted directly to the rig, see Figure 6.1b**.**

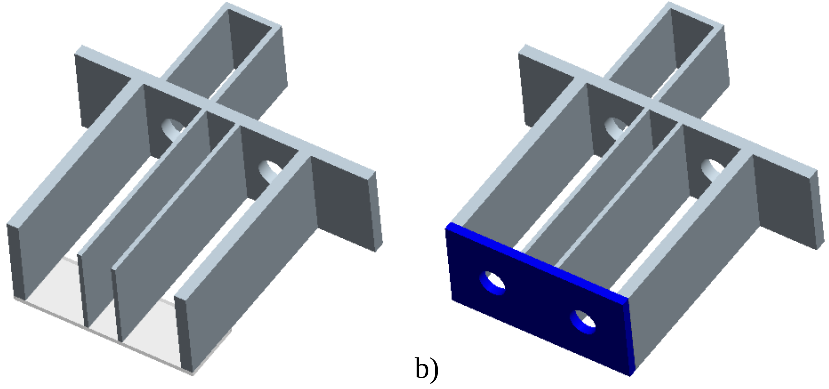
[3] Meeting on October 5th 2009. Participating were M. Lindström, J. Stridh and F. Blennow.



a) b)

Figure 6.1 a) Removal of long section. b) Addition of plates.

Some of the TBUs have obstructions such as hand brakes that extend downward. Concept AC originally had a plate across all the stations beneath the rear of the rig where the TBUs are to be mounted. Its function was to increase the stiffness of the whole rig. Because of the obstructions this plate was moved and placed behind the TBUs giving the rig a complete box shape, see Figure 6.2. In addition this new plate was made thicker. Two equally large holes as those in the front plate were also made in the rear plate in order to give the lab technicians access to the adjustment screw situated on the back of the TBUs.



a) b)

Figure 6.2 a) Removal of plate. b) Addition of plate.

With the present design of the concept it is hard to mount TBUs on the sides if TBUs are in place in the middle. There is not enough space for the bolts. To solve this, the stations in the middle were made wider and the TBUs are no longer mounted to the side plates. Instead they use the same fixture plates as the long stations, used today in the old test rig, Model I. This change is presented Figure 6.3 below.

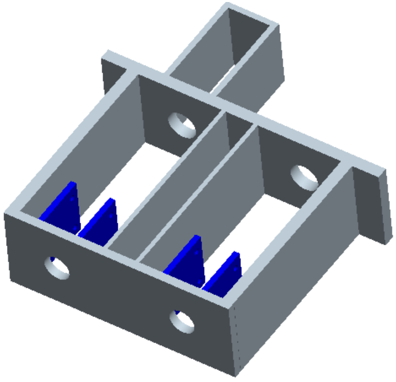
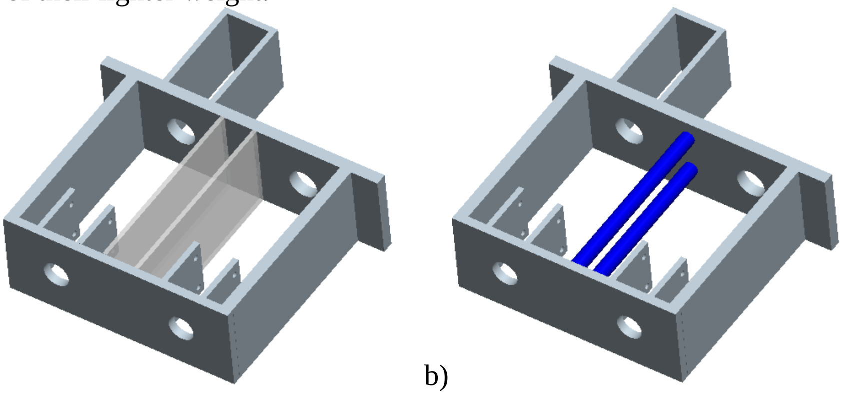


Figure 6.3 Fixture plates.

The two plates in the middle felt abundant when the TBUs are no longer to be mounted to them. They were both heavy and expensive and were consequently removed. In order to maintain a sufficient stiffness of the whole rig two round bars were added in their place, see Figure 6.4. These are considered cheaper than the plates because of their lighter weight.



a) b)

Figure 6.4 a) Removal of plates. b) Addition of round bars.

During the same meeting it was discussed whether or not the deformation on the sides of the front plate would be greater than what was set as a limit. From the restrictions of the project, Appendix A, the maximum allowed deformation when the brakes are applied is 0.02 [mm/kN]. Totally, when full force is applied, the deformation can get up to

$$70 \cdot 0.02 = 1.4\ [mm] \quad (6.1)$$

As the ends of the front plate extend freely outside the box the bending stiffness is not optimized. In an attempt to increase the bending stiffness and lead the force more efficiently into the side plates of the rig, two stiffening plates were added on each side, see Figure 6.5 on the next page.

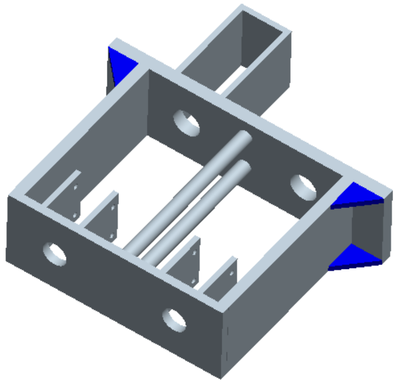


Figure 6.5 Stiffening plates.

Commentary

One change in particular had a profound influence on the design of the rig. When the long station was removed it resulted in a completely new design of the rig's layout. Perhaps if the decision to lose the long station had come earlier the layout might be different as all concepts were based on a long station. It was reflected upon whether or not new concepts were to be thought up but the design so far felt satisfying to most of the specifications in a very good way. It is small and relatively simple while still being very flexible.

6.2 Minor changes

These previous changes were the main changes made to the design of the rig in the beginning of the detailed design phase. In addition to these a number of less significant changes were made. The hole pattern in the plates on the sides, where the TBUs are to be mounted, were made to be able to hold both the TBUs with parking brakes to the side and the two most commonly used TBUs. All in all three different types of TBUs can be mounted to each side of the rig. The two stations in the middle can house any type of TBU except the two that have parking brakes to the sides giving these two stations a great flexibility. All that has to be done to change TBU type in the middle stations is to replace the fixture plates.

The length of the side plates was long debated. A summary of the five different TBU's normal lengths (not including the four long TBUs) revealed that the maximum length was 365 mm. The summary is presented in Appendix B. With a maximum length of 365 mm and a length of a load cell of 100 mm and the total length for the elasticity and adjustability of the load cell mounting needing to be about 200 mm, the side plates must be at least 665 mm (365 + 100 + 200). Given that the TBUs cannot be mounted to the extreme end of the plate due to the bolts holding the plates together the length of the side plates were set to 700 mm, see Figure 6.6.

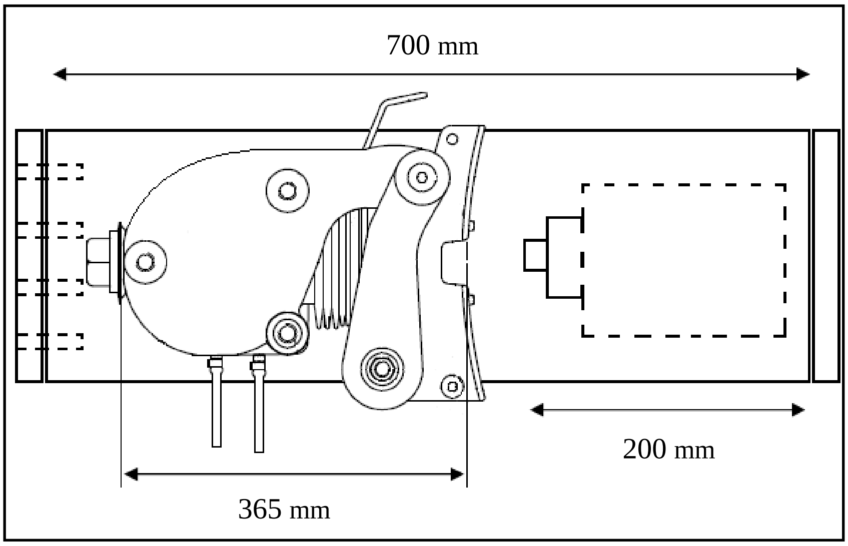


Figure 6.6 Schematic figure of usage of space.

Lastly an additional 22 threaded M12 holes were added to different parts of the rig as seen in Figure 6.7. This was required to meet the demand that the rig had to be able to have a number of extra air cylinders mountable to it when performing endurance tests on parking brakes. Six holes were placed on each side plate and the remaining sixteen on the rear plate.

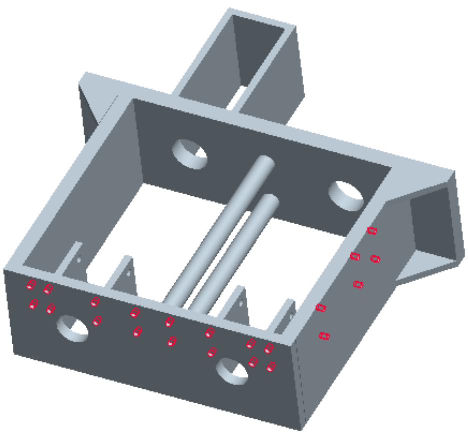


Figure 6.7 Added threaded holes.

6.3 Sub problems

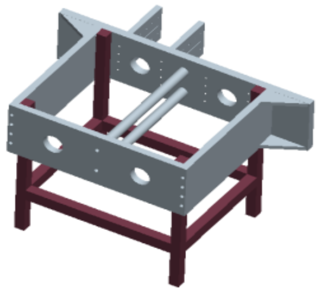
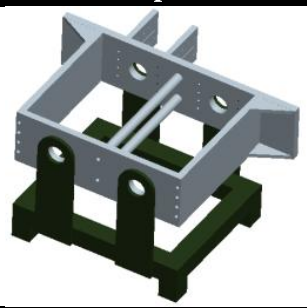
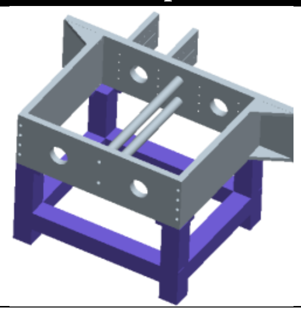
Now that the rig's base itself had a complete design the remaining parts of the test rig will be designed. These parts were of different complex nature and to efficiently solve all remaining issues they were made into sub problems solving one at a time. Solutions to the problems were either thought up using the brainstorming technique, found when consulting Faiveley's own expertise or the result from external benchmarkings. The remaining issues to be solved are presented below.

6.3.1 Foundation

From the target specifications it can be concluded that the design of the foundation does not have a crucial impact on the final result. The design of the upper part of the test rig should guarantee good stability and its design is of high importance for the other specifications as well. Never the less some specifications depend on the foundation such as weight, size and cost. The most important issue is that today's equipment such as a fork lift can be used when mounting TBUs and moving the test rig around. In the beginning of the project it was discussed whether the test rig should have wheels or not to make it more mobile but as work progressed this function felt abundant. Especially since the weight of the rig is high and it is supposed to sit in the same part of the lab for a long time. It was decided, when consulting Faiveley, that there was no need for wheels.

A short brainstorming process resulted in three different solutions which are described in Table 6.1. Because of the simplicity of this sub problem it was agreed upon that it was enough to have a short discussion with F. Blennow[4] to decide which concept to choose. Concept C1 was chosen because its design does not interfere with the main structure but also because it was expected to be the least costly.

Table 6.1 Concepts for different foundations.

Concept A1	Concept B1	Concept C1
		
This design consists of solid beams. The beams are welded together and attached to the upper structure by screws.	This foundation is attached to the test rig with "ears" like Model I described in section 4.4.1. Further on the "ears" are welded to hollow beams.	This design consists of hollow beams that are welded together. It is attached to the upper structure by screws in the bottom side.

[4] Meeting on October 6^{th}. Participating were M. Lindström, J. Stridh and F. Blennow.

6.3.2 Load cell fixture

This sub problem was the most thorough and had itself a number of sub problems:

- How should the load cell fixture be mounted?
- How should the elasticity specification of 0 – 0.3 mm/kN be fulfilled?
- How should the distance between the TBU and the load cell be adjusted?

It was specified from Faiveley that a load cell of type HBM C2 [2], Figure 6.8, will be used when measuring the brake forces. It was also in the company's interest to manufacture four dummies with which to replace the load cells during the endurance tests.

Figure 6.8 Load cell of type HBM C2.

In today's test rigs Faiveley uses disc springs to achieve the wanted elasticity. Due to the disc springs' flexibility, they are very easy to rearrange to achieve another spring constant as can be seen in Figure 6.9, and because of the economic advantage to use components the company already have in the lab it was decided to keep this solution.

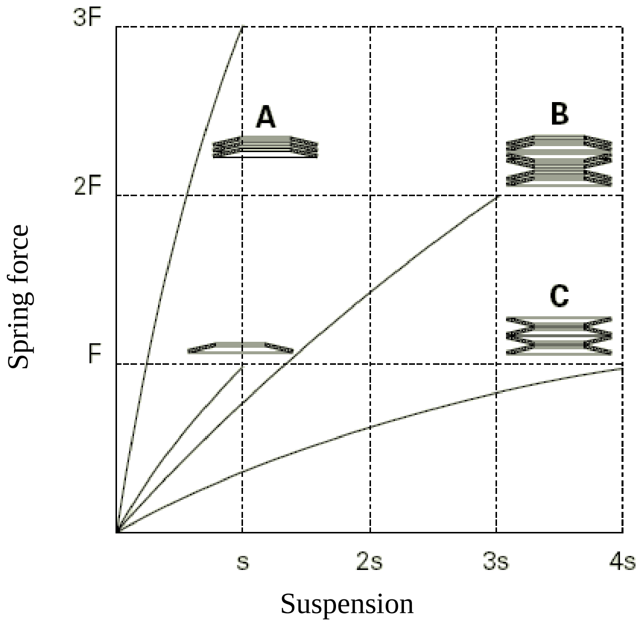


Figure 6.9 Spring constant for different configurations[5].

To calculate the maximum number of disk springs needed for the elasticity of 0.3 mm/kN data from the disc spring producer Lesjöfors' webpage [3] was used. The dimensions of the disks used are presented in Figure 6.10 and are $D_i = 61$ mm, $D_e = 125$ mm, L_0 =9.6 mm and $t = 6$ mm respectively.

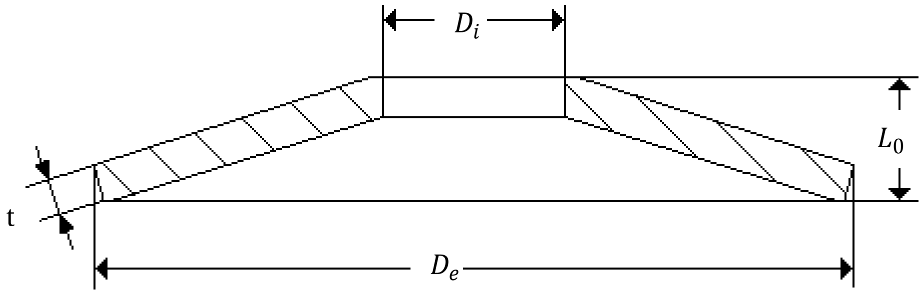


Figure 6.10 Dimensions of a disc spring.

From the data sheet [3 p. 126] the elasticity for each disk spring then could be calculated as

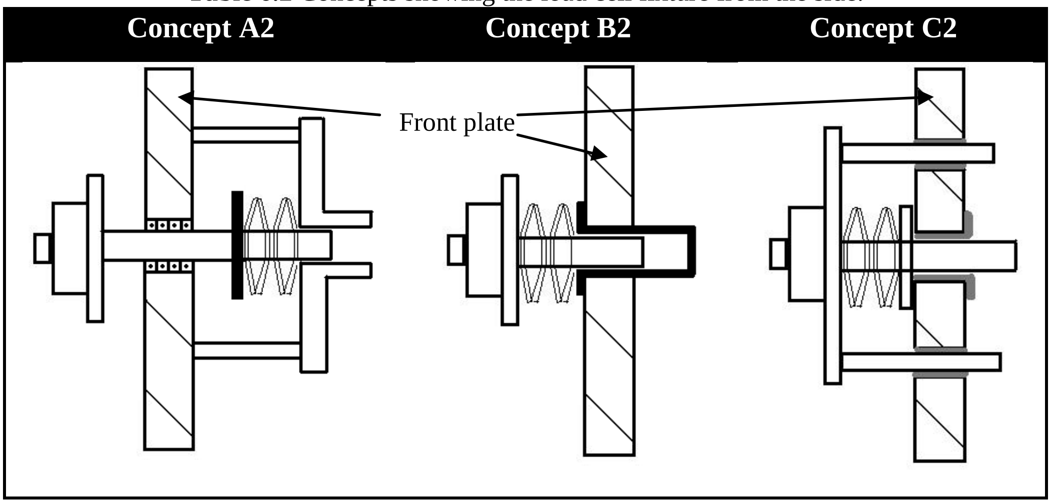
$$\frac{spring\ suspension}{maximum\ force} \Rightarrow \frac{4.00\ [mm]}{66.70\ [kN]} \approx 0.06\ mm/kN \quad (6.2)$$

[5] Figure borrowed from [3 p. 121].

The required elasticity of 0.3 mm/kN is then achieved if five disk springs are placed parallel like alternative C in Figure 6.9, thus for each station there has to be enough space for five discs which equals a length of $5 \cdot 9.6 = 48$ mm.

Now the remaining problem to solve was how to mount the load cell fixture to the test rig and to adjust it back and forth easily. After a brainstorming three concepts were generated which are shown in Table 6.2. The three concepts were quite similar when it came to the basic function. The load cell is mounted on a plate which is attached to a 60 mm axle that slides in the braking direction. The axle can easily be removed so additional disk springs and inserts can be added or removed.

Table 6.2 Concepts showing the load cell fixture from the side.



Concept A2
A2 differs from the other two in the way that the additional components are placed to the right of the front plate making the distance between the rear plate and the load cell as long as possible. In the front plate there is a ball bearing which will keep the axle aligned and help it slide more smoothly. The force will be applied on the structure to the right of the front plate and the load cell plate will never touch the front plate.

Concept B2
B2 has a drilled pipe which is force fit in the front plate. The pipe has very narrow tolerances and a thin layer of lubricant so the axle will slide correctly and with low friction. The force is first absorbed by the pipe which directly leads it to the plate.

Concept C2
In the last concept, C2, the axle just slides through a plastic bushing which also is force fit in to the front plate. Furthermore C2 also has two extra rods helping the load cell fixture to be perpendicular and positioned as wanted, these rods will slide through two small plastic bushings.

A2 was rejected because the rig has sufficient space to the left, thus there is no interest to choose this more expensive and complicated solution. After a discussion with F. Blennow comparing the two last alternatives concept C2 was finally chosen. The two main reasons why concept C2 was considered as a better solution were because the two extra rods were useful for positioning and that B2 with its pipe with narrow tolerances was more expensive.

After choosing C2 it had to be developed further. The axle was decided to be hardened to be resistant against wear from the very hard disc springs. Likewise were two hardened washers added to be placed on both sides of the disc springs. The bushing in the middle will be force fit but to prevent any movement it will also have a flange to the right and a small screw that locks its position. The small bushings will on the other hand move freely with the rods as the load cell fixture moves back and forth. The most extreme distance the fixture will move while in use is about 21 mm which is easily calculated from the elasticity specification of 0.3 mm/kN times a maximum force of 70 kN.

To adjust the position of the load cell inserts of different sizes are used. To simplify these adjustments of the fixture the rods are threaded and to secure it at the wanted position six nuts are used. The final design is shown below in Figure 6.11 and Figure 6.12.

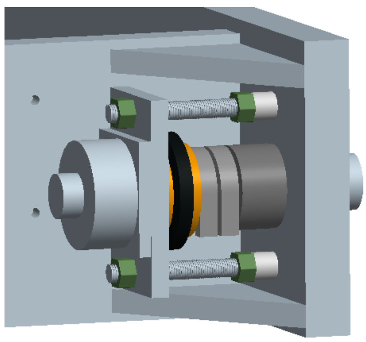


Figure 6.11 Final design of the load cell fixture.

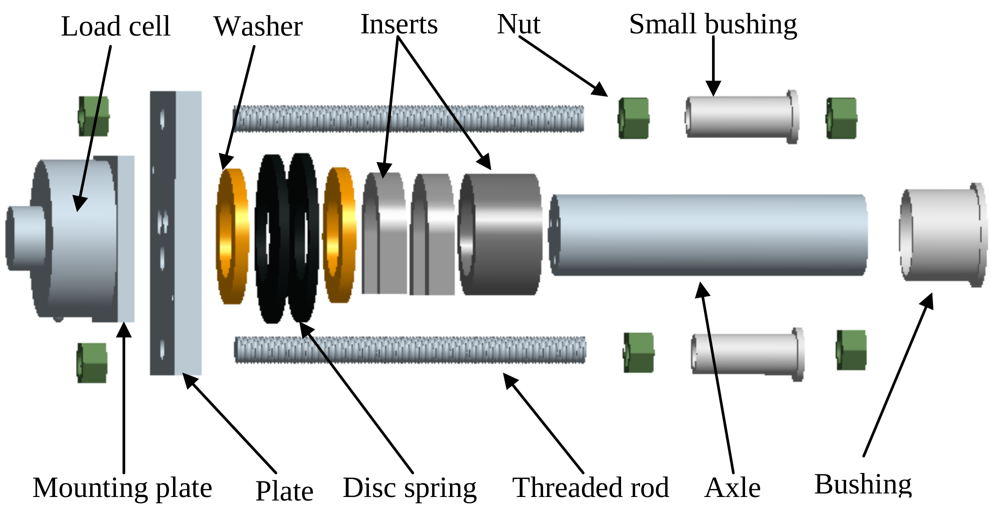


Figure 6.12 Exploded view of load cell fixture.

6.3.3 Rotating spindle

The next sub problem to solve was the rotating spindle for the long TBUs. This is, when braking a torsional moment of 300 Nm arises as shown in Figure 6.13. It is of great importance that the applied force on the load cell is perpendicular, so therefore something has to keep the spindle end hole horizontal but still let the spindle move back and forth easily.

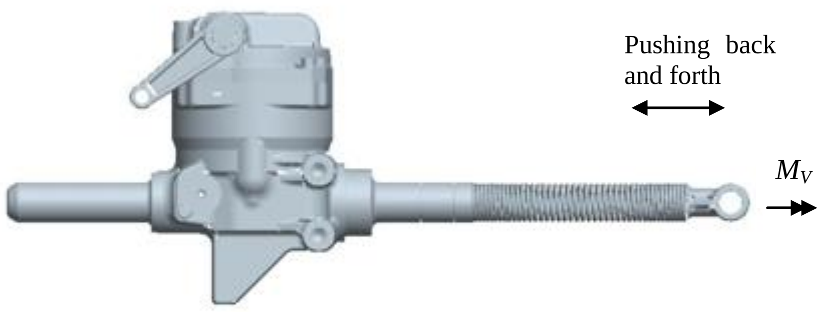


Figure 6.13 Torsional moment of the spindle.

Concepts

After a short discussion it was decided that the currently used fixture for the holes at the spindle end would be used in the new test rig as well. Faiveley has a number of different versions in their lab but they all have the same fundamental appearance as seen in Figure 6.14.

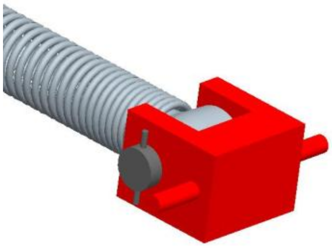


Figure 6.14 Fixture for spindle hole.

As can be seen there is a small axle with a pin that keeps the fixture aligned with the spindle hole. Two M12 screws on both sides (they are cut in half in the figure) of the fixture are then supported so that the front side stays vertical and can lead the force correctly to the load cell. Below three different concepts that were generated are described.

A3 - Rail guides with a carriage
In this solution the screws of the spindle fixture are attached to two carriages, one on each side. The carriages are then mounted to rail guides who permit them to run linearly back and forth. A suitable type of rail and carriage is LLRHS 20 from SKF. The carriages are able to carry loads in all directions including moments that seek to tilt the carriage [4 p. 59]. The rails lead the spindle very smoothly back and forth in the horizontal direction.

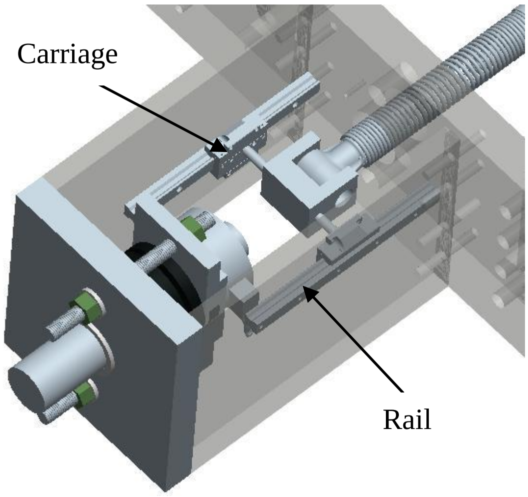


Figure 6.15 Concept A3, rail guides with a carriage.

B3 - Wheels
In today's rigs the rotating spindle problem is solved by using wheels, as is done in Model I. In this concept smaller versions of that concept are used. The wheels are made of hard plastic, their surface is flat and they are intended to roll against a flat surface that is milled in the two extension plates, see Figure 6.16 below. Within the plastic wheel there is a ball bearing which is force fit on to a plastic bushing and

attached to the screw. It is held together in one direction by an M12 nut. The wheel is locked in the other direction by a flange which makes it possible to be mounted from only one side.

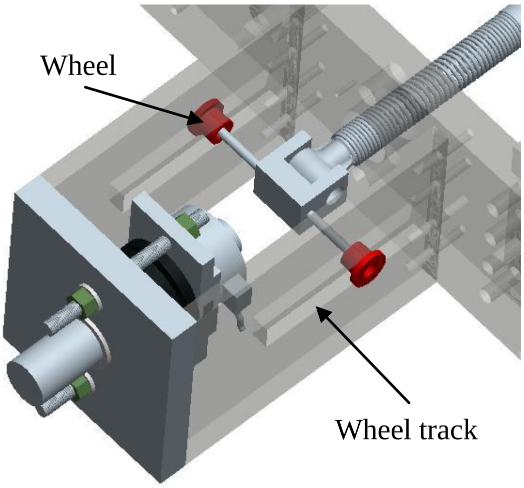


Figure 6.16 Concept B3, wheels.

C3 - Slide plates

This concept consists of two hardened plastic plates that slide back and forth in a slot in the extension plates as shown in Figure 6.17. The material is chosen so there is as little friction as possible and a suitable candidate material could be Delrin 100. This plastic is hard with a yield strength of 71 MPa [5 p. 1]. It is commonly used in gears and electrical mechanisms. In time the plastic plates might have to be replaced due to wear.

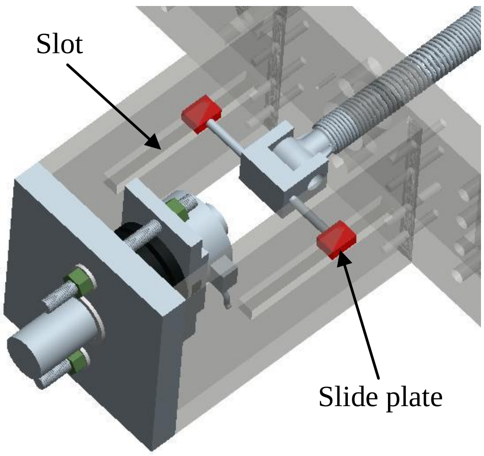


Figure 6.17 Concept C3, slide plates.

Concept scoring

A concept scoring was used because a weighted score was considered necessary. It was decided that three criteria were enough for this scoring, in which the criteria *Function* describes how well the concept fulfills its purpose; how easily you can attach a TBU and so on. The wheel concept was chosen as a reference due to its similarity to the existing solutions in use at Faiveley today. In **Table 6.3** the results can be seen. Concept B3 was ranked as number one.

Table 6.3 Concept scoring for rotating spindle problem

Criteria		**Concept**					
		A3		**B3**		**C3**	
	Weight	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score
Cost	0.5	1	0.5	**3**	1.5	4	2.0
Function	0.3	4	1.2	**3**	0.9	2	0.6
Resistance to wear	0.2	5	1.0	**3**	0.6	1	0.2
Total score			2.7		3.0		2.8
Rank			3		1		2

Commentary

First of all it has to be said that the three criteria integrates but it is hard to avoid. A3 was without a doubt the best solution if it came to functionality and durability. But because of its expensive design it was too excessive for this task. C3 was considered as the least expensive solution but did not get rating 5 because it might have to be replaced after a number of cycles. B3 had the advantage that it was similar to today's solution that functions properly.

6.4 Remaining design issues

A few different segments of the final pre design remains to be designed; how the various parts of the rig will be joined together and the dimensions of the different plates used. So far these dimensions have been arbitrarily chosen. It was decided to use steel as the construction material due to its low price and good characteristics in machining.

The old test rig models, Models I – III, are made of FORMAX steel from manufacturer Uddeholm Svenska AB [6]. This is a high strength steel with good mechanical properties. The plate thicknesses on the new rig were made about the same as those previously used on the old rigs with one exception. The front plate is longer than 1030 mm which is the maximum length of a FORMAX plate. Hence the material of the front plate had to be changed. A discussion was held with Uddeholm[6] and led to the decision to use UHB 11 steel in the front plate. All dimensions and material of the four load-bearing plates are presented in Table 6.4 on the next page.

[6] Telephone call on October 6th 2009 between M. Lindström and a representative of Uddeholm Göteborg AB. The topic was a choice of material.

Table 6.4 Plate dimensions.

Plate	Dimensions [mm] [L x W x T]	Material
Front plate	1170 x 315 x 40	UHB 11
Rear plate	990 x 300 x 32	FORMAX
Side plates (2)	700 x 300 x 32	FORMAX

One of the main needs stated by both the managers and the lab technicians was that the rig is made flexible. On the subject of joints this was interpreted as the rig must be easily demountable. Hence welding the plates together or an adhesive joint are not optional. Then remains screw joints. Screw joints are commonly used and are relatively cheap as the amount of work to the plates is low. Holes have to be drilled and threaded which is easy and fast with any type of CNC mill.

On the old test rig models, M20 screws have been used to join the thick plates together. Because of this the same dimension of screws will be used to join the front and rear plates to the side plates. Four M20 screws are considered sufficient per joint. Figure 6.18 shows the various screw joints of the rig.

To join the stiffening plates to the side and front plates four M12 screws are used on each stiffening plate. Two screwed into the front plate and two into the side plate.

To join the extension plates to the front plate, M12 screws are used. According to common screw joint rules [7 p. 51] a greater number of small screws is preferred to a few larger sized screws. Hence four M12s per extension plate will be used, a total of eight for the complete extension. Compared to the four M20s that will hold the thick plates together these eight screws will be subjected to the force from only one TBU whereas the M20 joints could be subjected to the double force.

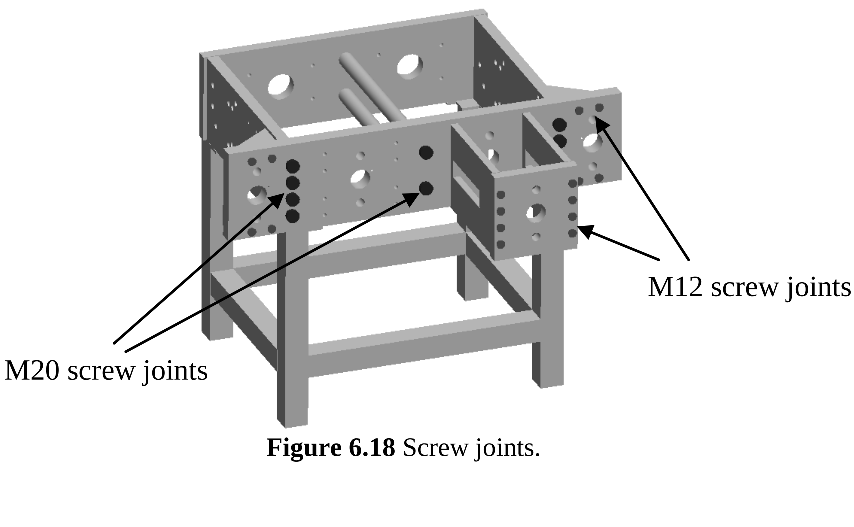


Figure 6.18 Screw joints.

Commentary

Even though all problems were solved separately the other existing components of the test rig of course had to be considered while generating different concepts. Sometimes the awareness of the overall design restricts your imagination and the best solution might not always be found. This problem is discussed further in the final conclusions, see section 10.

If you compare concept AC chosen in section 5.3 with the final design the difference is quite large. The reasons for this are many. When generating the first concepts the main focus was to design a test rig with five stations able to test the several numbers of different TBUs. Gradually changes had to be done due to other target specifications. For example the test rig could have been designed in a more compact way but then the specification concerning clearance for mounting would not have been fulfilled. Another reason for many changes is the simple fact that the dimensions first set did not conform to the standard dimensions sold by Uddeholm and would not be justified in a cost efficient point of view. For instance the height of the plates was chosen to 270 mm but had to be changed to 300 mm and 315 mm respectively.

7 Calculations

As with any type of structure subjected to any kind of external or internal load it is of upmost importance that it does not fail. The failure can occur due to a number of different cases of loads; tension, shearing, creep, fatigue, buckling and more. To cover most of these in a reasonable amount of time it was decided[7] to primarily focus on an overall structural analysis of the rig, a fatigue analysis in areas with special geometries and finally the rig's screw joints. During the same meeting it was agreed that the necessary safety factor against fatigue in any area would be set to 1.5.

If the rig should theoretically fail in any of the aspects it will be re-dimensioned until it can handle the loads in a satisfying way. The parts of the rig that can easily be re-dimensioned are the plates and its thicknesses and the dimensions of the various holes on the rig.

7.1 Structural analysis in ANSYS Workbench

A complete functional design of the test rig was presented in section 5.3. From this design a 3D model was made using Pro Engineer Wildfire 3. A finite element method (FEM) application would be needed for the actual analyses. Therefore the 3D model was imported to ANSYS Workbench before any analyses could be performed.

7.1.1 The mesh

In order for the FEM analysis to produce a credible result it is very important that the element size and configuration – called the mesh – are properly set. Too few elements and the result will not be good because of the occurrence of singularities around holes and other complex geometries. Too many elements and the result will be good but the time to perform the calculations by the computer can be very long.

A simple way to make an efficient mesh in ANSYS is to carry out a mesh convergence. Here the element size and total number are set quite low and the program itself increases them in iterations until the result is no longer affected by the elements. In this convergence the maximum percentage of allowed change in the result between iterations was set to 4 %. When the change in results from one calculation to the next was less than 4 % the mesh was considered satisfying.

To simplify the analysis and reduce the number of elements, unnecessary parts of the rig were removed. The force from a TBU is lead into the front plate through the load cell and all the disc springs and the hardened washers. Hence this whole package can be reduced down to the last washer with the force applied directly to it.

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[7] Meeting on September 24th 2009. Participating were J. Stridh, M. Lindström and F. Blennow. The topic was what types of analyses were to be carried out.

7.1.2 Boundary conditions

There are four TBUs that can be in place on the test rig at the same time. As the rig is symmetrical around its center (the round bars in the middle) and the maximum force per TBU is 70 kN there are a total of eight different cases of loads, see Figure 7.1 where the rig is shown from above.

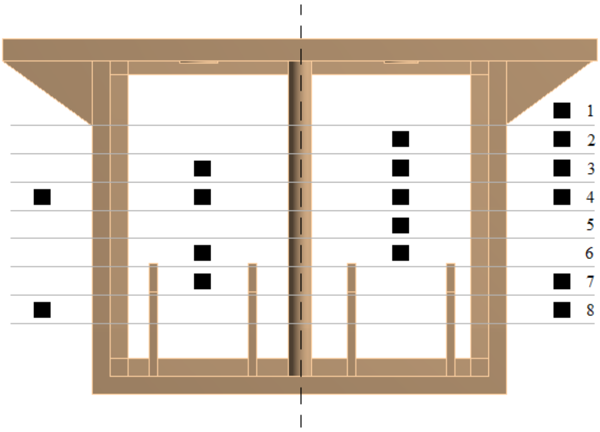


Figure 7.1 Different cases of loads.

Each line and number corresponds to a specific case of load and each square black dot corresponds to a TBU being in that place. As the maximum allowed deformation is limited to 1.4 mm the case of load which produces the largest deformation is the one that needs to be focused on.

To determine which this is each of the eight cases were tested in ANSYS. The boundary conditions were set as follows; the bottom of the rig's foundation was set as *Fixed Support* and the forces were applied to the washers on the front plate. Furthermore the counter forces were applied to the holes in the mounting plates on the rear plate and the mounting holes on the side plates, these counter loads also included the weight of the TBUs (65 kg). The full setup of boundary conditions for a specific case (no. 7) is presented in Figure 7.2.

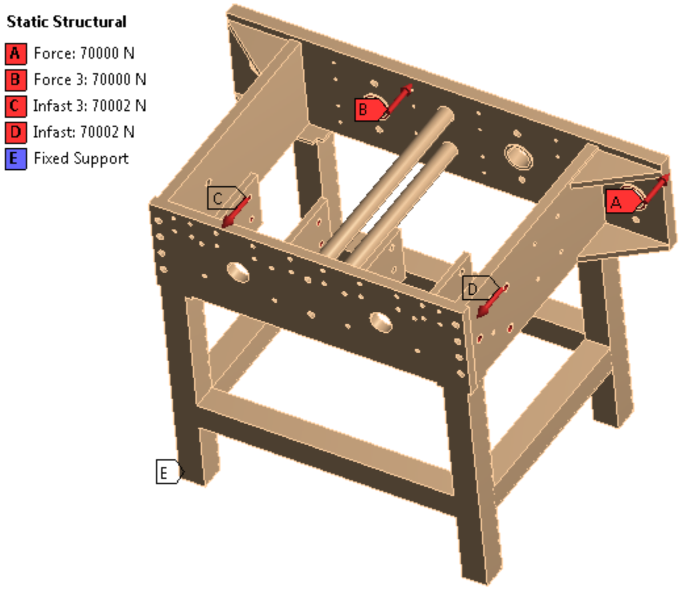


Figure 7.2 Boundary conditions.

7.1.3 Results

When all eight cases had been analyzed it was found that no. 7 produced the largest deformation around any of the washers in all of the cases. This was in fact what could be expected as the theories behind the analyses suggest the same thing. The front plate can be thought of as a beam and the two side plates and the round bars can be removed and replaced by supports. If the beam is put on these three supports according to Figure 7.3 and forces are applied on two places and super positioned together, the deformation on the right side end will be large. This is because both applied forces act to bend the beam in the same way.

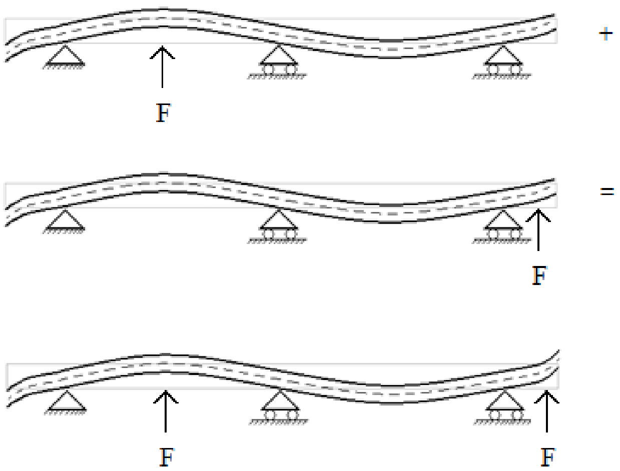
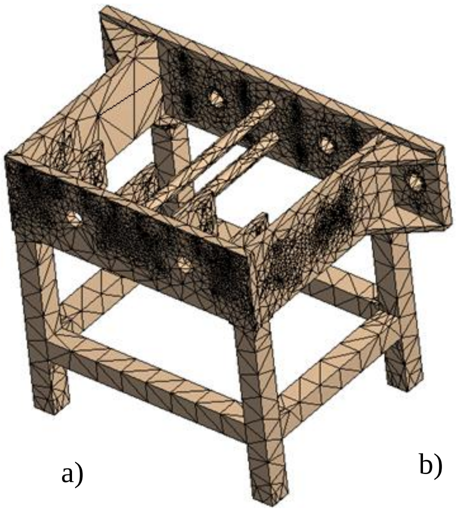


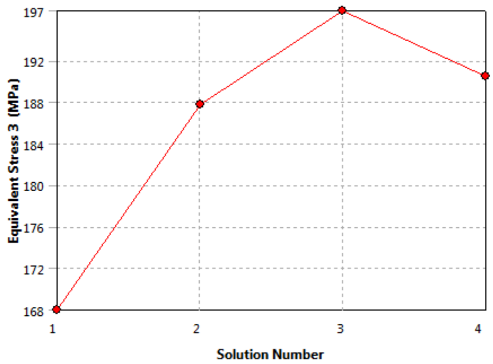
Figure 7.3 Bending of a beam.

The following results of the calculations are those carried out with the no. 7 case of load. The maximum value of the stress in the reviewed area converged after four refinements. The number of elements rose from just over 31 000 to almost 137 000 throughout the rig. As can be seen in Figure 7.4 the mesh is concentrated with a high elemental density around the holes on the plates. The foundation is meshed with a low elemental density as this area is relatively unaffected by the loads.



a)

b)



	Equivalent Stress 3 (MPa)	Change (%)	Nodes	Elements
1	167,96		58110	31068
2	187,76	11,134	68629	37640
3	196,81	4,7048	99751	58145
4	190,48	-3,2684	212187	136947

Figure 7.4 a) The mesh. b) Convergence.

With a satisfying mesh the results of the structural analysis could be reviewed. The total deformation in the area where the load cell is to be mounted on the side of the front plate ran to 0.64 mm, see Figure 7.5. Compared to the maximum allowed deformation of 1.40 it has a safety factor of

$$\frac{\Delta_{allowed}}{\Delta_{actual}} \Rightarrow \frac{1.40}{0.64} = 2.19 \quad (7.1)$$

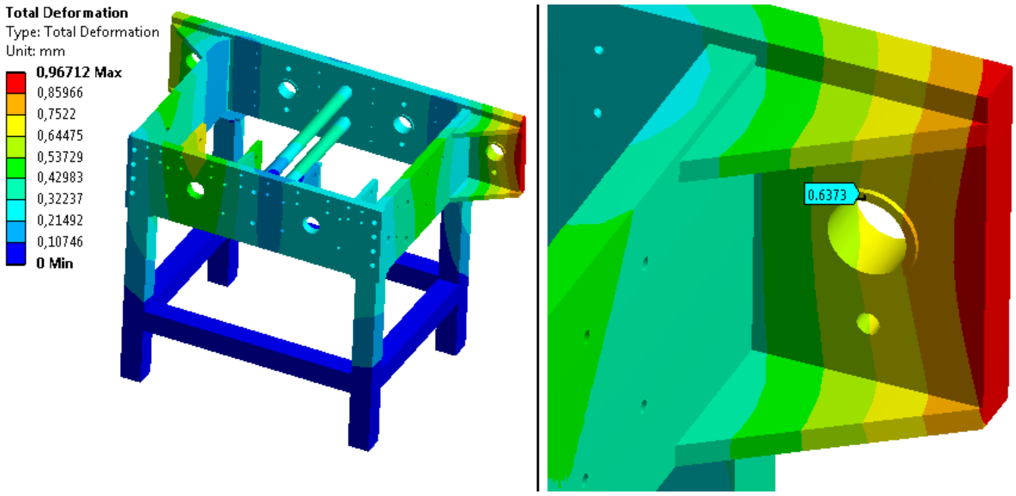


Figure 7.5 Total deformation when using 40 mm and 32 mm plates.

Even though the deformation is larger at the extreme end of the front plate the deformation at the location of the load cell is the one that matters. This is because the accuracy of the measurements by the load cell cannot be guaranteed if the mounting flexes too much.

The safety factor was considered to be just about enough, however Faiveley wished it would be a little bit higher. To determine the how much the thickness of the various plates influences the results it was decided to re-do the ANSYS analysis with thicker plates. The old rig of type Model I uses 50 mm plates and a new CAD model was made with a front plate with a thickness of 50 mm and the rest of the plates were made 40 mm thick. As for the rest nothing changed, the new model was meshed in the same way and the loads were applied equally. Below a comparison of the stresses and deformations of both sizes of the rigs are shown in Figures 7.6 – 7.7.

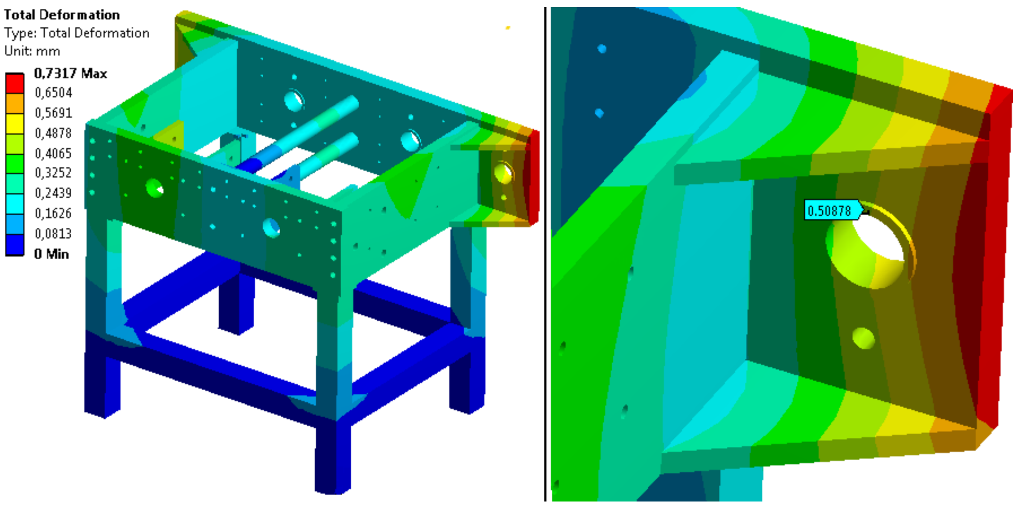


Figure 7.6 Total deformation when using 50 mm and 40 mm plates.

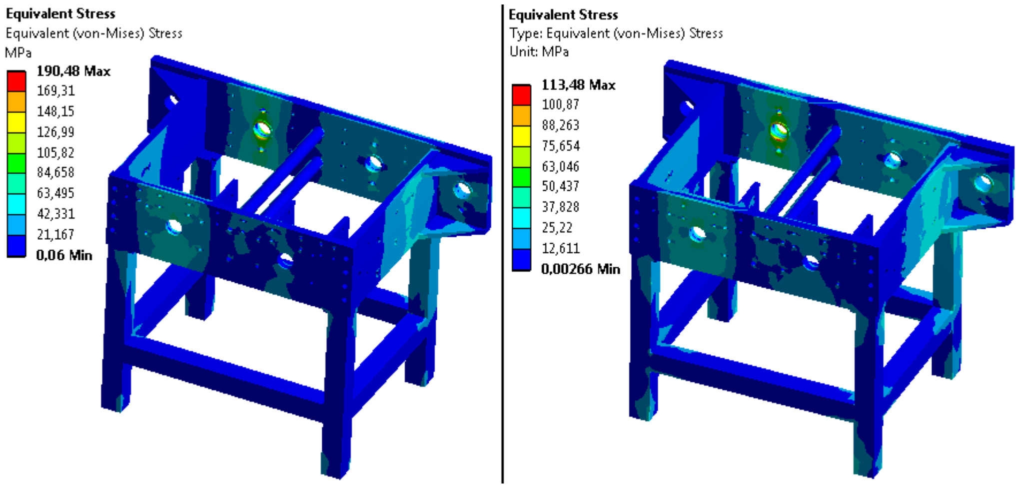


Figure 7.7 Equivalent stresses for. 40/32 plates (left) and 50/40 plates (right).

The deformation at the location of the load cell dropped from 0.64 to 0.51 which increased the safety factor from 2.2 to 2.7 using equation (7.1). The equivalent von Mises stresses throughout the rig were lower in the thicker rig. At the area of the applied force to the middle station the stress dropped from 190 MPa to 113 MPa. These stresses fall below the yield strength of 340 MPa which is what the material UHB 11 can withstand according to Uddelholm [6]. When reviewing the result an agreement came with Faiveley to use the thicker plates. Especially since the difference in price between the two versions is low, only 1366 SEK[8] in total.

Commentary

The increase in the plate thicknesses is motivated by the increase in stiffness and the insignificantly higher cost. The total dimensions do not get much bigger nor is the increased weight any problem. According to ANSYS, with a density of 7850 kg/m^3 throughout the rig, they weigh 420 kg and 498 kg respectively. Both fall below the marginal value of 1000 kg which is the target specification.

In another aspect the thicker plates probably makes the service life longer which is preferred as the new test rig is to be used for a long time.

7.2 Fatigue

One of the test rig's primary functions is to study how the TBUs applied braking force changes over time. This will require a great number of cycles with one cycle being the time it takes to apply the force of 70 kN and keeping it applied until the force is removed. One cycle takes about thirty seconds if pneumatics is used and the TBUs' service brake will be tested for up to half a million cycles. The parking brake and its emergency release cable is usually tested for 3 000 cycles each. The reason for the slow cycle time while on pneumatic drive is that if it gets higher, gaskets and other parts of the TBU can fail.

[8] Telephone call on October 14th between J. Stridh and a representative of Uddeholm. The topic was the costs of different materials and dimensions.

The plates themselves are thick enough to resist the influence of fatigue but around the many holes through the plates stress concentrations might occur. Therefore it is of great importance that the most exposed areas are examined and dimensioned in such a way that they can withstand an unlimited amount of cycles.

The fatigue calculations are carried out as taught by the Division of Solid Mechanics at LTH and Handbok i hållfasthetslära [8]. First the area of interest is decided and then the case of load is determined. The next step is to calculate the nominal stress in that area using solid mechanics analyses or if it is a simple case, just find the elemental case to which the case of load corresponds.

When the nominal stress is known it is time to gather information about the material itself. Its mechanical limits are required such as yield strength, ultimate stress and the pulsating fatigue characteristics. The last limit is needed as the force from the TBUs will pulsate between zero and 70 kN.

The mechanical limits are then reduced by certain shape factors which rely on the geometrical aspects of the area of interest and its surface roughness. When the limits are reduced they are compared to the nominal stress to see if the area has a fatigue limit or not. This comparison is preferentially done in so-called Haigh-diagrams where amplitude and mean stresses are calculated.

In this analysis the Haigh-diagram will also include the actual stress in the area of interest calculated in the FEM analysis using ANSYS. This is done in order to compare the theoretical value to the one produced computationally. Probably the true value lies somewhere in between.

Below the data gathered about the different materials are presented in Table 7.1. Exact fatigue data could not be found for either FORMAX or UHB 11. However according to manufacturer Uddeholm their characteristics match a standard construction steel of type SS 1650 for which data can be found in Broberg [8 p. 372 – 373]. These values are marked with a *.

Table 7.1 Characteristics for materials.

Characteristic	Material	
	FORMAX	**UHB 11**
Yield strength [MPa]	320	340
Ultimate strength [MPa]	560	640
Young's modulus [MPa]*	$206 \cdot 10^3$	$206 \cdot 10^3$
Alternating stress (bending, $\boldsymbol{\sigma_{ub}}$) [MPa]*	±270	±270
Pulsating stress (bending, $\boldsymbol{\sigma_{ubp}}$) [MPa]*	240±240	240±240

Below the different areas of the test rig that were deemed interesting from a fatigue point of view are presented.

7.2.1 Holes for M20 and M12 screws

There are three areas where holes are made in the front plate in which to screw the M20s; at both of the ends of the plate where the side plates are connected and in the middle where the two round bars are connected. The cases of loads are different for the two. Furthermore the extension plates are joined by eight M12 screws which also require holes through the front plate.

For the holes near the ends, the case of load is indexed (1) whereas the case of load for the two middle holes is indexed (2). The case of load for the holes for the M12s is indexed (3). They are presented in figures 7.8 –7.9 below.

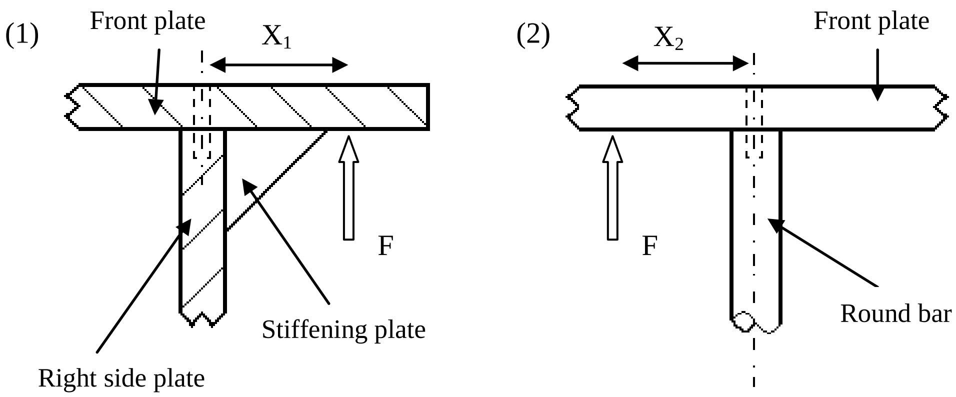


Figure 7.8 Cases of load (1) and (2).

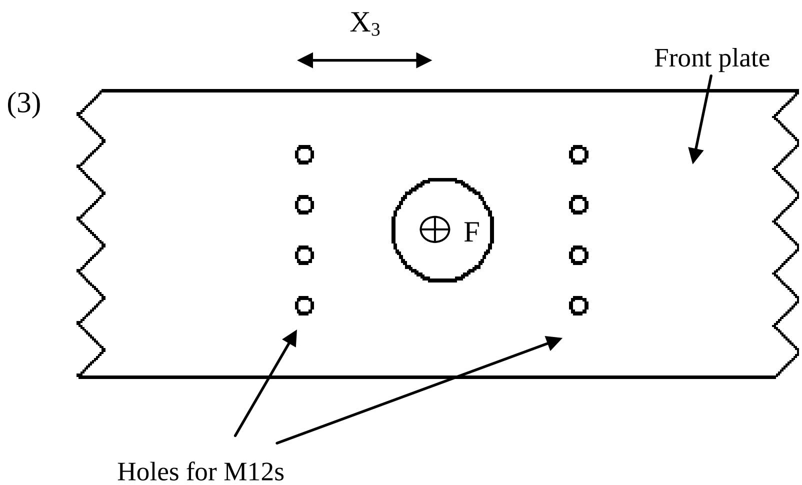


Figure 7.9 Case of load (3).

For (1) – (3) the force F will produce a pulsating bending moment of $M = F \cdot x_i$ where the holes are located on the front plate. In this design $x_1 = 120$ mm, $x_2 = 225$ mm and $x_3 = 125$ mm. These three cases of loads correspond to a flat plate with a hole through it subjected to a bending moment which is a shape factor case in Broberg [8 p. 356]. From that elemental case the nominal stress can be calculated as:

$$\sigma_{nom} = \frac{6M}{(B-2r)h^2} \quad (7.2)$$

B is the height of the plate; in this case it is one fourth of 315 mm for (1) and (3) because there are four holes and one half of 315 mm for (2) because there are two holes. *2r* is the diameter of the holes and h is the thickness of the plate; in this case 21 mm for M20s and 13mm for the M12s. The thickness of the front plate is 50 mm.

$$(\sigma_{nom})_1 = \frac{6 \cdot 70\,000 \cdot 0.12}{\left(\frac{0.315}{4} - 0.021\right) \cdot 0.05^2} = 343.2\ MPa \quad (7.3)$$

$$(\sigma_{nom})_2 = \frac{6 \cdot 70\,000 \cdot 0.225}{\left(\frac{0.315}{2} - 0.021\right) \cdot 0.05^2} \cdot \frac{1}{2} = 137.4\ MPa \quad (7.4)$$

$$(\sigma_{nom})_3 = \frac{6 \cdot 70\,000 \cdot 0.125}{\left(\frac{0.315}{4} - 0.013\right) \cdot 0.05^2} \cdot \frac{1}{2} = 159.8\ MPa \quad (7.5)$$

The nominal stress for case (2) is divided by 2 as the applied force from the brakes is considered to be absorbed equally by the round bars and by one of the side plates. In case of load (3) the force will instead be absorbed by two extension plates and thus the stress around the M12 holes will be halved. In reality it is unclear exactly how much of the force is absorbed by each plate or bar, a seemingly probable assumption of 50 % was used because of its simplicity.

From the nominal stress the mean and amplitude stresses are achieved. Since F_{min} = 0 kN both σ_{mean} and $\sigma_{amplitude}$ equal $\frac{\sigma_{nom}}{2}$.

$$(\sigma_m)_1 = (\sigma_a)_1 = \left(\frac{\sigma_{nom}}{2}\right)_1 = 171.6\ MPa \quad (7.6)$$

$$(\sigma_m)_2 = (\sigma_a)_2 = \left(\frac{\sigma_{nom}}{2}\right)_2 = 68.7\ MPa \quad (7.7)$$

$$(\sigma_m)_3 = (\sigma_a)_3 = \left(\frac{\sigma_{nom}}{2}\right)_3 = 79.9\ MPa \quad (7.8)$$

The next step is to acquire the shape factors in order to reduce the fatigue data that was presented in Table 7.1. The first factor, K_t, is taken directly from the diagram of the shape factor case in Broberg [8 p. 356]. The notch sensitivity, *q*, is taken from figure 25.9 in Broberg [8 p. 294]. It is the same for (1) and (2) and with an ultimate strength of 340 MPa and a radius of 10 mm it is 0.9. For (3) it gets approximately 0.88. With K_t and *q* known the fatigue stress-concentration factor K_f is calculated as

$$K_f = 1 + q(K_t - 1) \quad (7.9)$$

The remaining two factors, K_d and K_r, are taken from figures 25.10 and 25.12 in Broberg [8 p. 296 – 297]. The shape factors and their values are presented in Table 7.2 below.

Table 7.2 Shape factors for (1) – (3)

Shape factor	(1)	(2)	(3)
$\boldsymbol{K_t}$	1.95	2.30	2.55
$\boldsymbol{q}$	0.90	0.90	0.88
$\boldsymbol{K_f}$	1.86	2.17	2.36
$\boldsymbol{\frac{1}{K_d}}$	0.96	0.96	0.98
$\boldsymbol{\frac{1}{K_r}}$	0.88	0.88	0.88

The fatigue data is reduced from (σ_m, σ_a) to

$$\left(\sigma_m, \sigma_a \frac{1}{K_f \cdot K_d \cdot K_r}\right) \quad (7.10)$$

The result is presented in the Haigh-diagrams below where the nominal stress value is marked with ⊕. The values calculated in ANSYS are marked with × and taken from Figure 7.10 which shows the stresses in all three cases of loads, (1) – (3). Additionally the safety factors for each case are also presented. When dealing with a pulsating load the safety factor that considers both the amplitude and the mean safety factors had to be calculated. These are calculated as

$$\frac{\sigma_a}{\sigma_{a_{allowed}}} \quad (7.11)$$

$$\frac{\sigma_m}{\sigma_{m\,allowed}} \quad (7.12)$$

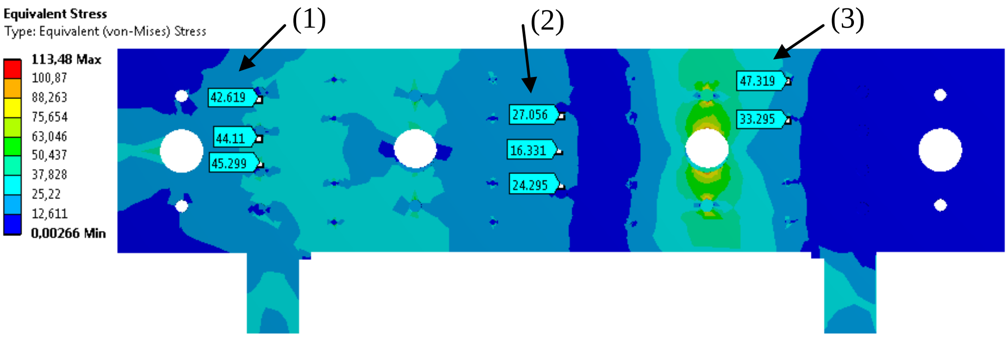


Figure 7.10 Stresses for cases (1) – (3) in ANSYS.

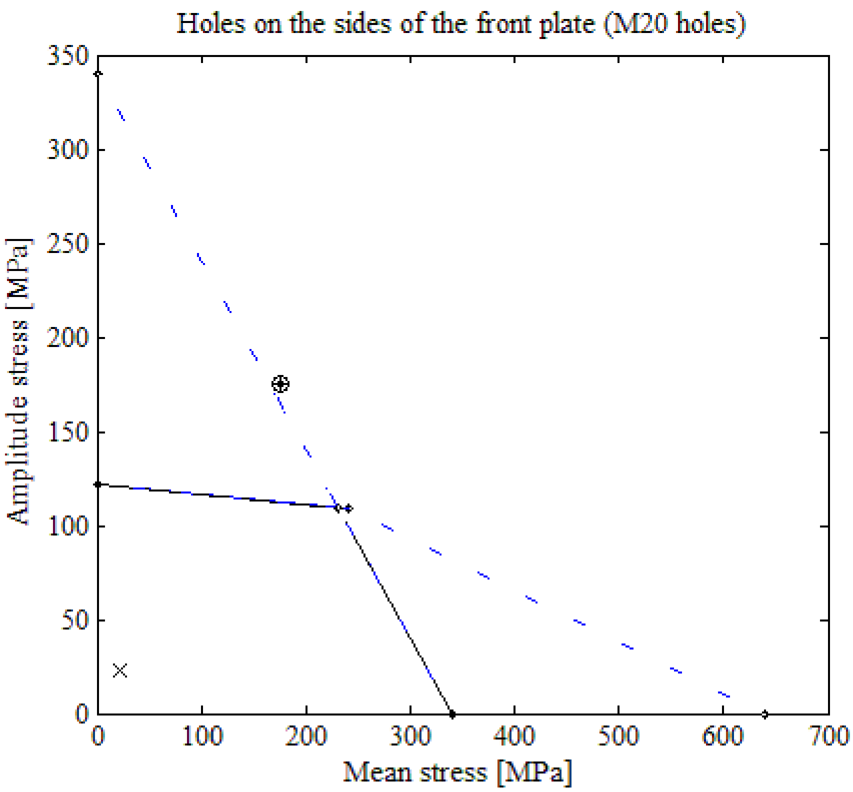


Diagram 7.1 Haigh-diagram for case of load (1).

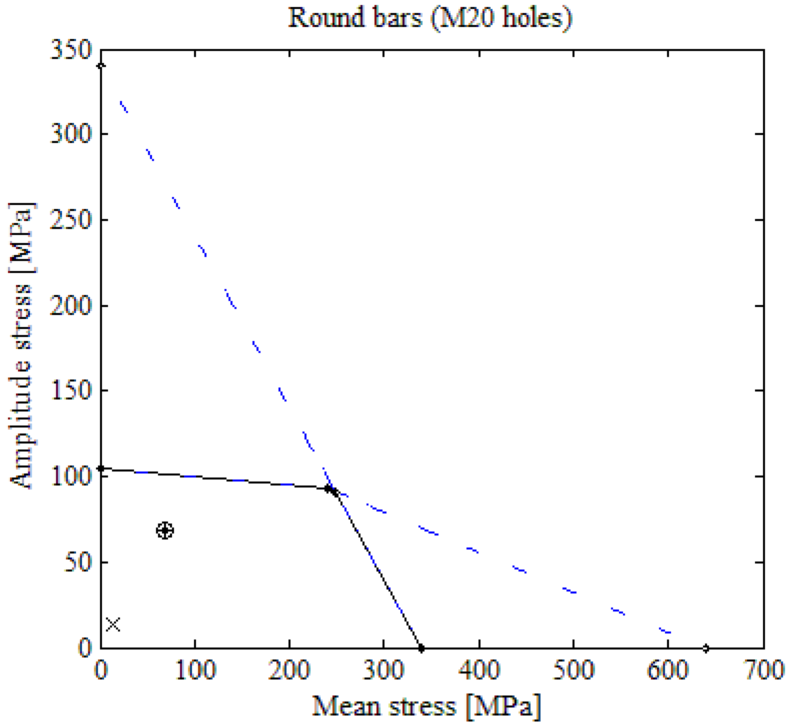


Diagram 7.2 Haigh-diagram for case of load (2).

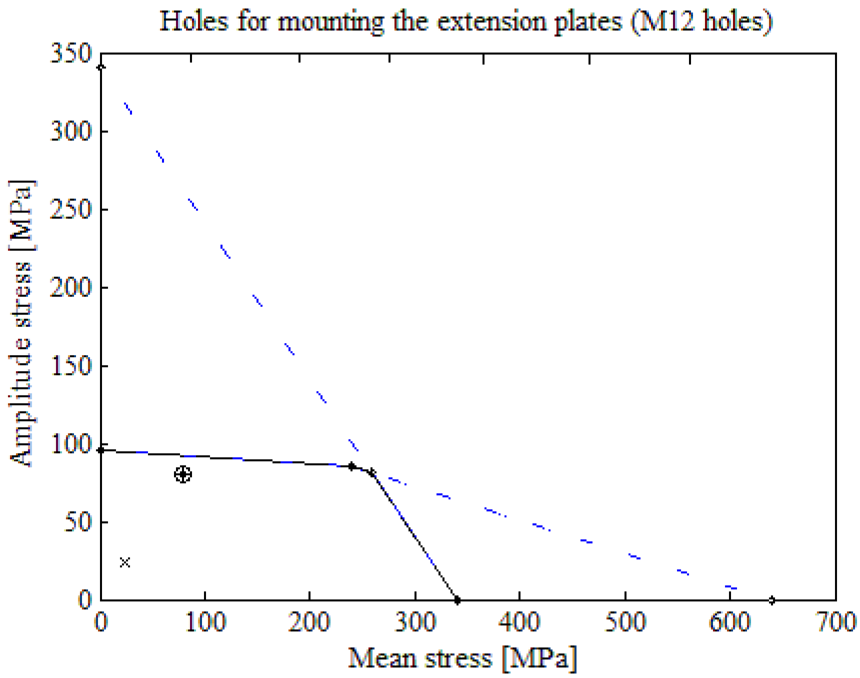


Diagram 7.3 Haigh-diagram for case of load (3).

Table 7.3 Safety factors against fatigue.

Case of load	**Safety factors**	
	Theoretical	From ANSYS
(1)	0.66	5.11
(2)	1.45	7.25
(3)	1.15	3.89

Commentary

The most noticeable about the result of the theoretical fatigue calculations is that the holes for the M20s on the sides of the front plate end up over the limit. Meanwhile according to ANSYS the stresses around the same holes only run up to about one fourth of the fatigue limit. The main reason for the difference is that in the theoretical analysis no consideration was taken to the stiffening plates. These carry a substantial part of the load directly into the side plates before it reaches the M20 holes. As this is a statically indeterminable case of load it is hard to calculate it theoretically. In this analysis the worst case scenario was considered where 100 % of the load reached the holes. Moreover it is hard to find elemental cases in the literature that fits the actual cases of loads on the rig.

Furthermore both the other cases of loads manage to stay under the fatigue limits, both theoretically and with the stresses calculated using FEM in ANSYS. The rig should now be dimensioned in such a way that the service life is long and that holes and joints hold for an unlimited amount of cycles.

7.3 Screw joints

The next step in the calculation process was to determine whether the chosen screw joints throughout the test rig could withstand the forces applied to them. It should be mentioned that even though plenty of research has been done about screw joints there is no single true knowledge in this complex area but new discoveries arise all the time. The calculations below are carried out as taught in Å. Burman’s Skruvförband [7].

Even though the forces on the rig often are eccentric to the joints only balanced cases have been calculated. The reason for this is that the combination of very stiff plates and large clearing holes for the screws makes the bending effect less important.

The first step in the calculations of the screw joints was to define the stiffness coefficients for each screw size and material. To be able to carry out these calculations standard dimensions tabulated in Teknikhandboken [9] as well as some other coefficients and parameters were used. The calculations can be followed through tables 7.4 – 7.7.

Table 7.4 Tabulated values of screws.

Dimension	M12	M20	Note
d	12	20	Diameter [mm]
d_1	10.106	17.294	Inner diameter [mm]
d_2	10.863	18.376	Average diameter [mm]
d_h	13.5	22	Hole diameter [mm]
d_y	19.2	32.6	Screw head diameter [mm]
P	1.75	2.5	Pitch
L_k	40 mm and 50 mm respectively		Length of pinched material [mm]

Table 7.5 Estimated coefficients.

Coefficient	Note
$E_g = E_s = 2.1 \text{ x } 10^5$	Estimated Young's modulus for the screw and the material [N/mm^2]
$\mu_g = \mu_u = 0.15$	Estimated friction coefficient for the screw and the material[9]

Table 7.6 Equations used to achieve stiffness coefficients.

Equation	Note
$A_s = \frac{\pi}{4}\left(\frac{d_2 + d_3}{2}\right)^2 \quad (7.13)$	Tension area for the screw, where d_3 is defined as $d_3 = d_1 - \frac{H}{6}$ and H as $H = \frac{d - d_1}{2}\frac{8}{5}$
$d_a = d_y + L_k \ (7.14)$	Diameter of pinched material
$A_{ers} = \frac{\pi}{4}(d_y^2 - d_h^2) + \frac{\pi}{8} d_y (d_a - d_y)\left(\left(\left(\frac{L_k d_y}{d_a^2}\right)^{\frac{1}{3}} + 1\right)^2 - 1\right) (7.15)$	Substitute area for pinched material
$C_g = \frac{E_g A_{ers}}{L_k} \quad (7.16)$	Stiffness coefficient of material
$C_{s1} = \frac{E_s A_s}{L_k} \quad (7.17)$	Stiffness coefficient of screw regarding the elongation
$C_{s2} = \frac{E_s \pi d}{1.6} \quad (7.18)$	Stiffness coefficient of screw regarding the deformation of screw head and nut
$C_{s3} = \frac{E_s \pi d_1^2}{2 d} \quad (7.19)$	Stiffness coefficient of screw regarding the screw part inside the nut
$C_s = \frac{1}{\frac{1}{C_{s1}} + \frac{1}{C_{s2}} + \frac{1}{C_{s3}}} \quad (7.20)$	Total stiffness coefficient of screw
$C_t = \frac{C_s C_g}{C_s + C_g} \quad (7.21)$	Stiffness coefficient of screw joint
$\varphi = \frac{C_s}{C_s + C_g} \quad (7.22)$	Percentage of total stiffness the screw make up to

[9] Consultation with senior instructor L. Vedmar on October 22nd.

When putting in all the numbers for the different sizes of screws the following results were achieved. It should be noted that there are different lengths of the pinched material. For screw joints used in the front plate $L_k = 50$ mm has been used whereas $L_k = 40$ mm has been used elsewhere.

Table 7.7 Stiffness coefficients.

	M12 ($L_k = 40$)	**M12 ($L_k = 50$)**	**M20 ($L_k = 40$)**	**M20 ($L_k = 50$)**
C_g	$3763\ {}^{kN}/_{mm}$	$3011\ {}^{kN}/_{mm}$	$6821\ {}^{kN}/_{mm}$	$6281\ {}^{kN}/_{mm}$
C_s	$355\ {}^{kN}/_{mm}$	$296\ {}^{kN}/_{mm}$	$907\ {}^{kN}/_{mm}$	$771\ {}^{kN}/_{mm}$
C_t	$324\ {}^{kN}/_{mm}$	$269\ {}^{kN}/_{mm}$	$801\ {}^{kN}/_{mm}$	$689\ {}^{kN}/_{mm}$
φ	0.09	0.09	0.12	0.11

The next step is to define the bolt pretension. This depends on the characteristics that the screws have but above all on how accurate you are when assembling the test rig, i.e. if a dynamometric wrench is used or not. For the calculations below the bolt pretension was set as suggested by Teknikhandboken [9 p. 63] for 8.8 quality screws. The values are shown in Table 7.8. These forces are set to obtain 65 % of the yield strength.

Table 7.8 Bolt pretension values.

Screw	**Bolt pretension, F_f**
M12	35 kN
M20	102 kN

7.3.1 M20 screw joints in the front plate

There are three different screw joints with M20 screws in the front plate, the two side plates attached to the left and to the right as well as the round bars in the middle. Because of the obvious symmetry it was only necessary to do two calculations, one of the side plates and one for the round bars. The worst scenario for the side plates' screw joints are when two TBUs are breaking simultaneously as shown in Figure 7.11, where the forces, F_1 and F_2, are 35 kN and 70 kN respectively. The reason for F_1 being 35 kN is that the other half of the force is estimated to be absorbed by the round bars.

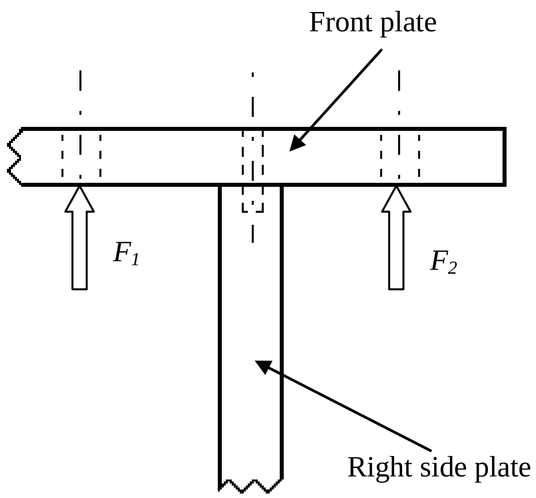


Figure 7.11 Screw joint fastening front plate to right plate.

The force, F_l, absorbed by each screw then becomes

$$F_l = \frac{Force}{No.of\ screws} \Rightarrow \frac{35\ [kN] + 70\ [kN]}{4} = 26.25\ kN \quad (7.23)$$

For the screw joint between the round bars and the front plate which is shown below in Figure 7.12, where F is 35 kN with the same reasoning as above, F_l instead becomes

$$(F_l)_2 = \frac{35\ [kN] + 35\ [kN]}{2} = 35\ kN \quad (7.24)$$

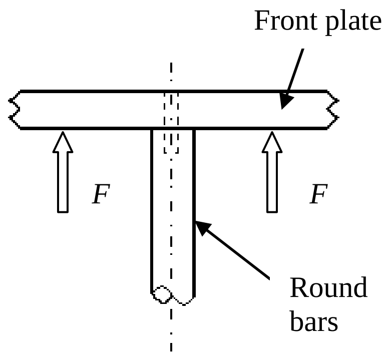


Figure 7.12 Forces applied on both sides of the round bars.

The load on the joints holding the round bars in place is higher than that on the joints on the side plates. Because of this only the joint on the round bars will be considered from now on.

To achieve the effective stress in the screws, according to von Mises, the following equations were used

$$F_s = F_f + F_l \varphi \quad (7.25)$$

$$M_g = F_s\left(0.16\, P + 0.58\, \mu_g\, d_2\right) \quad (7.26)$$

$$\sigma_{Mises} = \sqrt{\left(\frac{F_s}{A_s}\right)^2 + 12\pi \frac{M_g^2}{A_s^3}} \quad (7.27)$$

Here F_s is the total force in the screw and M_g is the thread moment. When using F_l = 35 kN, this gives the effective stress of σ_{Mises} = 542 MPa. This is lower than the yield stress of 640 MPa for 8.8 quality screws. Except the maximum effective stress, the obtained alternating stress amplitude is of high interest to make sure no failure due to fatigue will occur. The amplitude is simply calculated from the equation

$$\sigma_{amp} = \frac{1}{2}\left(\sigma_{Mises} - \sqrt{\left(\frac{F_l}{A_s}\right)^2 + 12\,\pi\, \frac{M_g^2}{A_s^3}}\right) \quad (7.28)$$

This gives σ_{amp} = 4.6 MPa which can be compared to the fatigue limit of 50 MPa for an 8.8 quality screw [10 p. 205].

7.3.2 M20 screws in the rear plate

The rear plate is joined to the side plates by four M20s, just as the front plate is. The most obvious difference between the front and rear plates is that the rear plate is shorter and not as thick, $L_k = 40$ mm. The loads from the TBUs are also absorbed differently. The TBUs on the sides are mounted directly to the side plates and therefore do not seek to disjoin the screw joint between the rear and side plates. Hence the only cases of loads that affect the rear plate are the ones where TBUs are mounted in either or both of the middle stations.

Comparing with the case above it is obvious that Figure 7.12 describes the worst case as well for the rear plate. The only difference is that $L_k = 40$ mm. Changing this parameter and using the same equations as before give $\sigma_{Mises} =$ 543 MPa and $\sigma_{amp} = 5.0$ MPa respectively. Thus the difference in L_k has a negligible effect on the resulting σ_{Mises}.

7.3.3 M12 screws in the extension plate

To fasten the extension plates to the front plate a total number of eight screws are used, four for each plate.

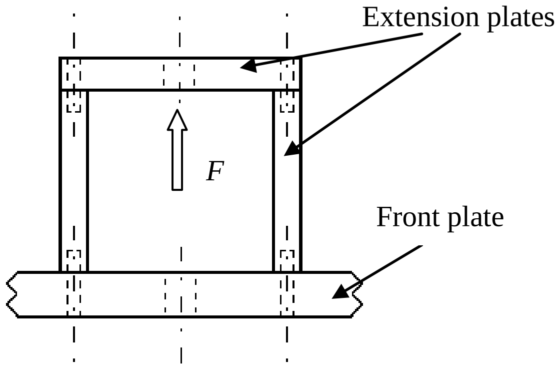


Figure 7.13 Force, F, acting on the screw joint.

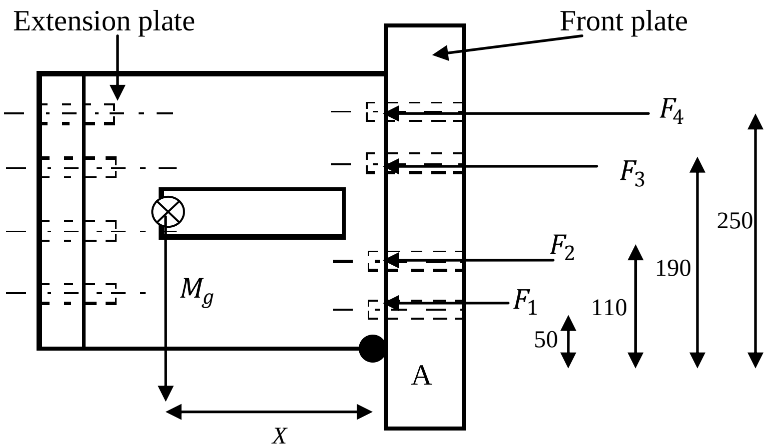


Figure 7.14 Forces acting on the screw joint due to gravity.

As shown in Figure 7.13 and Figure 7.14 there will be a load, M_g, due to the weight of the plates and the load cell package, in addition to the brake force acting on the screw joints. The brake force, F, is obviously 70 kN whereas further calculations had to be done to achieve the resulting force due to M_g.

In Pro Engineer the weight of the centre of mass was calculated as done before, by setting the density of all materials to 7800 kg/cm^3. The load M_g and its distance X from the front plate were estimated to 445 N and 300 mm respectively.

As seen in Figure 7.14 the maximum force will occur in the top screw, thus this force is the interesting one. This force, F_4, can easily be achieved by calculating a moment equilibrium around point A.

$$M_g\, x = F_1 50 + F_2 110 + F_3 190 + F_4 250 \quad (7.29)$$

$$F_1 = F_4 \frac{50}{250}, \quad F_2 = F_4 \frac{110}{250}, \quad F_3 = F_4 \frac{190}{250} \quad (7.30)$$

$$(7.29) + (7.30) \Rightarrow F_4 = \frac{M_g x}{452.8} = 295\ N \quad (7.31)$$

The force, F_l, is then determined by equation (7.32) where F_4 is divided by two as there are two extension plates. For the same reason the applied load of 70 kN is divided by eight as there are two plates and four screws per plate.

$$F_l = \frac{F_4}{2\ [screws]} + \frac{70\ [kN]}{8\ [screws]} = 8.9\ kN \quad (7.32)$$

Thus the resulting force due to M_g is very small compared to the braking force. With help of equations (7.25) – (7.28) σ_{Mises} and σ_{amp} were calculated to 544 MPa and 3.5 MPa respectively. Hence the M12 screw joints will withstand the forces applied on them without breaking.

7.3.4 M12 screws in the stiffening plates

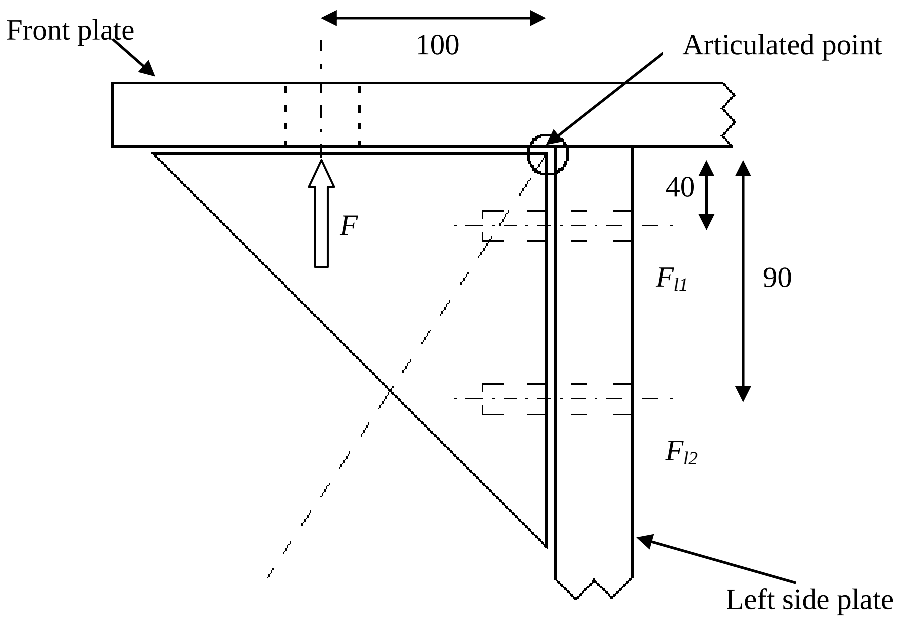


Figure 7.15 Force acting on the stiffening plate.

For simplicity the stiffening plate is considered to be articulated in a point in the upper corner. The force *F* will then create a moment around this point that the two screws must counter. *F* on one of the two stiffening plates per side is 35 kN. The screw furthest out on the stiffening plate will carry more of the load *F* than the inner one. The relation is considered linear

$$\frac{40}{90} = \frac{F_{l1}}{F_{l2}} \quad (7.33)$$

The moment produced by F and its lever of 100 mm is countered by the reaction forces from the two screws.

$$F \cdot 100 = F_{l1} \cdot 40 + F_{l2} \cdot 90 \quad (7.34)$$

From (7.33) and (7.34) F_{l2} can be calculated as

$$F \cdot 100 = F_{l2} \cdot \left(\frac{40^2}{90} + 90\right) \Rightarrow F_{l2} = 0.93F = 34.48\ kN \quad (7.35)$$

This is the actual load on the outer screw. Its size was earlier set to M12 for which the bolt pretension is 35.0 kN. From Table 7.7 φ was calculated to 0.09. With this given, the amount of the load that is absorbed by the screw is calculated by equation (7.36).

$$F_{ls} = \varphi \cdot F_{l2} = 0.09 \cdot 34.48 = 2.10\ kN \quad (7.36)$$

Lastly the effective von Mises stress and the amplitude stress are calculated from (7.25) – (7.28) giving $\sigma_{Mises} = 564$ MPa and $\sigma_{amp} = 13.8$ MPa.

Commentary

In Table 7.9 below the results are summarized and as can be seen no cases of loads exceed the yield stress or 50 MPa for amplitude stress. Since Faiveley has had no problems with the screw joints in prior test rigs it was expected that no cases of load would cause failure. The highest von Misses stress and amplitude stress occur in the screw joints fastening the stiffening plates.

Table 7.9 Summary of stresses in screw joints.

Case of load	Size	Pretension [kN]	Load [kN]	von Mises stress [MPa]	Yield stress [MPa]	Amplitude stress [MPa]
Front plate (middle)	M20	102	35	542	640	4.6
Rear plate (middle)	M20	102	35	543	640	5.0
Stiffening plates	M12	35	34.5	564	640	13.8
Extension plates	M12	35	8.9	544	640	3.5

Regarding to Skruvförband [7] a coefficient η, whose value depends on where the load is absorbed, should be used to decrease the effect of the load F_l. But to increase the safety marginal $\eta = 1$ has been used throughout the calculations.

It can seem like the screw joints are too well dimensioned but there are above all two reasons why not smaller and/or fewer screws were chosen. First of all this is a test rig which is not supposed be optimized, but well dimensioned to handle rough handling for many years. Secondly, the screw joints are calculated upon the belief that they are perfectly tightened but in reality this is not the case, thus their strength might be weaker than calculated. Other reasons are that there is not much money to save by using screws of lesser quality and if the TBUs' braking force will increase in the future the test rig will still be capable to handle it.

8 Final design

Before drawings of all the parts of the new test rig could be handed over to the manufacturer approval from Faiveley had to be given. A meeting where the test rig's complete design and the results of the calculations were presented[10] was held on October 21st. As members of Faiveley have been active in all major decisions throughout the design and development most of the concepts were familiar to them. For that reason the rig was approved with only minor changes made to the design before manufacturing it. An example of a suggested change was the location of the extra holes (size M12) that are to be used for holding the extra air cylinders in place. These were moved a bit closer together to simplify the mounting.

The only parts that now were missing on the completely functional rig were the pneumatic devices that control and run the TBUs. This report has mainly focused on the development process of the rig itself but within this process consideration has been taken to the pneumatics. For example the target specifications specified how much free space there must be around the TBUs in order to easily make the pneumatic connections. In addition to this the space beneath the rig had always been reserved to house the pneumatic devices.

When the rig's design was complete and the drawings were sent to the manufacturer it was time to find these necessary pneumatic devices. Faiveley felt that re-using concepts that are well functioning today is the best way to go. Hence it was decided to use the same software and programs that control the braking sequences for the TBUs today, LabVIEW from National Instruments. The pneumatic valves were of 5/2 type from Festo. A control unit was also needed to give power to the valves at the right time and Festo suggested[11] a CPX control unit and a MPA valve terminal to make the entire device as small and compact as possible. All measurements done will be carried out separately and the signals will go straight from the load cells and pressure sensors to a PC.

To protect the electrical and pneumatic equipment a casing, made of thin aluminum sheets, was designed and fitted below the rig.

Figures 8.1 and 8.2 show the design of the test rig as a 3D model where all inbound parts are mounted to the rig, including the pneumatics and the casing. In the latter a detailed view of the extension and the load cell fixtures is presented. An assembly drawing is provided in Appendix D.

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[10] Meeting at Faiveley. Participating were M. Lindström, J.Stridh, F. Blennow, A. Arnell and T. Persson.
[11] Meeting with Festo to discuss pneumatic solutions. Participating were J. Stridh, M. Lindström, F. Blennow and P. Lindgren from Festo.

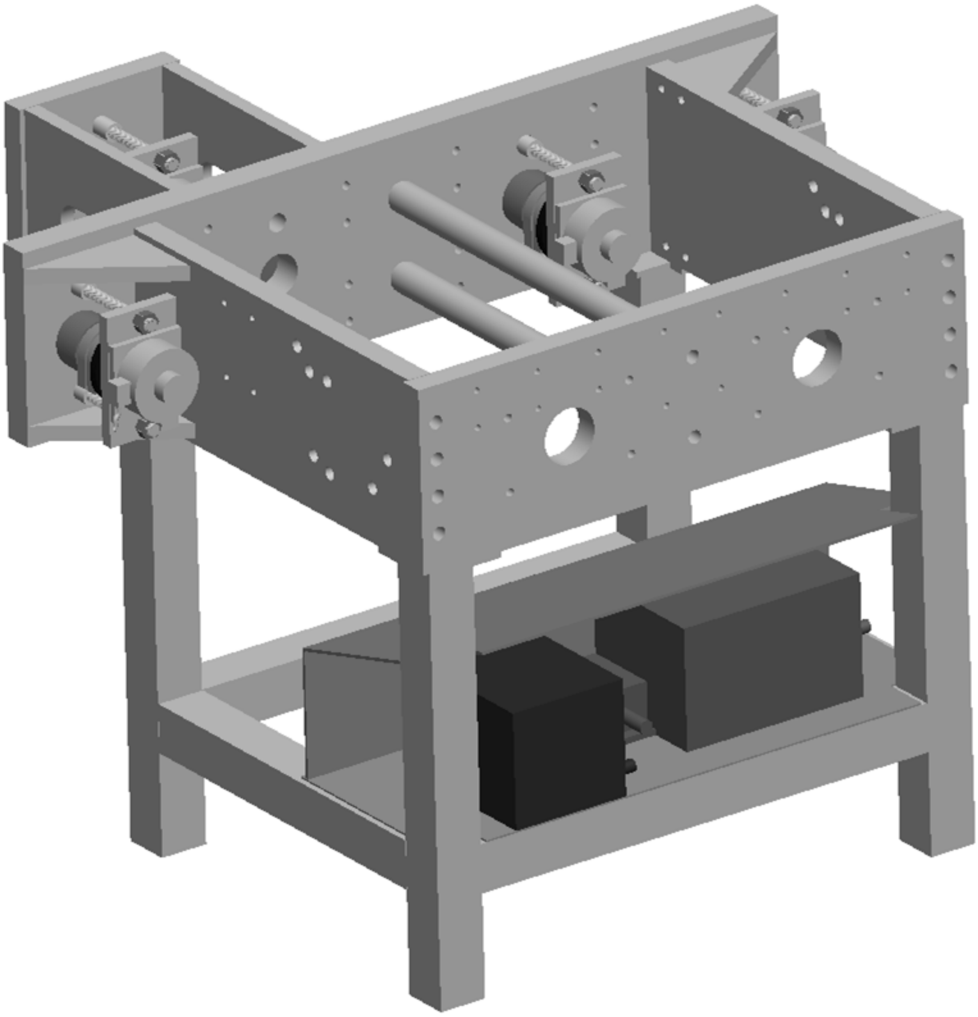


Figure 8.1 Complete rig with pneumatics and the casing.

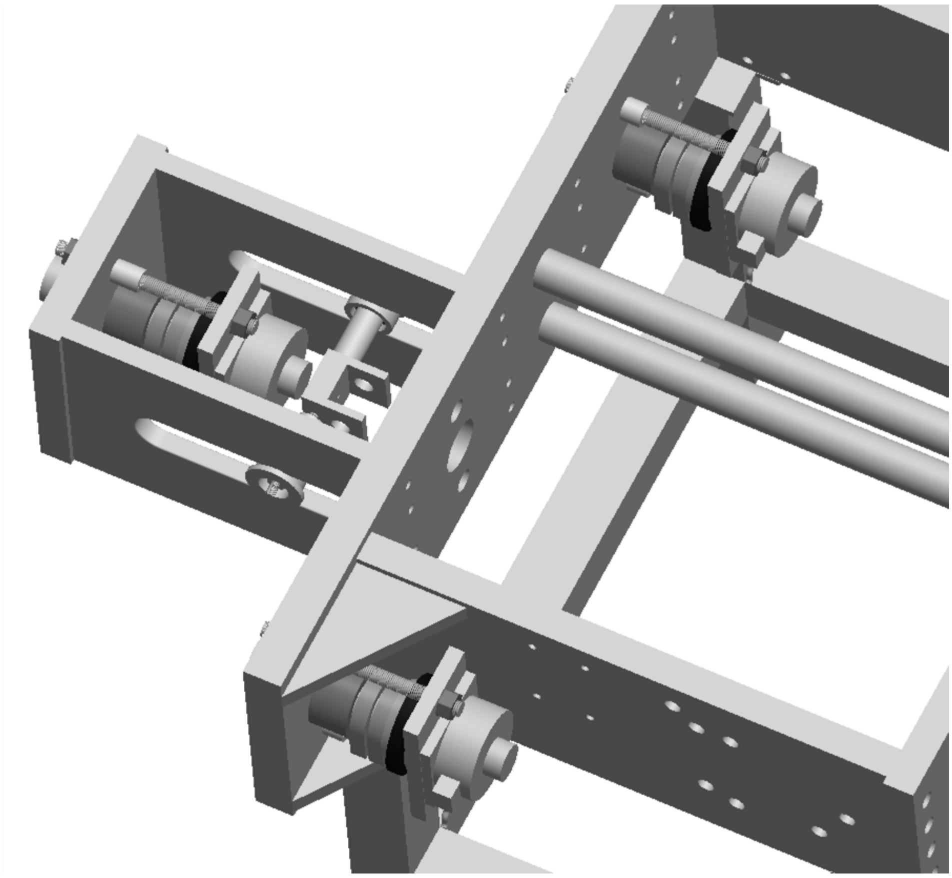


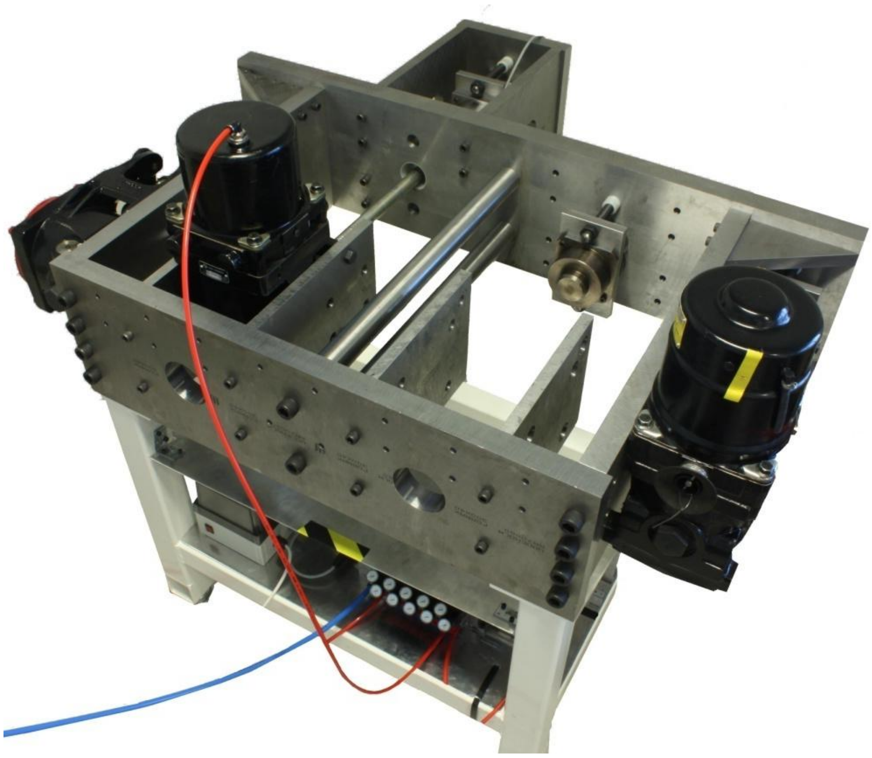
Figure 8.2 Detailed view of the rig.

9 Start up

The drawings for the test rig were handed over to the manufacturer, Ingenjörsfirma Jeppsson AB, in Ystad on the 23rd of October. According to the manufacturer all inbound components fit together as planned, no problems or incorrect measurements could be found. The complete test rig was delivered to Faiveley in Landskrona on December 2nd.

9.1 Concept testing

The next step was to evaluate the rig to see if it would meet the required demands, both structurally and in the sense of adaptability; can all TBUs be mounted to the new rig? Is it rigid enough? Picture 9.1 below shows the complete rig with the pneumatics and its casing mounted below the rear plate of the rig.



Picture 9.1 The test rig with TBUs.

9.1.1 General testing

This first step of testing was basic but still very time consuming. Each station was tested separately, starting with station number one (furthest to the left). The answers to the following questions were of essential importance;

- Was the mounting of the load cell fixture functioning as wished?
- Was the load cell fixture moving correctly under load?
- Were all holes for mounting correctly placed, i.e. was it possible to correctly mount all TBUs intended for this station?
- How long time and how much effort does it take to change a TBU or make adjustments of the load cell fixture?
- Was the TBU's centre line in line with the centre line of the load cell fixture?
- Were the fixtures and other equipment such as wheels correctly designed?
- Was it enough space for tools, fork lift and other mounting equipment?

Results

The outcome of the general tests was very positive but of course there were some remarks. First of all the rig has to be placed so you have sufficient space around it to get access with the fork lift when mounting TBUs.

The load cell fixture with all its components is quite heavy and cannot be handled in a perfectly ergonomically way. Changing TBU also requires some effort and it is preferred if two persons are doing it. Since the rig is designed for tests where changes are not done for several weeks these remarks were expected. The most important result from the general testing was the importance of mounting the TBUs correctly, i.e. all screws have to be tightened properly so the force is acting in the load cell fixture's centre line.

9.1.2 Hysteresis test

In a way of measuring the rig's stability, hysteresis test were performed on each of the four stations. A TBU that was calibrated and correctly functioning was mounted to a station. An actual load cell was mounted on the load cell fixture and it too was calibrated to produce accurate numbers. The air pressure to the TBU was set to 5 bar.

In a hysteresis test the air does not flow directly to the TBU, instead it flows via a throttle valve that slowly increases and decreases the air pressure in the TBU. This is done in order to obtain a braking cycle that is very slow, with the special valve it takes about a minute for the pressure to go from 0 to 5 bar and back to 0 again. In this time a sufficient number of readings from the load cell can be logged onto a computer. Both the air pressure and the braking force from the TBU is measured and logged.

These two units can then be plotted against each other in a diagram. This visualizes how the TBU performs in loading and unloading. Normally if the ratio between them is constant during loading it is evidence of a correctly functioning brake. In this case though, as the TBU is already properly functioning, a constant ratio will be evidence of a rigid test rig that does not flex too much when the brake is applied. During unloading the ratio is never constant as there is some inertia in the system, this gives the same output brake force even though the pressure is dropping. Diagram 9.1 shows the result of the hysteresis test, this particular one was performed on station one (furthest to the left).

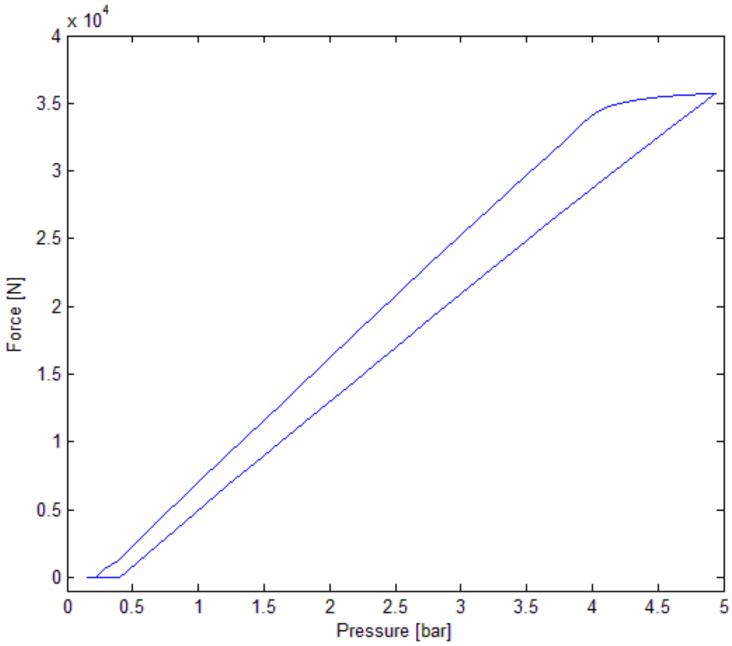


Diagram 9.1 Hysteresis test.

Results

All hysteresis tests gave positive results. The test rig did not flex when applied to high forces, but the ratio between the pressure and force stayed constant as long as the pressure increased. As said earlier it is really important to mount the brakes and the plate mounted on the brake shoe perpendicular to the load cell fixture to obtain correct measurements.

9.1.3 Structural test

Finally two tests of the rig's structural integrity were performed. One measured the deformation of the ends of the front plate. The other measured the stress in a point in the middle of the front plate with a strain gage. These were primarily made to find out how accurate and reliable the ANSYS analyses were.

The deformation was measured on four different locations on the back of the front plate, see Picture 9.2. It was measured with a depth gauge of type S229 from Sylvac [11]. Two TBUs, one with an output force of 50 kN and one with 43 kN, were mounted just as in the ANSYS analyses with the one with the highest force mounted on the end. One on the side station and one on the opposing middle station. The result is presented in Table 9.1.

New ANSYS analyses were made, the applied loads on the rig were set as in the real test. This time the directional deformation in the y-direction was calculated, see Figure 9.1.

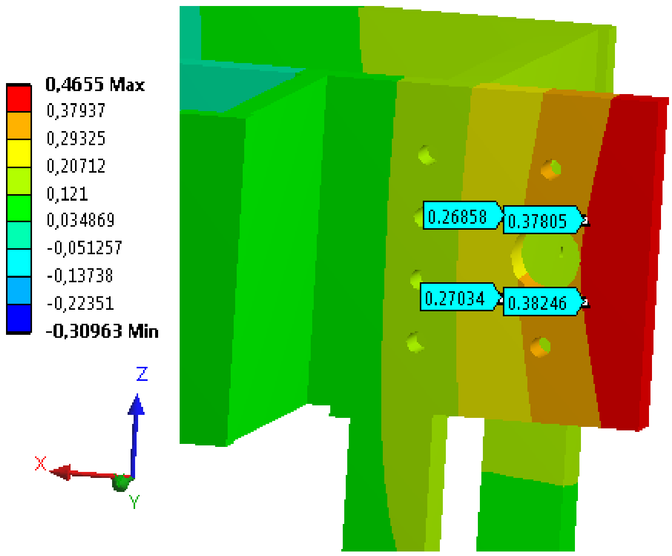
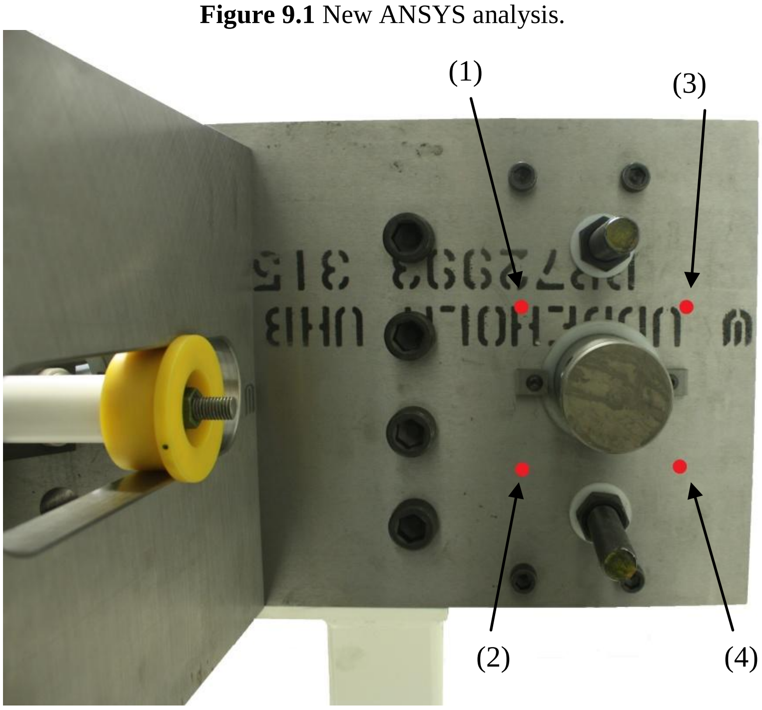


Figure 9.1 New ANSYS analysis.



Picture 9.2 Measurement points.

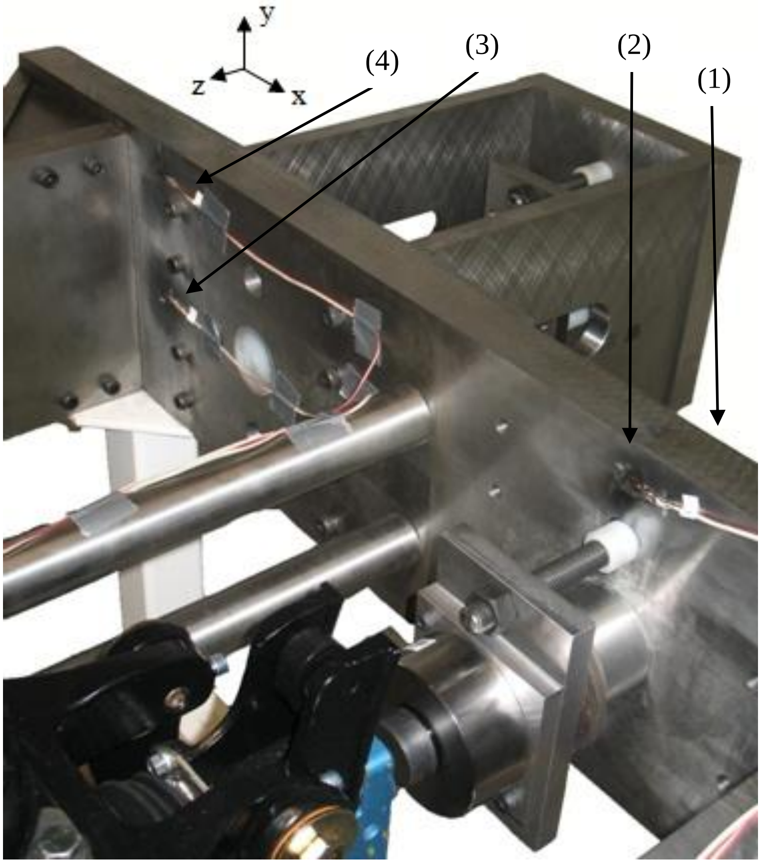
Results

All measurement points gave very accurate values comparing to the ANSYS values. This is of course very good, as it indicates that the FEM calculations in ANSYS can be trusted and furthermore that the test rig does not flex more than allowed. It should be noted that the brake unit in the middle station did not contribute much to the deformation.

Table 9.1 Result of deformation.

Measuring point	Real value [mm]	ANSYS value [mm]
1	0.262	0.269
2	0.272	0.270
3	0.388	0.378
4	0.405	0.382

In the stress measuring a strain gage of type 120LY13 from HBM was used. They were attached to the front plate in four places according to Picture 9.3. Gauge no. 1 cannot be seen as it is situated directly behind no. 2 but on the rear side of the front plate.



Picture 9.3 Strain gauge locations.

The front plate is considered very stiff and bends only a small amount when under load. The highest stresses due to bending will occur on the surfaces of the front – and rear sides of the front plate, in the x-direction. Because of this the strain gauges were attached in an orientation so that they measured in this direction. The strain was multiplied with Young's modulus (206 GPa) to acquire the value of the stress.

Figure 9.2 below presents the calculated values of the stresses with the new ANSYS analysis. The principal stresses had to be reviewed as these represent the tensile and compressive stresses.

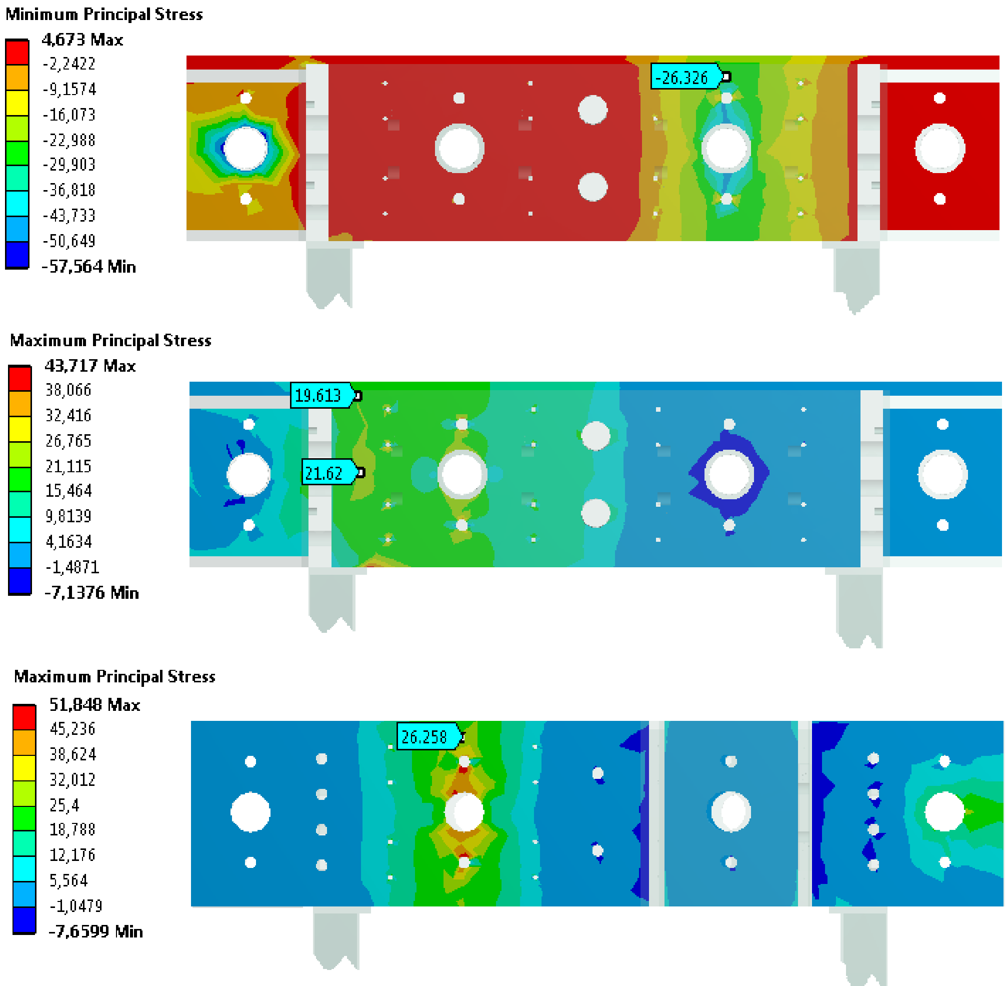


Figure 9.2 Principal stresses on the front plate.

Results

The signs of the each of the stresses were just as anticipated. Positions 1, 3 and 4 were positive which indicates a tensile stress whereas position 2 had a negative stress value which indicates a compressive stress. This proves that the assumption of how the beam would bend in section 7.1.3 was accurate.

Table 9.2 Measured stresses.

Measuring point	Real value [MPa]	ANSYS value [MPa]

1	29.0	26.3
2	-27.1	-26.3
3	25.6	21.6
4	24.9	19.6

Furthermore it can be noted that the measured stresses and the calculated ones are quite alike. One source of error is that the strain gauges were not attached 100 % straight in the x-direction. This test indicates that the ANSYS analysis was quite accurate and that the mesh and boundary conditions were correct.

9.2 Final specifications

As a concluding presentation, the exact specifications of the new test rig are presented. The final values were measured on the real rig or graded subjectively after having run endurance tests on the rig for a while. The acceptable and ideal values were presented in section 4.3.

Table 9.3 Final specifications.

Spec. No.	Specification	Unit	Final value
1	Maximum braking force of TBUs	N	70 000
2	Maximum weight of TBUs	kg	75
3	Cost	SEK	80 000
4	Elasticity	mm/kN	0 – 0.30
5	Size (W x D x H)	m^3	1.15 x 0.8 (1.15) x 1.0
6	Mass	kg	500
7	Stiffness	mm/kN	0.020
8	Clearance for mounting TBUs	mm	100
9	Clearance beneath the rig	mm	170
10	Service life	Years	-
11	Clearance for bulky TBUs	Subjective	Very good
12	Flexibility	Subjective	Very good
13	Changing time of TBU	Hours	0.5
14	Safety	Subjective	High
15	User friendly	Subjective	Medium
16	Number of various types of TBUs at each station	Amount	4+

Commentary

The total size of the rig surpassed even the ideal values, in this sense the rig had truly become very compact. The depth has two values, one with and the other without the extension plates. Despite the fact that the rig is compact it still has a lot of clearance around the TBUs. 100 mm from the fixture plates to the nearest obstruction makes the mounting easy compared to the old test rigs at Faiveley.

Some specifications are hard to measure, service life is one of these. According to the fatigue calculations the most exposed areas will withstand a theoretically unlimited life time. It is more probable that screws and threads might be worn which have to be replaced.

Specification no. 16 specifies that in average any given station can hold four out of the nine original models of TBUs. The plus sign indicates that in reality more models of Faiveley's product portfolio are mountable as long as they have the same mounting hole positions.

10 Conclusions

This has been a very informative project that has included the entire product development chain from early ideas to the actual manufacturing and testing of the product. Apart from this the project has also increased our knowledge in pneumatics as a considerate amount of time was spent on setting up and controlling the pneumatics. This had to be done before we could test the rig appropriately. Faiveley assisted with information concerning designing, calculations and such.

In this type of project, where many objectives and restrictions are set up early on, it is likely that some of them will change during the course of the project. This might lead to changes in the concepts and redesign, this is the iterative process that is typical in a product development project.

An example of this is the *Cost* specification. In the beginning of the project it was given a relatively low importance. As work carried on and the rig began to take shape, it became clear how many parts it would consist of. These were of various sizes and complexities and each redesign increased the estimated price. This lead to a wish from Faiveley's side for us to focus more on low cost solutions, this became apparent in the project when the *Cost* criterion was weighted the highest in the concept scoring.

During the designing of the last parts, we felt that we had become more efficient in finding good, cheaper solutions and sort out more expensive ones. Especially since already existing concepts were improved and implemented. We have realized that as engineers working with designing, the solutions will always be a tradeoff between optimized, efficient solutions and the ones that are the cheapest.

Because of the iterative process changes have been made continuously within all branches of the project but this report has not presented all these changes and redesigns. Also many of the designing sub problems were handled parallel and had work in progress at the same time. To get a clear and easy-to-follow theme in this report each sub problem is presented and solved within each section even though this did not entirely follow the real process.

As for the future there are a number of things that can be done to the rig be it improvements or new added parts. When it has been used for a while the lab technicians will get a good view of what they feel is good and what is lacking. For instance bins for storing nuts and bolts and possibly tools can be added to the rig in a smart place.

Further on holes can be drilled virtually anywhere without jeopardizing the rig's stability or function. The holes can be used to hold levers and such when using either the external air cylinders or some kind of measurement devices.

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Objectives and restrictions

- Maximum number of test stations is to be 5.
    - 2 must be able to handle brakes with parking brakes.
    - 1 station must be able to handle long brakes.
- 9 different types of brakes of which 2 are of particular importance (marked *Important*).
    - Drawing No FT 0080025
    - Drawing No FT 0080026
    - Drawing No FT 0080102
    - Drawing No 170582 (*Important*)
    - Drawing No 170758
    - Drawing No 270277
    - Drawing No 270581
    - Drawing No 270643 (*Important*)
    - Drawing No 270670
- Stiffness limit maximum 0.02 mm/kN.
- Elasticity of 0 – 0.3 mm/kN can be reached by demand.
- Maximum force applied to rig at each station is 70 kN (incl. fatigue).
- There must be room for at least 3 sensors (2 pressures and 1 load).
- 2 air cylinders for release of parking brakes must be included.
- Possibility for future brake models to be adapted to the rig is desirable.
- Cost is to be kept as low as possible.
- The rig does not need to have security certifications.
- A manual for using the rig will be made.

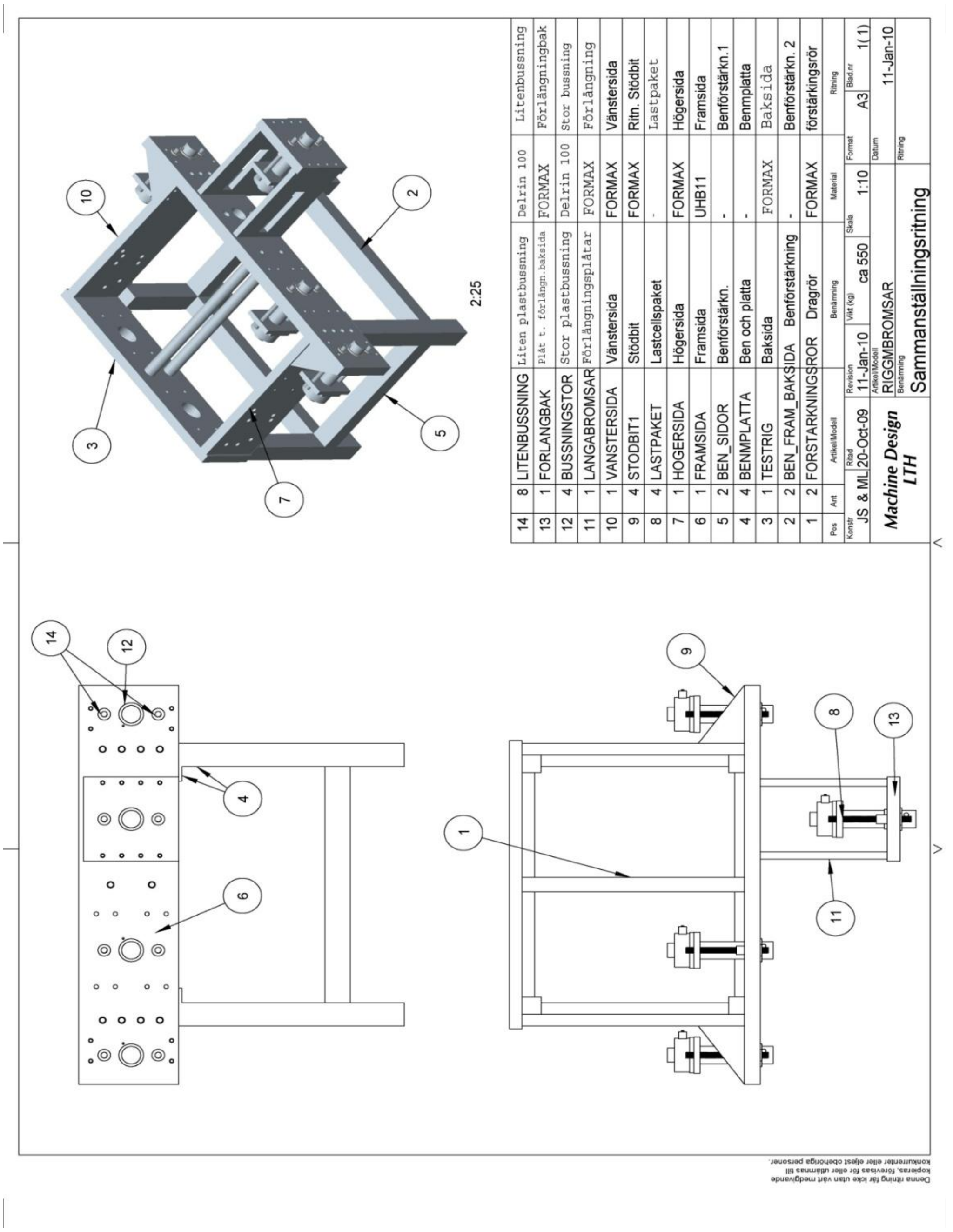
The project should be completed by the end of January 2010.

Drawing no.	Mounting	No. of holes	Weight (kg)	WxH (mm)	Force (kN) at given air pressure (kPa)	Parking brake (PB)	Length (max. length) (mm)	Obstructive parts
170758	right	3	62	436x373	34.5 at 380	yes	365 (465)	PB to the left.
270581	bottom/top	2	47	200x589	26.9 at 345	yes	600 (703)	Long rear.
270643	right/left	3	68	200x450	34.5 at 380	yes	338 (443)	Emergency release sprint extending to the rear.
270670	left	4	48	465x200	28.7 at 300	yes	362 (467)	PB to the right.
170582	right/left	3	62	200x467	25.8 at 380	yes	334 (439)	Emergency release sprint extending to either side.
270277	bottom/top	2	51	200x550	36.4 at 380	yes	890 (1054)	Large hand brake extending downward.
FT 0080025	bottom/top	2	27	200x326	50.0 at 500	no	600 (703)	Long rear.
FT 0080026	bottom/top	2	33	200x360	50.0 at 395	yes	600(703)	Long rear and hand brake extending to either side.
FT 0080102	right/left	3	38.6	210x256	50.0 at 380	no	360 (430)	-

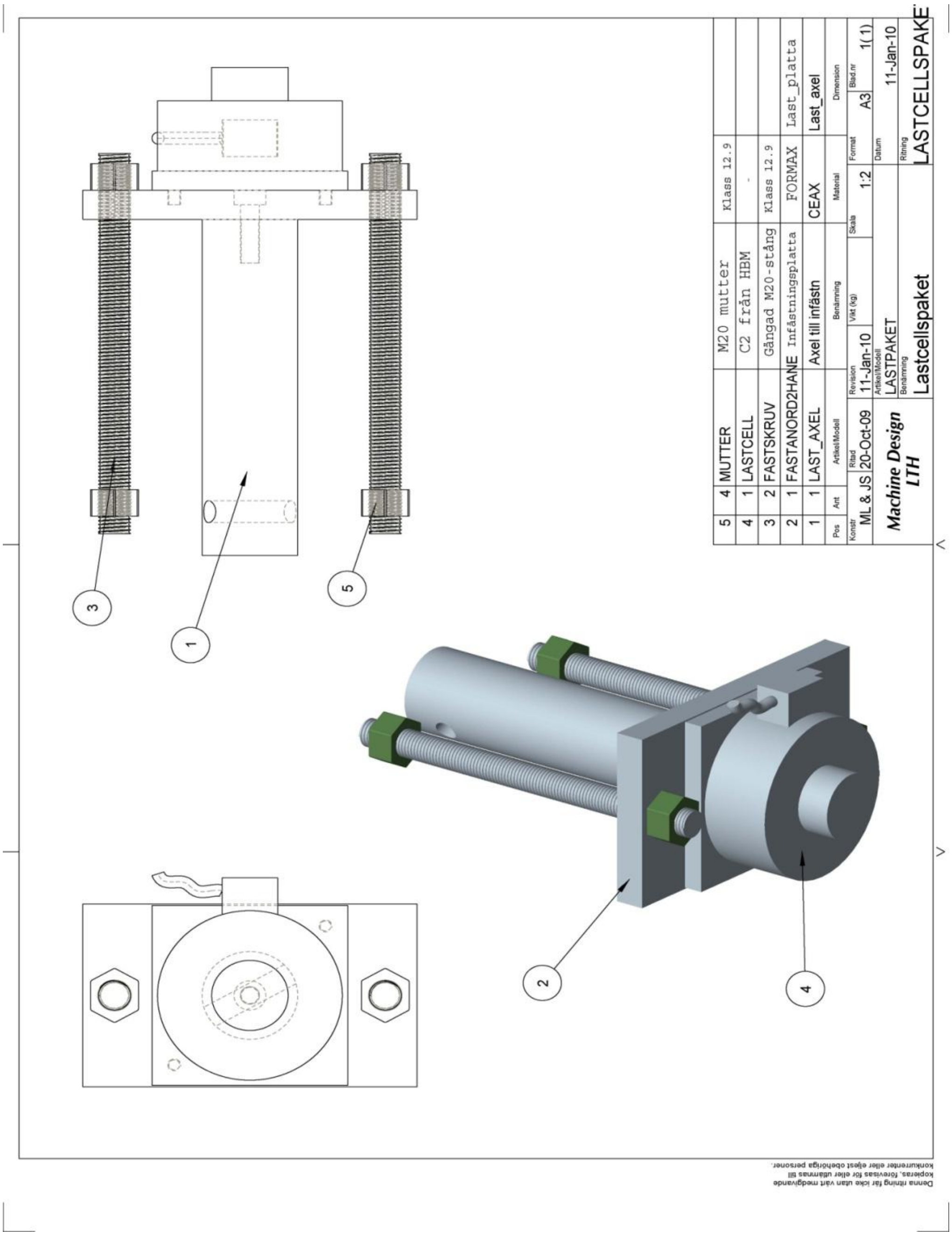
ANDREAS ARNELL	(5 most important)				
	1	**2**	**3**	**4**	**5**
The test rig is capable of running five simultaneous TBU tests.		X			
The test rig handles TBUs with various shapes, e.g. with parking brakes.				X	
The test rig will include one station with hydraulic connection.				X	
The test rig is capable of testing nine different predefined TBUs.					X
The test rig is rigid.			X		
The test rig provides an option to simulate elasticity.					X
The test rig is designed for fatigue.					X
The test rig allows space for test sensors.					X
The test rig provides space for additional air cylinders used for test of emergency release of parking brakes.				X	
The test rig is able to test future TBU models.			X		
The test rig is constructed in a cost efficient way.				X	
The test rig allows access for pneumatic connections.					X
The test rig allows access for a movable crane when loading/unloading TBUs.		X			
The test rig uses flexible fixtures that permit mounting of various TBUs.					X
The test rig has a set of different plates.		X			
The test rig allows an easy and ergonomic mounting of TBUs.			X		
The test rig provides storage for tools, bolts and such.	X				
The test rig provides sufficient space between TBU and the rig for tool access.				X	
The test rig allows an easy and quick changing of TBUs with various spindle lengths.		X			
The test rig is constructed of a material with good characteristics regarding weight and strength.		X			
The test rig allows access for a forklift truck when loading/unloading of TBUs.	X				
The test rig uses components that are rigid and resilient to wear.					X
The test rig provides a safe work environment.			X		
The test rig (without TBUs) is light enough to be lifted by the existing forklift truck.					X
The test rig is movable.					X
The test rig allows access to all stations.			X		
The test rig fits within and can be moved about in the company’s lab facility.					X
The test rig allows for a quick change of TBUs when loading/unloading.		X			

PER PERSSON	(5 most important)				
	1	**2**	**3**	**4**	**5**
The test rig is capable of running five simultaneous TBU tests.		X			
The test rig handles TBUs with various shapes, e.g. with parking brakes.			X		
The test rig will include one station with hydraulic connection.		X			
The test rig is capable of testing nine different predefined TBUs.				X	
The test rig is rigid.				X	
The test rig provides an option to simulate elasticity.					X
The test rig is designed for fatigue.					X
The test rig allows space for test sensors.		X			
The test rig provides space for additional air cylinders used for test of emergency release of parking brakes.				X	
The test rig is able to test future TBU models.			X		
The test rig is constructed in a cost efficient way.		X			
The test rig allows access for pneumatic connections.			X		
The test rig allows access for a movable crane when loading/unloading TBUs.					X
The test rig uses flexible fixtures that permit mounting of various TBUs.				X	
The test rig has a set of different plates.				X	
The test rig allows an easy and ergonomic mounting of TBUs.					X
The test rig provides storage for tools, bolts and such.		X			
The test rig provides sufficient space between TBU and the rig for tool access.			X		
The test rig allows an easy and quick changing of TBUs with various spindle lengths.				X	
The test rig is constructed of a material with good characteristics regarding weight and strength.			X		
The test rig allows access for a forklift truck when loading/unloading of TBUs.					X
The test rig uses components that are rigid and resilient to wear.				X	
The test rig provides a safe work environment.					X
The test rig (without TBUs) is light enough to be lifted by the existing forklift truck.				X	
The test rig is movable.			X		
The test rig allows access to all stations.				X	
The test rig fits within and can be moved about in the company's lab facility.			X		
The test rig allows for a quick change of TBUs when loading/unloading.				X	

J. STRIDH & M. LINDSTRÖM	(5 most important)				
	1	**2**	**3**	**4**	**5**
The test rig is capable of running five simultaneous TBU tests.			X		
The test rig handles TBUs with various shapes, e.g. with parking brakes.				X	
The test rig will include one station with hydraulic connection.			X		
The test rig is capable of testing nine different predefined TBUs.		X			
The test rig is rigid.					X
The test rig provides an option to simulate elasticity.					X
The test rig is designed for fatigue.					X
The test rig allows space for test sensors.				X	
The test rig provides space for additional air cylinders used for test of emergency release of parking brakes.			X		
The test rig is able to test future TBU models.		X			
The test rig is constructed in a cost efficient way.			X		
The test rig allows access for pneumatic connections.					X
The test rig allows access for a movable crane when loading/unloading TBUs.		X			
The test rig uses flexible fixtures that permit mounting of various TBUs.		X			
The test rig has a set of different plates.		X			
The test rig allows an easy and ergonomic mounting of TBUs.		X			
The test rig provides storage for tools, bolts and such.	X				
The test rig provides sufficient space between TBU and the rig for tool access.				X	
The test rig allows an easy and quick changing of TBUs with various spindle lengths.		X			
The test rig is constructed of a material with good characteristics regarding weight and strength.			X		
The test rig allows access for a forklift truck when loading/unloading of TBUs.		X			
The test rig uses components that are rigid and resilient to wear.				X	
The test rig provides a safe work environment.				X	
The test rig (without TBUs) is light enough to be lifted by the existing forklift truck.			X		
The test rig is movable.			X		
The test rig allows access to all stations.					X
The test rig fits within and can be moved about in the company's lab facility.				X	
The test rig allows for a quick change of TBUs when loading/unloading.	X				



Pos	Ant	Artikel/Modell	Benämning	Material	Ritning
14	8	LITENBUSSNING	Liten plastbussning	Delrin 100	Litenbussning
13	1	FORLANGBAK	Plåt t. förlängn.baksida	FORMAX	Förlängningbak
12	4	BUSSNINGSTOR	Stor plastbussning	Delrin 100	Stor bussning
11	1	LANGABROMSAR	Förlängningsplåtar	FORMAX	Förlängning
10	1	VANSTERSIDA	Vänstersida	FORMAX	Vänstersida
9	4	STODBIT1	Stödbit	FORMAX	Ritn. Stödbit
8	4	LASTPAKET	Lastcellspaket	-	Lastpaket
7	1	HOGERSIDA	Högersida	FORMAX	Högersida
6	1	FRAMSIDA	Framsida	UHB11	Framsida
5	2	BEN_SIDOR	Benförstärkn.	-	Benförstärkn.1
4	4	BENMPLATTA	Ben och platta	-	Benmplatta
3	1	TESTRIG	Baksida	FORMAX	Baksida
2	2	BEN_FRAM_BAKSIDA	Benförstärkning	-	Benförstärkn. 2
1	2	FORSTARKNINGSROR	Dragrör	FORMAX	förstärkingsrör


5 4 MUTTER M20 mutter Klass 12.9
4 1 LASTCELL C2 från HBM -
3 2 FASTSKRUV Gängad M20-stång Klass 12.9
2 1 FASTANORD2HANE Infästningsplatta FORMAX Last_platta
1 1 LAST_AXEL Axel till infästn CEAX Last_axel
Pos Ant Artikel/Modell Benämning Material Dimension
Konstr ML & JS Ritad 20-Oct-09 Revision 11-Jan-10 Vikt (kg) Skala 1:2 Format A3 Blad nr 1(1)
Artikel/Modell LASTPAKET Datum 11-Jan-10
Machine Design LTH
Benämning Lastcellspaket
Ritning LASTCELLSPAKET
1
2
3
4
5
Denna ritning får icke utan vårt medgivande kopieras, förevisas för eller utlämnas till konkurrenter eller eljest obehöriga personer.