Design and construction of the brake systems for the 'Formula Student'- Racing Cars BRC08 / BRC09

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Conclusion

The University of Applied Sciences in Berlin, HTW-Berlin, takes part at the 'Formula Student' with their own race cars since 2006. Historically derived from the 'Formula SAE', student teams are able to enter with self-built racing cars in an interdisciplinary competition. The regulations of the 'Formula Student' describes the conditions that a single-seater racing car must be build under within a year. For each competition year a new vehicle must be built.

For the 'Formula Student' – racing car BRC08 from the HTW-Berlin, the brake system was completely redesigned and supports the interpretation of the components consistently numerically. In the following vehicle BRC09 the brake system based on the BRC08 racing car was more developed, however most of the parts are identical to the previous vehicle.

In addition to optimization of selected brake pad friction materials, it shows the experimental verification of the realized brake system for the racing car BRC08, particularly in the design of the brake discs and their thermal load, yet the potential is not exhausted.

1 Introduction

The 'Formula Student' covers an interdisciplinary student contest with many teams around development, implementation and operation of a single-seater formula racing car that could theoretically serve as a prototype for a small series of 1,000 vehicles.

The participating student teams have the opportunity to consolidate their knowledge in the field of engineering sciences and to gain 'soft skills' in the same manner that for later professional life is becoming increasingly important.

The teams in the 'Formula Student' - projects are similar organized as company not only in technical tasks for construction of a Formula racing car, but also in relevant organizational and economic aspects, such as project management, financial planning and public relations involved.

1.1 Historical development of the 'Formula Student'

The idea of the 'Formula Student' was started 1981 in the United States of America by the SAE's (Society of Automotive Engineers) according to the "Formula SAE". The aim of the competition was then - and still is - a special student support in the field of engineering.

Since 1998, in cooperation with SAE and IMechE (Institution of Mechanical Engineers), the 'Formula Student' held in England. Meanwhile, the 'Formula Student' concept has found more followers in many countries. Now worldwide about 300 students cooperate at colleges and universities in interdisciplinary teams. Since 1999, the 'Formula Student', is also at home in Germany [1].

Under the term 'Formula Student Germany' the VDI (Association of German Engineers) this event is held annually on the Hockenheim Ring since 2006. The number of university teams participating has increased from 40 in 2006 to 78 in 2009, which documents the attractiveness of the 'Formula Student Germany' impressively.

1.2 Regulations

The regulations of the 'Formula student', based on the rules of the 'Formula SAE' [2], describes the key elements for realizing the single-seater formula racing car, which must be planned, designed and implemented by student work within a year. For the competition in the following year, a new vehicle must be produced.

Despite the extensive framework of the rules, the regulations give sufficient freedom, so that during the real implementation a creative design is possible. There is no prescribed minimum weight, the specific engine design of the required four-stroke engine can be freely chosen and it is possible to use modern materials.

However, for deviations from the regulations the teams must accept sensitive point deductions up to the disqualification and an exclution from the competition. The competition itself, as shown in Figure 1, is divided in static and dynamic disciplines (events), which are only presented in English.

The events are evaluated separately and sum up to the overall ranking of each team. The assessment of the events are carried through by engineers and professionals of racing and automotive industry. Not the fastest car wins automatically, but the best overall package of technology and economy.

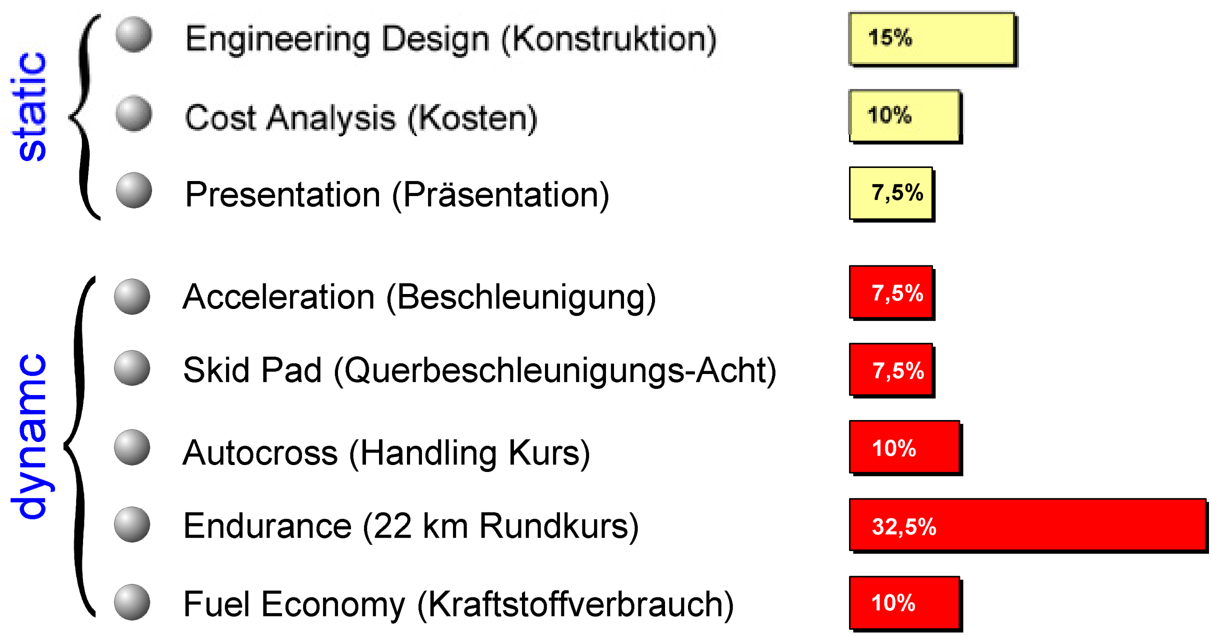

Figure 1: Disciplines

In static disciplines the costs will be analyzed and an imaginary formula budget plan for production of the car must be created. The 'engineers' of the team have to explain all structures and characteristics of the vehicle during the 'Design Report'. It is not the actual construction, but rather the professional background of the team and their understanding of the correlations that is evaluated.

The static disciplines are completed with the 'Business Plan Presentation'. A panel of industry experts review the final sales presentation of the formula car.

As the construction of cars is usually only possible with an intensive industry sponsorship, the placement of the teams is rather important in perspective of attractiveness to potential sponsors. Important basic characteristics of a 'Formula Student' - race car are shown in Figure 2.

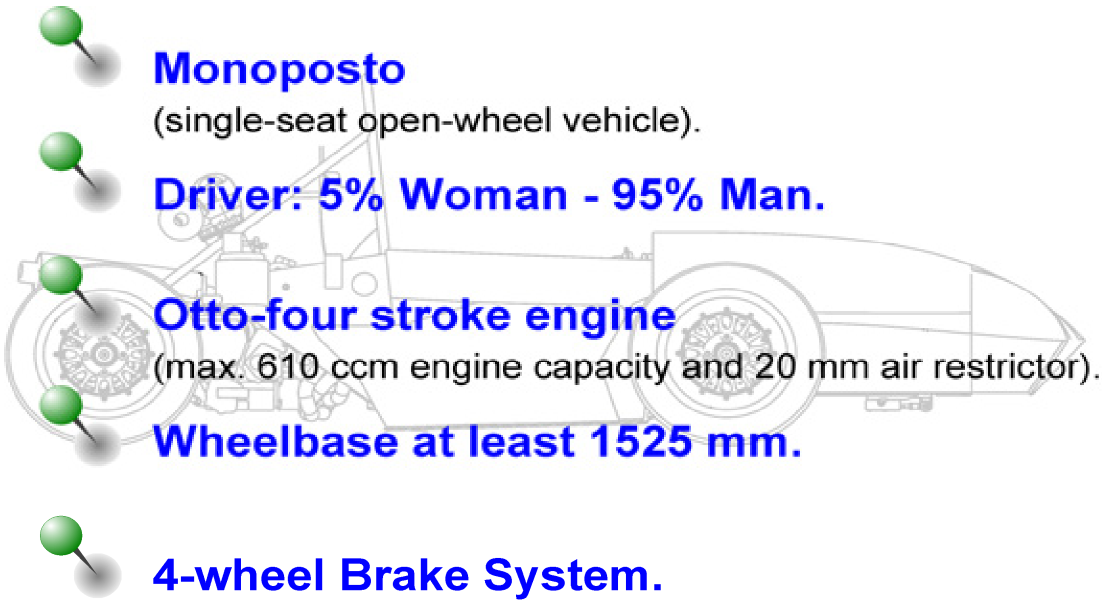

Figure 2: Important basic characteristics of a 'Formula Student' - racing car

2 Racing cars

Figure 3 shows the BRC08 Racing (Race Car Berlin, 2008) and BRC09 (Berlin Race Car 2009). The design of each builded vehicle was designed for the 'Formula Student' Germany (VDI competition of the 'Formula Student' on the track of the Hockenheimring) and corresponded to the rules of Formula SAE. Derived from the experiences of the years 2006 and 2007 following development priorities were set:

- Simple design, especially for the 'Formula Student' Germany developed.
- Empty mass of 200 kg.
- Expensive and complex auxiliary systems replaced by simple constructions.
- Component savings by reusing them.
- Consistent calculation and optimization of all mass relevant components with FE methods.

Figure 3: Racing cars BRC08 und BRC09

In the area of the drive and the suspension both vehicles have an identical concept. The frame of the BRC09 however, was developed from scratch.

The vehicle dynamics focus lies with the respective racing cars on a particularly good acceleration and braking ability, which is required in the dynamic competitive examinations. Table 1 provides information on important vehicle data.

Table 1: Vehicle data

The maximum speed of cars is due to the high lateral acceleration request during the dynamic tests not in the foreground, so that in comparison to the cars of previous years (v_{max}: 170 - 200 km / h) the max. speed is reduced by the locking the higher gears to 120 km / h. This has a positive effect of less traction interruptions in the acceleration test.

3 Construction and design of the braking system

The brake system in a formula car chassis design is next to the wheel suspension and the choice of the tire, the most important component. Especially with the winding course of the 'Formula Student' Germany the acceleration and stopping power for fast lap times has special importance. The more efficient the brake system decelerates the vehicle, the longer can be the range of high speeds. Ideally, the car will direct to the curve at full speed, delayed with maximum friction potential of the vehicle and in the top of the curve must start with the reacceleration.

To allow late braking good 'cold friction properties' of the pads are required because the brake pads have not reached their optimum operating temperature at the beginning of the braking process. A rapid increase of the braking deceleration to the maximum level is important in order to initiate the braking process late.

During the 'Endurance' (see also Figure 1, p.212), which represents a 22 km handling course, in addition to the 'cold friction properties' the thermal stability of the brake system is important. The braking system is stressed dynamically strong by rapid load change processes. An extreme thermal load on the brakes, leading to high temperature gradients and heat flows, will lead to thermal effects in the entire system. As a consequence of this the discs, the friction material, brake fluid and the peripheral components are stressed highly thermal wise. These following temperature effects are taken into account:

- Brake disc → Heat cracks
- Friction materials → Changes of friction or. loss of friction up to fading
- Brake fluid → Changes of compressibility → pedal travel extention
- Periphery → Fire

The high operating frequency during the ride also requires a ergonomic operation of the brake system, so that the human-machine interface gets an as high importance as the pure deceleration performance of the brake system.

The brake system of the racing car BRC08 was developed from scratch and as basis for future BRC-planned vehicles. The reusability of the brake system design was implemented by using the BRC09. The step towards modularisation of individual subsystems is a prerequisite for a drastic reduction in development times for new vehicles.

It further speaks for the modularization - especially of the brake system - that the dimensions of the brake components at an early stage of vehicle development must be fixed, because this has a major influence on the construction of the entire vehicle (wheel carrier, wheel hubs, frames).

The conceptual development of the braking device was characterized primarily by the need to save weight and improve the usability of the previous system. The numerical analysis of the heat budget could be well solved in the weight saving compromise regarding stability and performance.

3.1 Requirements by the 'Formula Student' rules

In the set of rules based on the 'Formula SAE Rules' [2], the specifications for the construction of the brake system been described. Where:

- The car must be equipped with a brake system that acts on all four wheels and is operated by a single control.
- There are two independent brake circuits required.
- For each brake circuit a separate reservoir must be present.
- In case of failure of a brake circuit two wheels must still be able to brake.
- A single brake acting on a limited-slip differential is acceptable.
- Brake-by-wire systems are prohibited.
- Unarmored plastic brake lines are prohibited.
- The brake systems must be protected with scatter shields from failure of the drive train or from minor collisions.
- The brake system will be dynamically tested and must demonstrate the capability of locking all four (4) wheels and stopping the vehicle in a straight line at the end of an acceleration run specified by the brake inspectors (50 km/h).
- A brake pedal over-travel switch must be installed on the car. This switch must be installed so that in the event of brake system failure such that the brake pedal over travels, the switch will be activated and will stop the engine from

running. This switch must kill the ignition and cut the power to any electrical fuel pumps. Repeated actuation of the switch must not restore power to these components, and it must be designed so that the driver cannot reset it. The switch must be implemented with analog components, and not through recourse to programmable logic controllers, engine control units, or similar functioning digital controllers.

- The car must be equipped with a red brake light of at least 15 watts, or equivalent, clearly visible from the rear. If an LED brake light is used, it must be clearly visible in very bright sunlight.
- This light must be fixed between the wheel centerline and below the driver's shoulder level vertically and approximately on vehicle centerline laterally.

3.2 Brake force distribution

For the brake system the brake force distribution and brake force distribution graph plays an important role. In contrast to required brake force distribution, in which the lock sequence of front and rear axle is fixed, for vehicles for public transport this does not count for the 'Formula Student' racing cars. The braking force during braking force distribution diagram is therefore not restricted. However, the braking force distribution diagram for the usual stationary conditions are assumed:

- Movement on a plane surface (no ascent influence).
- Neglect of the chassis suspension (no movement along the z-direction.
- Constant tire rolling radius (no tires suspension).
- Neglect of the wheel resistance coefficient.
- Neglect of the wheels moment of inertia.
- No effect of aerodynamic resistance forces.
- No lift- and downforces.
- Neglect of the engines brake torque.

Figure 4 shows the brake system parameters required for the geometry and strength of the vehicle.

Figure 4: Center of gravity

For the description of the ideal braking force distribution to the front ($F_{B,v,ideal}$) and the rear axle ($F_{B,h,ideal}$) is for a vehicle with the mass m_{Fzg} and gravity h_S, l_v, l_h if taking into account the dynamic weight distribution on the axles ($F_{G,\ v,\ dyn}$, $F_{G,\ h\ dyn}$), the wheelbase l and the acceleration of gravity g in the steady deceleration $\ddot{x}_B$ [3]:

$$F_{B,v,ideal} = F_{G,v,dyn} \cdot \frac{\ddot{x}_B}{g} = F_{G,v,stat} \cdot \frac{\ddot{x}_B}{g} + m_{Fzg} \cdot \ddot{x}_B \cdot \frac{h_S}{l} \cdot \frac{\ddot{x}_B}{g} = m_{Fzg} \cdot g \cdot \frac{l_h}{l} \cdot \frac{\ddot{x}_B}{g} + \frac{m_{Fzg} \cdot \ddot{x}_B^2 \cdot h_S}{l \cdot g} \tag{Eq. 1}$$

$$F_{B,h,ideal} = F_{G,h,dyn} \cdot \frac{\ddot{x}_B}{g} = F_{G,h,stat} \cdot \frac{\ddot{x}_B}{g} - m_{Fzg} \cdot \ddot{x}_B \cdot \frac{h_S}{l} \cdot \frac{\ddot{x}_B}{g} = m_{Fzg} \cdot g \cdot \frac{l_v}{l} \cdot \frac{\ddot{x}_B}{g} - \frac{m_{Fzg} \cdot \ddot{x}_B^2 \cdot h_S}{l \cdot g} \tag{Eq. 2}$$

Taking into account the related gravity height to the wheelbase X,

$$X = \frac{h_S}{l} \tag{Eq. 3}$$

the part of the brake force at the rear axle Ψ

$$\Psi = \frac{l_v}{l} \tag{Eq. 4}$$

and the deceleration z

$$z = \frac{\ddot{x}_B}{g} \quad \text{(Eq. 5)}$$

follows the ideal relationship between the braking force based on the weight of all vehicle (see also [4]).

$$\frac{F_{B,v,ideal}}{F_G} = f(z) = z \cdot \left[(1 - \Psi) + z \cdot X\right] \quad \text{(Eq. 6)}$$

$$\frac{F_{B,h,ideal}}{F_G} = f(z) = z \cdot [\Psi + z \cdot X] \quad \text{(Eq. 7)}$$

For the ideal braking force distribution followed by elimination of the parameter z:

$$\frac{F_{B,h,ideal}}{F_G} = \sqrt{\frac{(1-\Psi)^2}{4 \cdot X^2} + \frac{F_{B,v,ideal}}{X \cdot F_G}} - \frac{1-\Psi}{2 \cdot X} - \frac{F_{B,v,ideal}}{F_G} \quad \text{(Eq. 8)}$$

Based on (Eq.8) Figure 5 shows the ideal braking force distribution of the BRC08's braking force distribution diagram. To obtain the best possible deceleration the constructive normal distribution needs to be orientated on the maximum of tire adhesion.

While for other Formula cars front and rear wings provide some considerable aerodynamic down force, in the Formula Student the maximum speeds for effective use of aerodynamic down force is too low. Therefore, the view can be limited in the standard construction to the range to about 1.0 g. For a wheel friction effort of 1.0 the front and rear axle will lock simultaneously as shown for the standard construction in Figure 5.

The BRC race cars do not have other brake force reducer and brake force distribution units, so the real distribution is always represented by a straight line.

Under different wheel friction coefficient from the standard brake force distribution the distance between the real and ideal brake force distribution is however quite large. As a result an extended stopping distance following the over braked front axle of the vehicle with an understeer self steering behaviour. In particular, the realizable maximum friction coefficient on the course can vary considerably depending on tire choice and road surface, so the braking force distribution must always be adapted to the actual road-friction situation before a competition.

For this purpose in racing cars two brake master cylinders, linked with a manually adjustable balance beam, are usual (see Chapter 4.1).

This results in a adjustable field to adapt the available adhesion of the tires to the ideal braking force distribution.

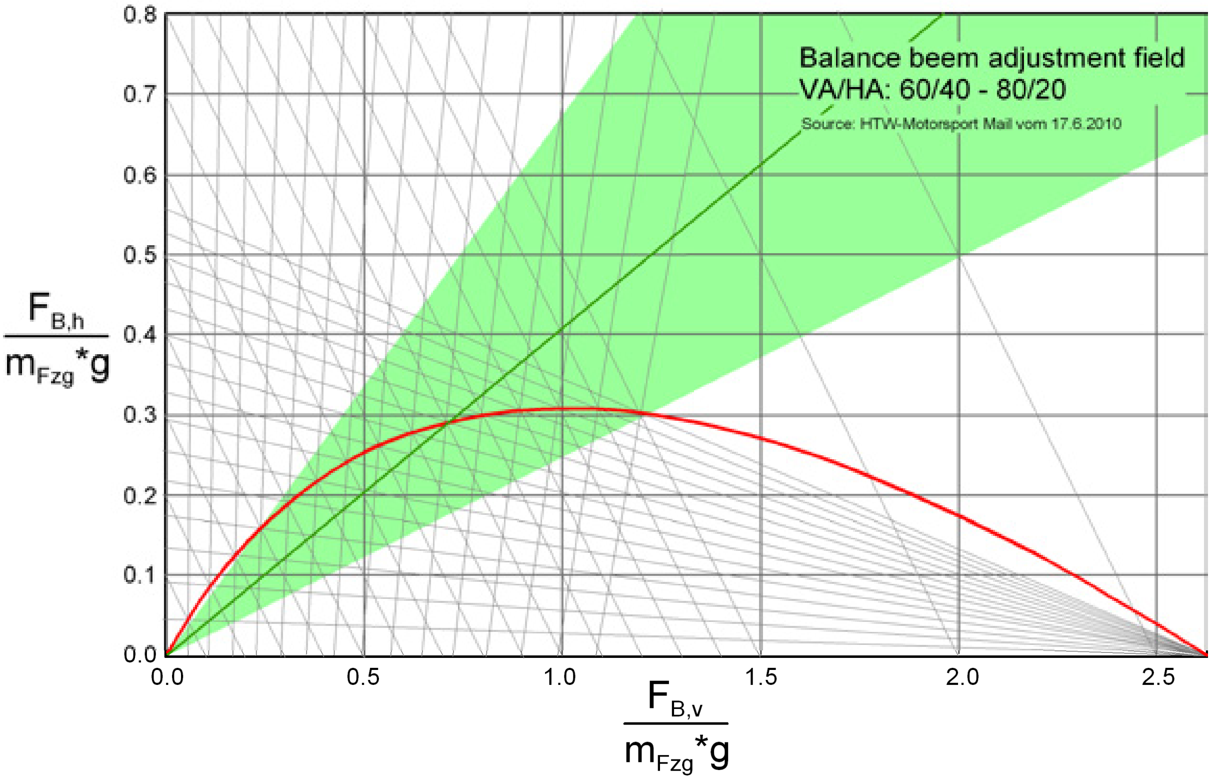

Figure 5: Brake force distribution diagram of the BRC08 with adjustment of the real brake force distribution

In a variation of the tire adhesion used for planning and occurring during the ride both axles do not lock at the same time in a deceleration with the maximum adhesion utilization, but there is a defined lock sequence. Without the use of ABS in principle it is necessary to lock the front axle before locking the rear axle to ensure the car's stability. The traction effort of the front axle must be greater than that of the rear axle.

When setting the actual brake force distribution, therefore, the traction effort of the axles has to be examined.

The brake stability can be assessed using the μ-z diagram. In this diagram, the ideal braking force distribution runs in a 45 ° angle. To evaluate the braking stability, the curves of the tire-adhesion limits are drawn in additional.

Regarding the rear axle's brake force portion the adhesion limits takes:

$$\Phi = \frac{F_{B,h}}{F_{B,h} + F_{B,v}} = \frac{F_{B,h}}{z \cdot F_G} = \frac{F_{B,h}}{z \cdot m \cdot g} \qquad \text{(Eq. 9)}$$

<u>Front axle:</u>

$$z_v = \frac{\mu \cdot (1 - \Psi)}{1 - \mu \cdot X - \Phi} \text{ or equivalent:} \qquad \text{(Eq. 10)}$$

$$\mu_v = \frac{F_{B,v}}{F_{G,v,dyn}} = \frac{1 - \Phi}{\frac{1 - \Psi}{z} + X} = \frac{F_{B,v}}{m \cdot g \left(\frac{l_h}{l} + z \cdot \frac{h_S}{l} \right)} \qquad \text{(Eq. 11)}$$

<u>Rear axle</u>:

$$z_h = \frac{\mu \cdot \Psi}{\mu \cdot X + \Phi} \text{ or equivalent:} \qquad \text{(Eq. 12)}$$

$$\mu_h = \frac{F_{B,h}}{F_{G,h,dyn}} = \frac{\Phi}{\frac{\Psi}{z} - X} = \frac{F_{B,h}}{m \cdot g \left(\frac{l_v}{l} - z \cdot \frac{h_S}{l} \right)} \qquad \text{(Eq. 13)}$$

Figure 6 shows an example of the adhesion potential of the vehicle BRC08 for the adhesion of 1.0 adjusted linear brake force distribution.

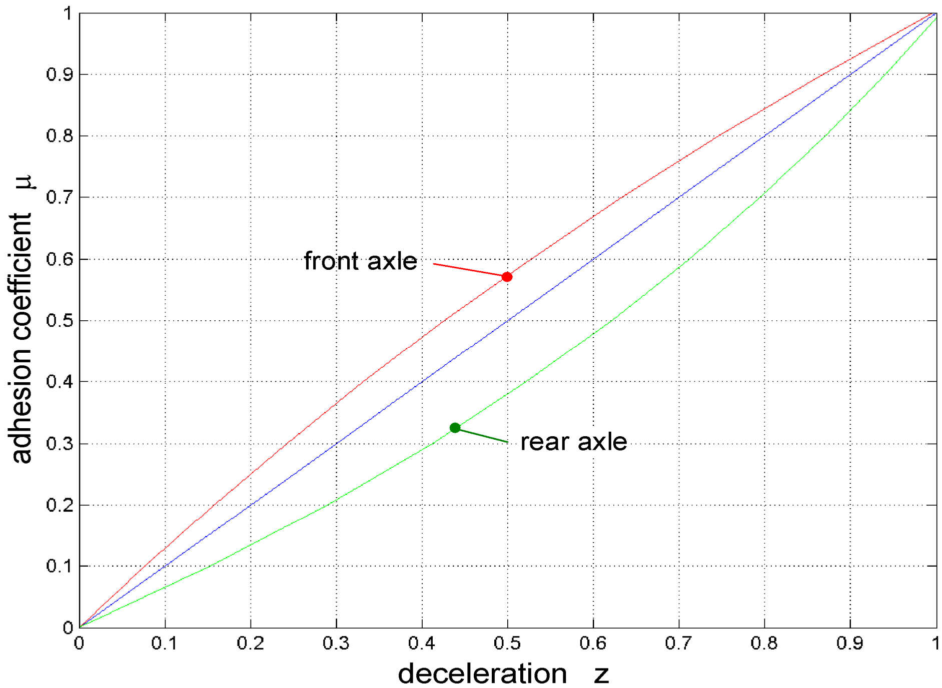

Figure 6: Adhesion potential to represent the braking stability (μ-z diagram)

3.3 Brake circuit distribution

In the 'Formula Student' rules two independent brake circuits are asked. In selecting the method of the apportionment fail safety and vehicle dynamics aspects has to be consider.

Failure of one or both brake circuits has two main reasons:

- Leakage
- thermal overload

The probability of failure of the brake system through leakage depend strongly on the system design. The failure probability rises with the increase of components and connectors. Permanently installed and fixed parts and static seals provide high security against default. The use of hose lines increases the risk of default, which is why the regulations do not allow unarmoured plastic pipes. For this purpose metal mesh sheathed cables (braided hose) are used. A higher probability of default, however, have moved seals, as used for example in the master brake cylinder and the brake calipers (also see [5]). An equally important aspect like the mechanical failure is the protection against a failure by thermal overload. By very frequent braking before each curve on the Endurance course the brakes are highly stressed. A failure, both by permanent damage to the brake lining or brake disc, and temporarily by the failure of the brake fluid must be considered in the selection of components.

The failure of the brake fluid can be felt as vapor lock, or by an improper extension of the brake pedal travel as a result of increased compressibility of the liquid. If there is a brake circuit failure, it must achieve the highest possible track stability are required in order to not overwhelm the driver. Because of using fixed caliper brakes on the front axles of the BRC08 and the BRC09 a slightly more positive scrub radius is set. This does not allow a diagonal brake circuit distribution, since the higher brake force at the front axle results in a brake-yaw moment that can not be compensated by an steer-yaw moment which will be initiated by the kingpin. The brake circuit distribution must therefore be designed as front axle / rear axle (black / white layout), which will also support the method of the adjustable brake force distribution using a balance beam (see also Section 4.1).

3.4 Thermal design

In view of a high thermal reliability while simultaneously reducing the masses a numerical calculation of the heat budget of the device is useful. During the braking process the disc has to save about 80-90% of the total braking energy as heat. Only about 10-20% of thermal energy can be delivered as forced or free convection or radiation into the environment. Therefore the thermal design of a brake system is made essentially on the dimensions of the brake disk.

The total heat flux on the disc is composed of the combined heat flux supplied by the wheel brake ($\dot{Q}_B$) and the proportions of heat radiation ($\dot{Q}_{St}$), free and forced convection ($\dot{Q}_K$) and to other components through directed heat flow ($\dot{Q}_L$).

Taking into account the portion of brake force at the rear axle by (Eq. 9), p. 221, for the actual speed v follow the implemented power at the front and rear wheel brakes:

$$P_{B,Scheibe,h} = \frac{1}{2} \cdot \boldsymbol{\Phi} \cdot m \cdot g \cdot z \cdot v = \dot{Q}_{B,Scheibe,h} = f(t) \tag{Eq. 14}$$

$$P_{B,Scheibe,v} = \frac{1}{2} \cdot (1 - \boldsymbol{\Phi}) \cdot m \cdot g \cdot z \cdot v = \dot{Q}_{B,Scheibe,v} = f(t) \tag{Eq. 15}$$

For the special case of the brake disc, the spatial dimension in the thermal calculation are reduced, Figure 7, in that only a cross section of the friction ring is considered.

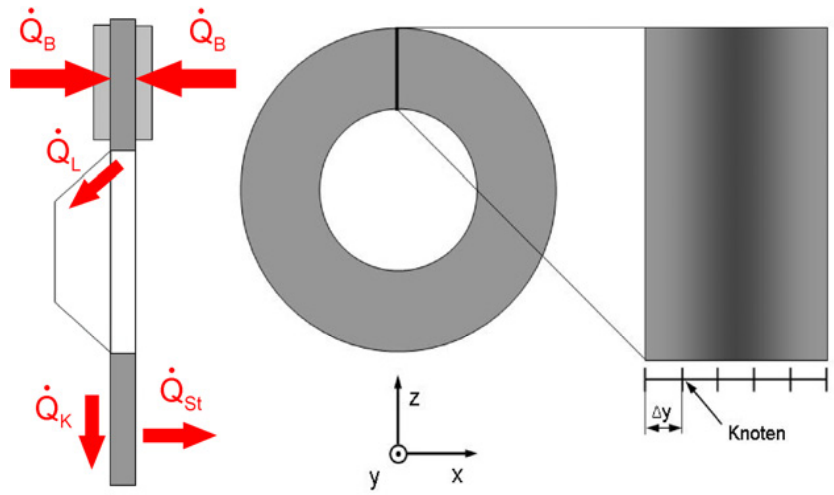

Figure 7: Brake disc model

In the model for calculating the brake disc temperature as a function of the dimensioning and choice of material, the heat current quantities in the brake pads and rotors in the brake disc pot will be summarize and build together with the convection and radiation heat flux a common percentage heat flow runoff.

4 Dimensioning

In a formula car a brake booster is usually not used, so that in addition to the deceleration behavior the clamping energy of the system is of great importance. Only the energy (W_{bet}) provided by the driver is vaillable. It is required for:

- Clamping energy (W_{Sp}) (expansion of the brake calipers)
- Strain energy ($W_{üb}$) of the transmission device (e.g. hose lines)

$$W_{Bet} = \int F_{Bet} ds_{Bet} = \int p_{Üb} dV_{Üb} + \int F_{Sp} ds_{Sp} \qquad \text{(Eq. 16)}$$

The actuation energy must be minimized because on the one hand the pedal travel is very limited through the existing space and on the other hand the driver is loaded by a high level of actuation force.

Consequences are reflected in the use of rigid tubes and hose lines, through the use of stiffer brake calipers, brake fluid with the use of low compressibility, maximizing friction radius and in a low system pressure level (30-50 bar). In parallel, the actuation of the brake pedal also has to be designed in terms of ergonomics.

4.1 Actuation device

The design of the actuation device of a formula car differs notably from the usual designs of passenger cars in the linkage of the brake pedal to the brake master cylinder. The actuation device must have an adjustable brake pedal ratio at one hand and an adjustable static brake force distribution at the other hand.

A small pedal travel and a stiff brake system leads to a secure feeling at the brake pedal. By the lying seating position of the driver, it is difficult to dose the braking force during emergency braking.

4.1.1 Calculation

The brake pedal ratio can be changed with help of the lever geometry or an adjustable angle between the brake pedal force (F_P) and the actuation force directed to the master cylinder ($F_{HZ,Bet}$). Figure 8 shows the forces at the actuation device.

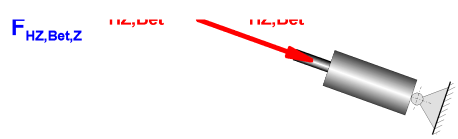

Figure 8: Forces at the actuation device

A flexible modification of transmission on the spot, however, can only reached with a complicated change of the brake lever's geometry, because the brake pedal bearing can be exposed to high loads.

Thus, the shown redirection of the actuation force - as in the vehicle realized BRC08 - leads to a high bearing load at the brake pedal through the force $\vec{F}_{P,Lager}$.

$$\vec{F}_{P,Lager} = \vec{F}_P + \vec{F}_{HZ,Bet} \tag{Eq. 17}$$

This was considered at the racing car BRC09 and the bearing of the pedal was positioned under the master brake cylinder(see Figure 13, p.230).

Better suited than the change of the brake pedal geometry for an easy adjustment of the pedal-transmission is a flexible change of the tilt angle α for the master brake cylinder. For the main brake cylinder directional control force is valid:

$$F_{HZ,Bet} = \frac{F_P \cdot L_1}{\cos(\alpha) \cdot L_2} \tag{Eq. 18}$$

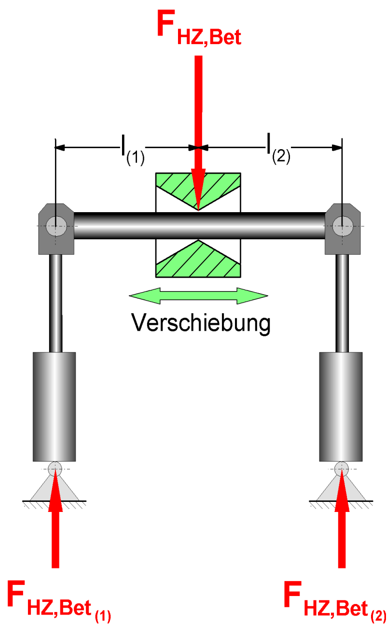

Figure 9: Balance beam

To adjust the static brake force distribution to the different adhesion conditions on the course the brake pedal force ($F_{HZ,Bet}$) will be divided on two main brake cylinders.

With the choosen brake circuit distribution at the vehicles BRC08 and BRC09 (see Cap.) the front and rear axle each operated with a single-circuit master brake cylinder. The distribution of the push rod forces assumes a balance beam with an adjustable position to conduct the actuation force ($F_{HZ,Bet}$), Figure 9, directed to the brake master cylinder.

For the forces to the brake master cylinders with $l = l_{(1)} + l_{(2)}$ follows from the equilibrium conditions on the balance beam:

$$F_{HZ,Bet_{(1)}} = \frac{F_{HZ,Bet} \cdot l_{(2)}}{l} \tag{Eq. 19}$$

$$F_{HZ,Bet_{(2)}} = \frac{F_{HZ,Bet} \cdot l_{(1)}}{l} \tag{Eq. 20}$$

Obtain together with Eq.18:

$$F_{HZ,Bet_{(1)}} = \frac{F_P \cdot L_1 \cdot l_{(2)}}{\cos(\alpha) \cdot L_2 \cdot l} \tag{Eq. 21}$$

$$F_{HZ,Bet_{(2)}} = \frac{F_P \cdot L_1 \cdot l_{(1)}}{\cos(\alpha) \cdot L_2 \cdot l} \tag{Eq. 22}$$

Because of the choice to a front / rear axle distribution an easy-to-implement basic brake pressure distribution can be made by means of a separate dimensioning of the main brake cylinder piston diameter ($d_{HZ(x)}$). In summary it is valid for the relationship between the brake pedal and the brake circuit pressure:

$$p_{(1)} = \frac{l_{(2)}}{d^2_{HZ_{(1)}}} \cdot \frac{4 \cdot F_P \cdot L_1}{\pi \cdot \cos(\alpha) \cdot L_2 \cdot l} \tag{Eq. 23}$$

$$p_{(2)} = \frac{l_{(1)}}{d^2_{HZ_{(2)}}} \cdot \frac{4 \cdot F_P \cdot L_1}{\pi \cdot \cos(\alpha) \cdot L_2 \cdot l} \tag{Eq. 24}$$

4.1.2 Components

4.1.2.1 Pedal box

The design of the pedal box, consisting of brake and accelerator pedal, has to be subject to the following conditions:

- compact and lightweight design
- space optimized
- stiff pedals
- rigid mounting
- adjustable in x-direction

– ergonomically optimized
– integration into the frame structure

For the driver to be able to take a fatigue free riding position, it is of advantage that the pedal box is movable in the X direction and the angle of the pedals can be adjusted. With the use of driver specific seat inserts an adjustment of the pedal box in X direction is not necessary. For a fatigue free position, it is also useful not to position his feet vertical on the pedals, but slightly inclined in the direction of travel. This can be done with the pitching of the pedals at about 5-10° from the Z-axis or by the offset of the tread of the pedal to the posting of the heel by about 10-30 mm. With an increasing angle, however, the usable actuation displacement is limited. In the sitting position of the BRC08/09-vehicles a maximum operating angle of about 35° is possible to not to overstretch the foot.

Figure 10 shows a sketch of the pedal box of the BRC08 consisting of brake and accelerator pedal. The balance beam to the brake master cylinder is located under the vehicle floor at the BRC08. A clutch pedal is not available because the clutch is electrically operated via the steering wheel. The real version in the BRC08 is shown in Figure 11.

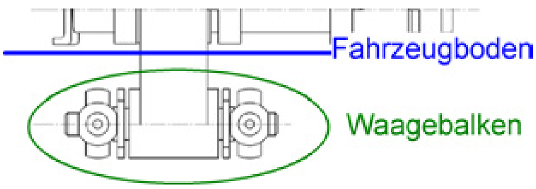

Figure 10: BRC08 pedal box sketch [6]

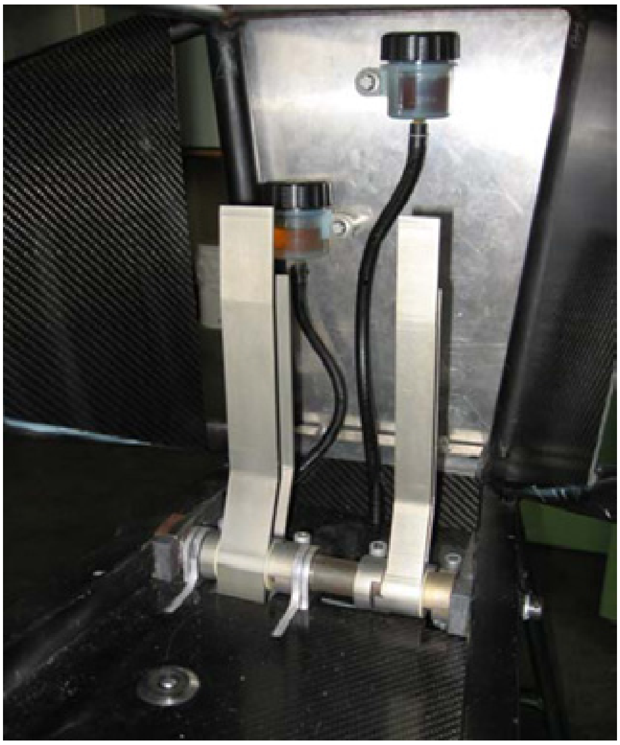

Figure 11: BRC08 pedal box [6]

The brake pedal has an adjustable stop in the released position, which ensures the safe opening of the compensation bore (sniffing hole) in the master cylinders, Figure 12. The distance between brake and accelerator pedal must be of sufficient size to safely prevent a contact of the shoes when operating.

As part of the development of the brake system from the BRC08 to the BRC09 the pedal box was particularly affected by changes. The pedals of the BRC09 do without a force redirection. The brake master cylinder therefore have been placed in the interior, however, the brake pedal is less burdened. Figure 13 shoes the pedal box of the BRC09.

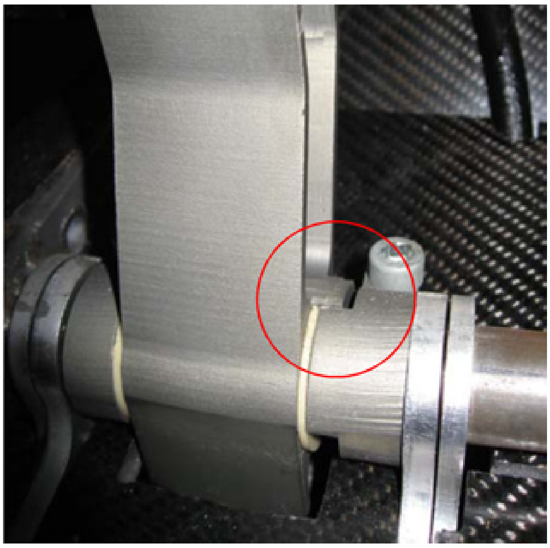

Figure 12: Brake pedal stop [6]

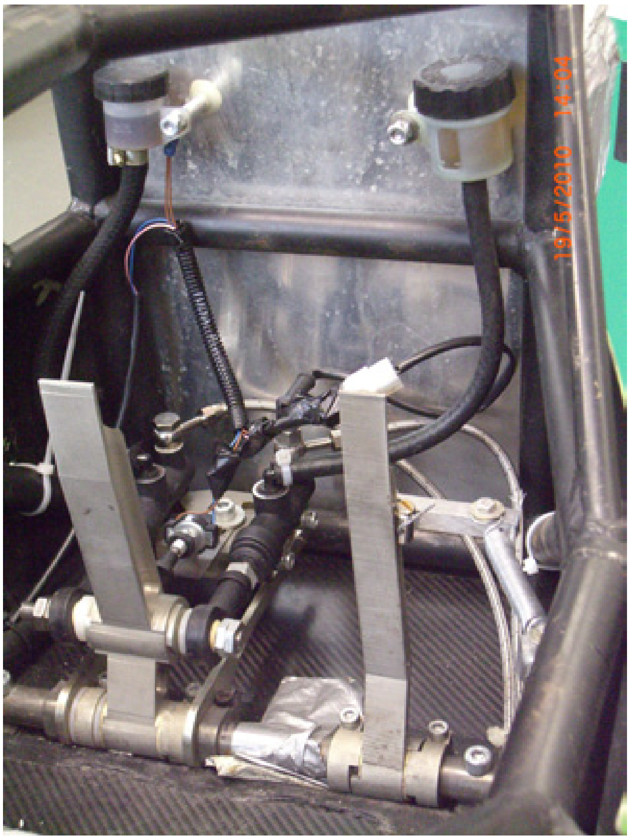

Figure 13: BRC09 pedal box

4.1.2.2 Balance beam

The structure of the balance beam is designed for both vehicles identically. Especially a short design of the beam was in focus to achieve high stiffness. The unassembled balance beam is shown in Figure 14 with and in Figure 15 without the sliding sleeve.

The sliding sleeve made of steel (1) is pressed into the brake pedal made of aluminum, to reduce the wear from the balance disk (2) which is supported inside the sleeve. The position of the balance disk to the two joint heads (3) regulates the brake force distribution.

The joint heads (rod ends) are mounted to the thread on the threaded rod (6) each by a centering disk (4) and a nut (5). When using a standard thread the core diameter of the threaded rod is too low, so it could get deformed. In order to prevent from turning the balance disk during the adjustment, it will be fixed by means of a pin (7) in the sleeve (1). With help of the centering disks the lateral clearance of the balance beam is adjusted so that the threaded rod can freely joggle in a failure of one brake circuit, but the sleeve can not move excessively during normal operation.

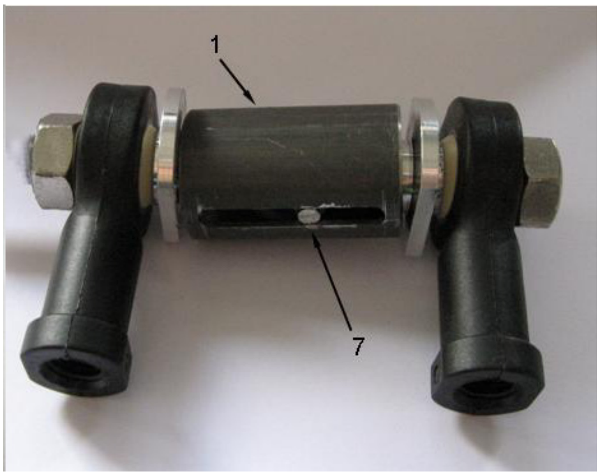

Figure 14: Balance beam with sliding sleeve [6]

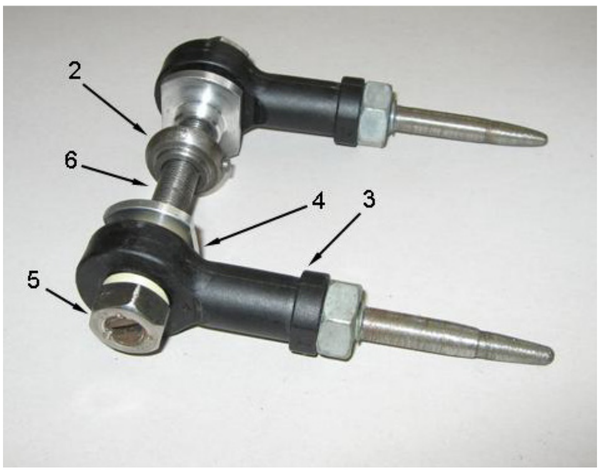

Figure 15: Balance beam [6]

In the case of error the centering disk is supported at the collar inside the sleeve. The diameter of this collar determines the lost travel during the failure of a brake circuit. Advantages of this system are simple manufacture and use of light weight-plastic bearings. A disadvantage is the wear of the balance disk in the sleeve for a unfavorable material selection, which can be reduced by the use of a 'Uniball' linkage. The use of a 'Uniball' linkage however requires an enlargement of the sliding sleeve's diameter.

To change the brake force distribution during the race an adjustment of the brake balance inside the cockpit is meaningful. For this the threaded rod of the balance beam is turned in the cockpit by means of a flexible shaft with a knob. Rest positions in this knob prevent an automatic adjustment of the brake balance while driving. The normal braking force distribution ratio of front / rear for the BRC08 is 68:32 on the balance beam.

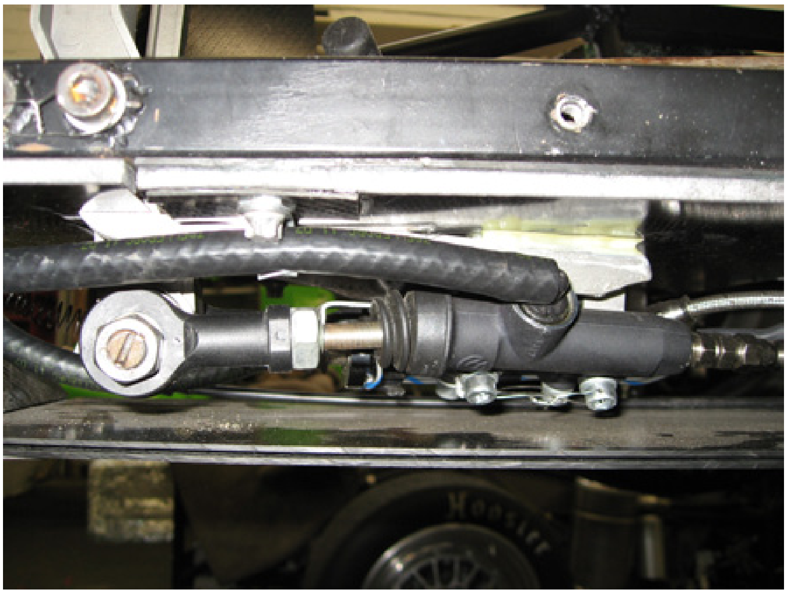

Figure 16: Master cylinder at the BRC08 [6]

4.1.2.3 Master cylinder

Figure 15 shoes the balance beam and the adaption of the master cylinders under the vehicle floor balance beam and the brake master at the BRC08. The adaption of the brake master cylinders at the BRC09 shows Figure 12. As a master brake cylinder, in principle, motorcycle master brake cylinders for foot actuation can be used.

These are build as a single-circuit device, have small dimensions and allow an external tank each. The used cylinder are made by the manufacturer Brembo ® come up for both brake circuits with a piston diameter of 13 mm, a component mass of 84 g and they are fixed mounted to the chassis. During the brake actuation, the push rod is guided on a circular line at the side of the pedal, so that the cylinder must have a joint between the push rod and piston, which can be bent at a range up to about 4 °. Figure 17 shows the basic structure. A disadvantage of this cylinder is the loss of piston force caused by the bend angle.

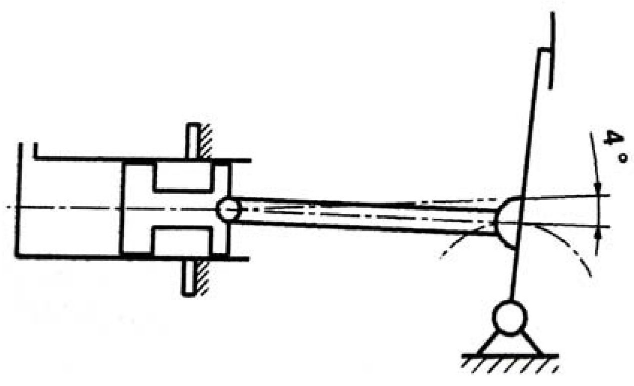

Figure 17: Master brake cylinder with angular deviation [7]

In principle, master brake cylinder with a rigid connection between the push rod and piston are possible to use the entire available actuation force.

However, the cylinder must then be pivoted. Such cylinders are equipped with a stronger push rod and have a higher weight.

The brake light of the vehicles which was prescribed by the regulations is activated by means of a hydraulic switch controlled with the brake circuit of the front axle. The switch is adapted directly to the pipe connection at the master brake cylinder, Figure 18. Because of the robust design LED lights are used, Figure 19.

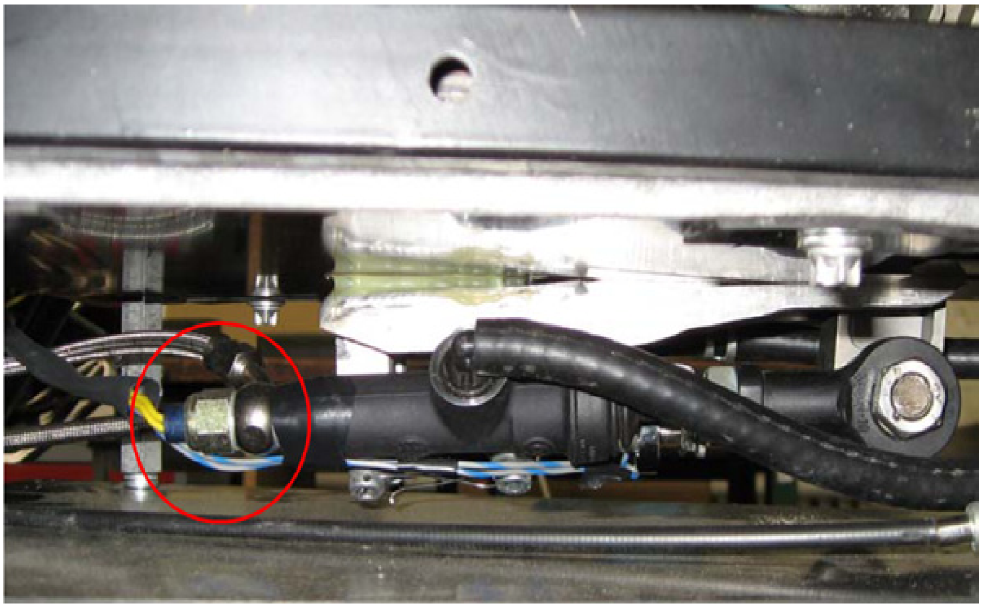

Figure 18: Brake light switch (BRC08) [6]

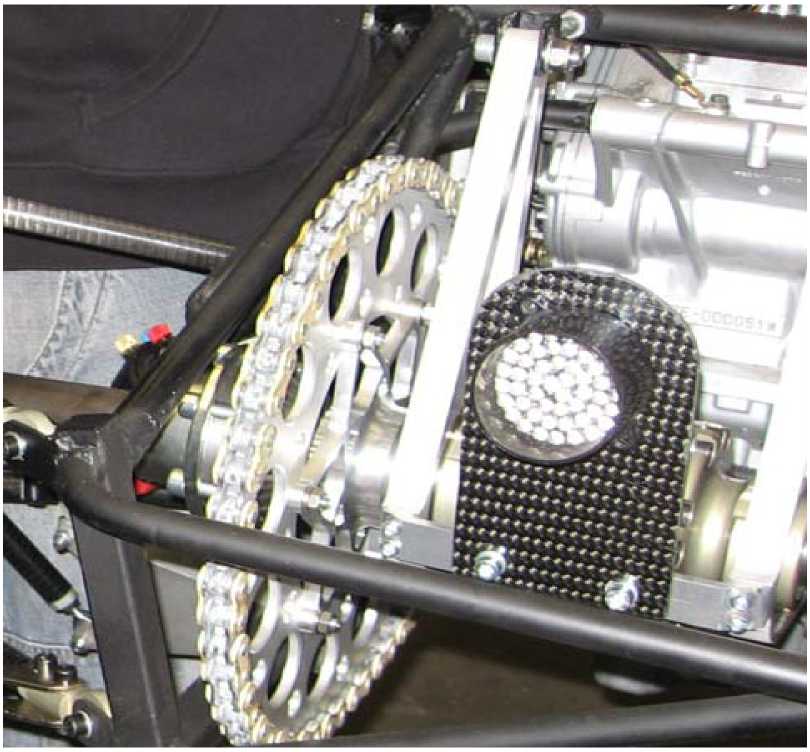

Figure 19: Brake light (BRC08) [6]

4.2 Transmission device

Figure 20: Reinforced hose at the BRC08 [6]

4.2.1 Brake lines

The transmission device of the vehicles consists exclusively of the hydraulic lines (and their contents) between the actuation device and the wheel brakes.

Brake force distributors or electronically based brake-control components are not available for the BRC vehicles.

Because of the small lengths the transmission device is exclusively build with reinforced hoses ('Stahlflexleitungen'). Although steel pipes have a higher compressive stiffness, however, the links to the necessarily flexible pipes in the axle and wheel area provide an additional risk of failure. The pipes are attached with hose clamps on subframe or frame tubes, Figure 20.

4.2.2 Brake fluid

The components used in the BRC-vehicles are mostly based on the motorcycle technology. The elastomers of the components are designed from the manufacturer for the use of polyglycolether.

Polyglycolether-based fluids are in contrast to silicone or mineral oils less compressible, so that the stiffness of the brake system is positively influenced. Another important criterion is the temperature stability of the brake fluid. Driven by the aim to save as much tire suspended mass of the brake system as possible at the wheels, only a limited potential for storage of braking heat is available, so above-average temperatures can be achieved. Modern liquids based on polyglycolether offer high boiling points that exceeds even the specification according to DOT 5 (e.g. Castrol® SRF).

4.3 Wheel brakes

It is useful to commit the number of pistons and the piston diameter of the brake calipers before planning the brake disc, because the discs have a huge design potential in terms of diameter, thickness and installation. An important parameter is the height of the brake pad, which, taking into account the pad pressure, affects the length of the pad and thus the number of brake pistons. The mean friction radius of the disc is favored by a low height of the pad. The tangential force on the disc acts on a radial surface, limited by the outer radius r_a and the inner radius r_i of the brake pad friction ring.

For the mean friction radius is valid:

$$r_B = \frac{r_a^3 - r_i^3}{r_a^2 - r_i^2} \tag{Eq. 25}$$

The required total brake piston area of the brake $A_{K,\,B}$ depends on the brake pressure of the respective brake circuit ($p_{(1/2)}$), the brake characteristic C*, the achievable braking force in the tire's contact plate of the appropriate axis $F_{B,(v/h)}$, the dynamic tire rolling radius r_{dyn} and the mean friction radius of the brake lining r_B.

$$A_{K,B} = \frac{F_{B,(v/h)}}{p_{1/2} \cdot C^*} \cdot \frac{r_{dyn}}{r_B} \tag{Eq. 26}$$

Table 2: Data of the wheel brakes (BRC08/09)

		Front	Rear
Caliper	Typ	Brembo P30B	
	∅ Piston [mm]	30	
	Lining-height [mm]	33	
	Mass [g] incl. organic based linings	470	
Brake disc	Material	42CrMo4	
	Outside ∅ [mm]	220	200

The required number of pistons can be assessed in conjunction with a desired friction ring on the disc, taking into account the permissible specific surface pressure of the brake pad and the related surface dimension.

For organic friction material the surface pressure should not exceed the value of 12 N/mm^2. The calipers and pads are identical in both vehicles on the front and rear axles. Table 2 summarizes the data from the caliper and the brake disc.

4.3.1 Caliper

Brake calipers for formula cars must be designed as light wight as possible and yet very stiff (volume capacity). For racing fixed caliper brakes are preferably used. These have a low volume capacity and can be designed light-weight. Due to the high stiffness of the fixed calipers they have a very even brake pad wear behavior. Due to high bending moments on the bridge of fist calipers it must be made very massive. Usually motorcycle fist calipers have therefore a higher mass than fixed calipers with the same performance. Because of the massive fist caliper bridge the friction radius must also be reduced of for a given rim clearance. Basically, two-piston calipers are sufficient with its brake force level for the 'Formula Student' vehicles. If the higher mass of multi piston fixed calipers (4, 6, 8 piston) can be accepted, this offers benefits of longer brake linings on the front axle. By reducing the hights of the friction area as a result of the longer brake lining brake discs with a smaller diameter can be driven. Since with multi piston calipers the piston area in general is greater than with 2-piston calipers, the brake pressure can be reduced. Also, the tilt-moment of the lining, caused by the friction force acting from the lever of the lining thickness at the pad back plate, can be compensated by the use of different piston diameters inside the multi piston calipers. However uniform surface pressure can also be reached with high-quality two-piston fixed calipers with guided brake pistons. The wheel brake of the rear axle of the BRC09 shows Figure 22. In Figure 21, the front axle brake of BRC09 as used during racing is shown.

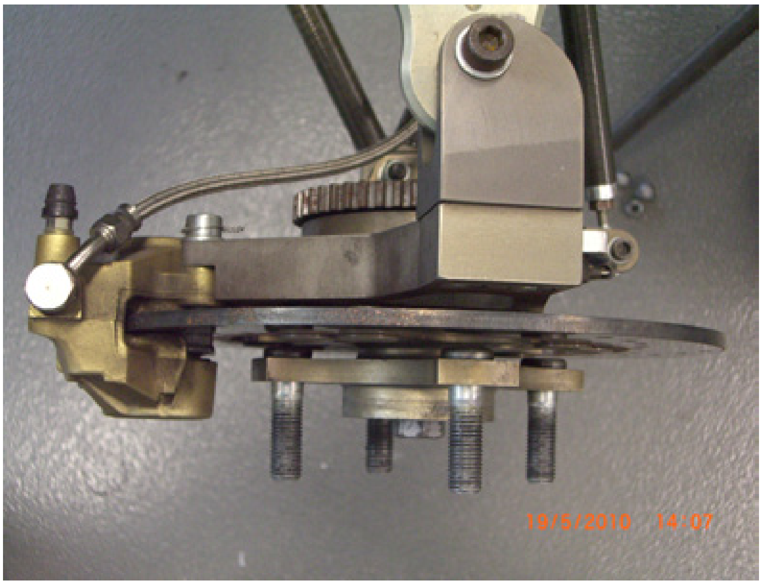

Figure 21: Front axle brake BRC09

Figure 22: Rear axle brake BRC09

The vehicles BRC08/09 are equipped with two-piston fixed calipers (P30B) with a piston diameter of 30 mm of the manufacturer Brembo®, axially mounted on the wheel carrier.

4.3.2 Linings

For choosing the brake pads it must first be decided in principle to use sintered metallic brake linings or organic based linings in conjunction with metallic brake discs. For organic based linings, the operating temperature is generally in the range 300-400° C, it must be calculated with an increased wear at a steady-temperature of approximately 200°. The friction coefficient of organic based linings is in the range of 0.3 to 0.5. Sintered brake linings can be driven at higher temperatures and have a more even friction gradient on the temperature. The friction coefficient of sintered linings is in the range from 0.4 to 0.7. Especially the cold friction coefficient is higher in comparison to organic based linings. Thereby a higher braking performance in the initial phase of the braking process can be achieved with sintered linings, so that this is better suited than organic based materials especially for the handling course of the 'Formula Student'. However, sintered linings create a high wear at the brake disc. The high thermal conductivity of brake pads with sintered linings also have high thermal requirements on the brake fluid.

For the used calipers (P30B; Brembo®) in the BRC08 and BRC09 sintered linings with a material mixing SR (SQR) are offered from TRW® . In the competition until now organic based linings with a high of 33 mm have been used. The use of sintered linings is an option for future optimization steps.

4.3.3 Brake discs

The brake disc offers extensive opportunities for the optimization of the brake system. Moreover from all parts of the brake system the brake disc is the component that can be easiest and cheapest self-made, because an internal cooling of the brake disc is not necessary.

Table 3: Brake disc design parameters

Table 3 gives an overview of the design parameters of the brake disc.

4.3.3.1 Mounting

On the driven axle (usually the rear axle) mounting of the brakes on the chassis has to be tested to keep the tire suspended mass at the wheel as low as possible. With the vehicles BRC08/09 the brake discs are mounted on the wheel carrier, because the space is very limited at the differential. With a view to tire suspended mass the brake disc mass has an appropriate high priority. Hence it follows the need for improvements regarding the heat capacity and structural strength as a function of brake material .

Analog to the disc brake adaption on motorcycles is a fixed or floating installation of the brake disc to the wheel carrier feasible. A floating mounting of the brake disk's friction ring and the disc's inner ring, which is used for the adaptation to the wheel hub, has the advantage that the expansion of the friction ring will be less hindered than the one of a fixed mounted brake disc. Thus the shielding of the friction ring under high temperatures and the threat of permanent distortion is minimized.

However, with a floating installation rattling noises can be occur while driving. This noise is avoided by using a 'semi' floating installation using floaters, which generate a preload by means of the installation of additional corrugated washers between the brake disc and the flange, Figure 23. Although the use of floating brake discs has more design and manufacturing complexity, through the use of aluminum for the inner ring of the brake disc however the mass of the floating mounted brake disc can be reduced compared with the fixed mounted disk.

Figure 23: Floater [6]

4.3.3.2 Materials and dimensions

The dimensions of the brake disc (rotor) will be determined by the selection of the material. As a economical material for the brake disc aluminum or steel can be used. Brake discs made of carbon fiber composites are often driven in racing, but for the route and driving profile in the 'Formula Student' this is not suitable because of the low cold friction coefficient and high production costs.

Brake discs of racing vehicles are often provided with holes or grooves. The grooves or holes are not intended to cool the brake disc, but clean the surfaces. By the elasticity of the lining it will be pushed into the recess during slip over and wear out during release. Thereby with perforated or grooved brake discs an increased wear of the linings occur, however, the contact pattern of the lining and the disc is more uniform and scoring can be prevented. Important is the uniform arrangement of the 'negative portions'. In the arrangement of holes in the brake disk, the mutual overlap of the holes on the orbits of the lining-area is important. As the coating is slipped over at each point of the perforation the formation of ripples through the perforations will be avoided.

The use of slots generally not have this problem. The negative portion which will slipped over from the lining should be equal throughout the extent of the brake disc. Thereby brake judder and a noise generation is suppressed. The resulting notch and thermal stresses during operation are relevant to the fatigue strength for the 'Formula Student' competition, however, the volume of material to absorb the braking energy stands solely in the foreground.

Especially when dimensioning the rear brake disc, a detailed stress analysis is required. Because of the low heat input due to lower braking forces, the rear brake discs are made theoretically very thin.

A critical load case occurs, however, if the vehicle moves backward after a "spin" with increased speed and the driver presses the brake pedal with full force. As the result of the dynamic axle load to the rear wheels and the produced high brake force at the brakes this can lead to buckling of the friction ring or tearing of the floaters.

The masses of the friction rings of front and rear axle must be aligned so that the temperature gradients during operation are similar. This demand arises from the temperature dependence of the friction coefficient of the linings. If, for example, the brake temperature at the rear axle is at a lower level, with increasing temperature difference compared to the front axle brake performance is given away because the friction coefficient of the linings achieved at the front axle a higher level than at the rear axle.

More dangerous is an increase of the friction coefficient at the rear axle. If the friction coefficient on the rear brake increases at a rising temperature faster than the coefficient on the front brake, this can reverse the locking order and the braking behaviour of the vehicle is unstable.

For the reduction of components and weight, it makes sense to use the brake disc at the same time as speed sensor wheel for a traction control system and to equip them with holes or grooves. This, however, requires the use of temperature stable speed sensors.

4.3.3.3 Realization

The friction rings of the BRC08/09-vehicles, manufactured by means of laser cut, are made of 42CrMo4, Figure 24. Here the thickness of the raw material was chosen to be 1-2 mm larger. If necessary, the friction ring can be still harden and temper, but this is not mandatory. Through the subsequent grinding and straightening of the discs, the final friction ring thickness and runout is made, Figure 25.

Figure 24: Friction ring cutted BRC08 [6]

Figure 25: Friction ring grinded BRC08 [6]

In addition to the friction ring thickness the integration of the inner ring into the wheel hub offers further potential for weight reduction. With the mounting of the brake disc's friction ring by means of floaters all screw connection will be escape, so that the distance between the flange of the rim and wheel can be reduced. Low bending moments in the journal of the axle are the results.

Figure 26 shows the advantageous integration of the brake disc's inner ring into the wheel hub on the BRC08. The diameter of the brake discs on the front axle of the BRC08 / BRC09 is 220 mm. During racing, the brake discs at the front of the BRC09 were identical to those of the BRC08. The rear brake discs of both vehicles have a diameter of 200 mm.

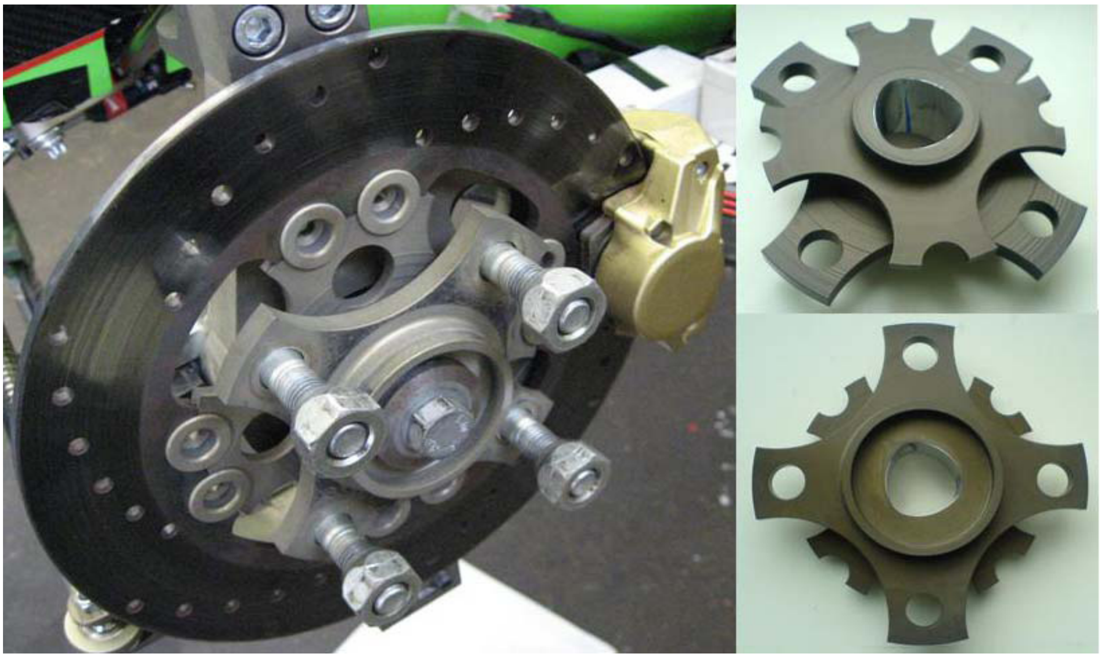

Figure 26: Integration of the disk inner ring into the wheel hub - BRC08 [6]

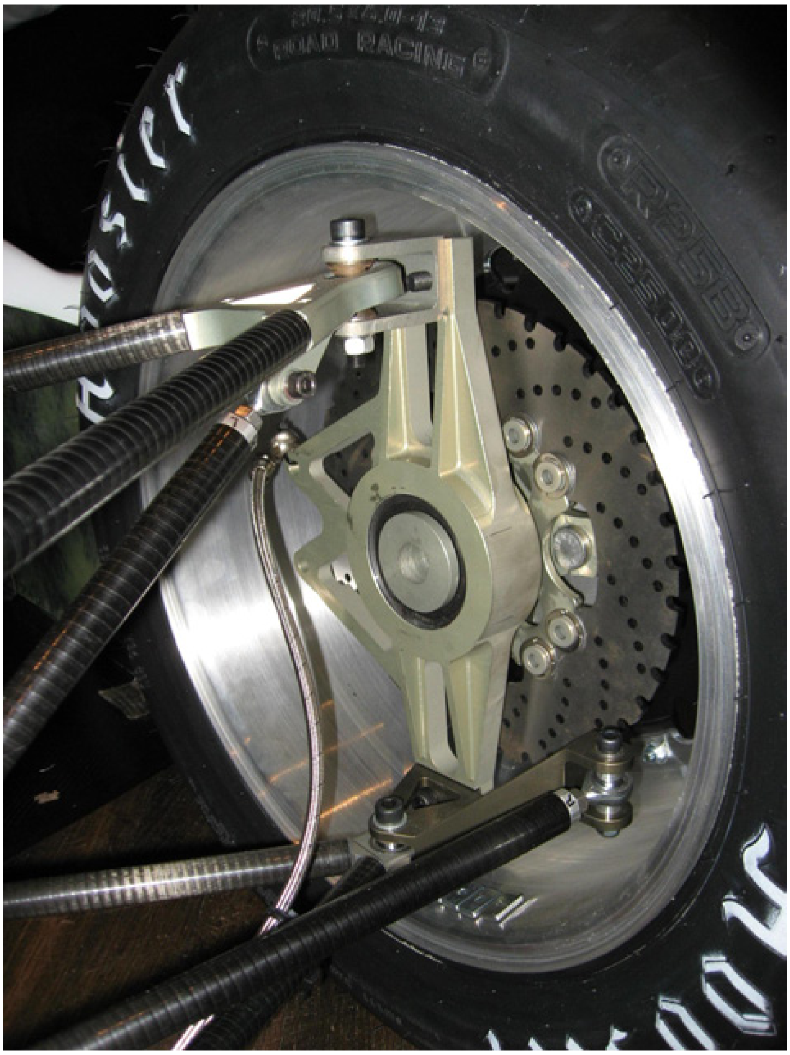

Figure 27: Alternative brake disc – front axle BRC09

For the BRC09 advanced brake discs with a toothed disc edge are available which can also used as a rotor for speed sensing (see Figure 22, Figure 27). The discs on the front axles of both vehicles are running with 5 mm thickness, to the rear, each with 4 mm.

5 Experimental verification

For the review of thermal behavior of the brake system, test runs were carried out with the race car BRC08. The experiments were carried through on the former military airport in 'Jüterborg/Altes Lager'. The test course consisted of a flat about 600 m long and 80 m wide concrete plateed path.

The selected route had nowhere flow out of tar at the interstices of the plates or growth of grass, Figure 28. Due to lack of road markings, a constant traction on the entire unpolluted track was expected. For data recording the image display unit MyChron 3 XG Log with integrated data logger of the instrument manufacturer AIM, shown in Figure 29, was used.

The data logger stores the data in an internal flash memory. By means of programmable analog channels pressure and temperature sensors (thermocouples and temperature resistance), linear and rotary potentiometers, lambda sensors and alarm LED's can be connected.

The sampling rate can be configured free for each channel. With additional input channel wheel sensors can be connected to the velocity measurement. Internal sensors allow the detection of longitudinal and lateral acceleration. There is also the possibility to measure data of the engine control CAN bus.

Figure 28: Test area in Jüterborg / Altes Lager [6]

The focus of the measurements was the detection of brake discs temperatures, because the braking force distribution is influenced by the temperature difference of the discs as a result of the temperature depending friction coefficient of the linings.

The heating of the brake discs were reached with an initial speed of 95 km/h with full decelerated braking to a stop as individual and sequenced brake maneuver with a braking pedal force of 250-270 N (just below the blocking threshold of the rear axle).

The temperature of the brake discs was recorded with conventional NiCr-Ni slip-elements.

Figure 29: VMyChron 3 XG Log [9]

The mounting of the brake temperature sensor was about 40 mm in rotational direction behind the outlet of the brake discs from the caliper. For this the sensor was positioned in the middle of the friction surface, taking into account the holes of the disc, Figure 30, The temperatures of the front and rear axle were measured on the right side of the vehicle.

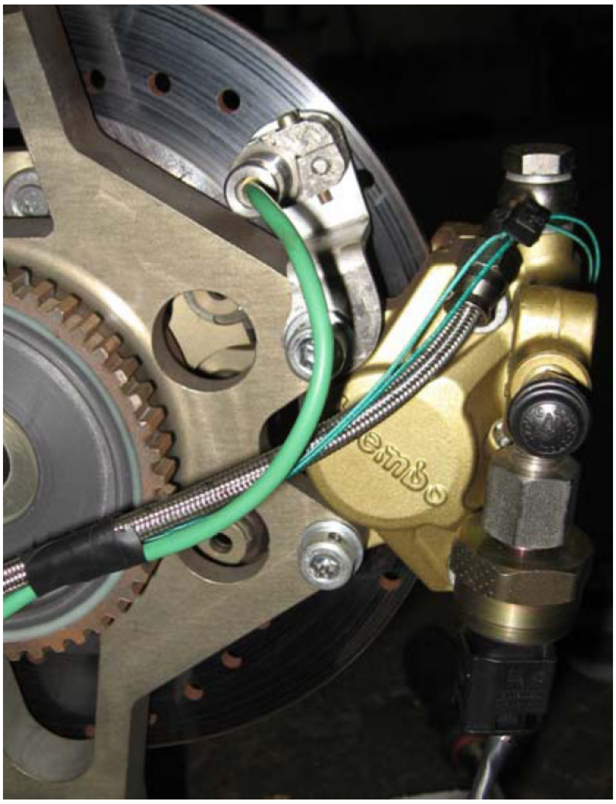

Figure 30: Mounting of the brake disc's temperature sensor at the BRC08 [6]

Because of the cleansing effect of the disk holes the perforated brake discs do not allow a permanent measurement of the friction temperature at the brake pad by means of integrated thermocouples inside the friction surface.

Besides the temperatures also the brake pipe-pressure were included in the experimental verification of the brake force distribution.

Table 4 shows the brake disc temperatures at the front axle ($\vartheta_{BS,v}$) and at the rear axle ($\vartheta_{BS,h}$), and the resulting temperature difference ($\Delta\vartheta$) as a function of the numbers of braking for a distribution of 62% front axle and 38% rear axle.

The rather low temperatures of the brake discs after 10 brakes suggests future possibilities for additional thermally motivated weight optimization.

Table 4: Comparison of the brake disc temperatures

At a fine-tuning also the respective friction temperatures on the brake lining has to be taken into account, which differ, depending on the thermal conductivity of the brake lining material, considerably from the disk temperature. Due to the low thermal conductivity of organic friction material it can be expected that this leads to a higher temperature difference between the surface of the brake disc and the surface of the lining as this will happen with sintered friction materials.

Table 5 shows the temperature difference between the brake disc and the organic based brake lining on the front axle of the BRC08.

Table 5: Comparison of brake discs and brake lining temperature at the front of the BRC08 when using an organic brake pad [6]

6 Outlook

In developing the race car BRC08 the brake system was newly designed and incorporated in the vehicle BRC09 for the competition with the exception of the rear brake discs. At the same time a closed brake calculation was introduced, used as a documentation of the 'Design Report' of the 'Formula Student' Germany.

The analysis of the conducted experimental studies points out the potential for future optimization. The operating temperatures of the brake discs, particularly at the rear axle, can be further increased. Through an exchange of the organic based brake lining material to sinter metal based linings, the braking performance of future vehicles can further be enhanced.

To avoid a mechanical destabilization of the discs during further weight reductions, for example, a larger negative part by means of an increase of the perforation is possible. Likewise, the use of light metal to produce the discs has to be tried out.

However, despite all will to optimize, the focus on the robustness of the system must not be lost because good placements in the 'Formula Student' competition can only be achieved with a fail-safe vehicle.

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