Building a drive train with Inline CVT in a Hybrid Electric Vehicle

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Summary

The University of California in Davis has built several Hybrid Electric Vehicles with which they have competed in several competitions over the years. One of their previous cars is the Chevrolet Suburban, called Sequoia, with which they competed in the Future Truck competition in the years 2000 and 2001. The car is a parallel, battery-dominant, plug-in hybrid electric vehicle with a manual transmission. To fully utilize the possibilities of this hybrid power train, Team Fate has been working on implementing an Inline Continuous Variable Transmission (CVT), instead of a manual transmission. Until the beginning of this report, Sequoia has never been running with this developed Inline CVT in it. To achieve this, it was necessary to design and fabricate numerous parts. The first goal is to get Sequoia running with the Inline CVT in it, to prove the concept of using a CVT within an SUV.

The first chapter of this report discusses all the designs of brackets that had to be made, to complete the whole drive train, so Sequoia will be able to operate with an Inline CVT. The design specifications for each different bracket are outlined and the different steps in designing the brackets are explained. The second chapter gives an outline of the implementation of the hydraulic system in the car, which operates the Inline CVT. In chapter three the design of the amplifier box is outlined, which operates the servo pumps of the CVT. A thermodynamic analysis was performed, to check if the heat sink, which was implemented to cool the amplifiers, would provide adequate cooling. The fourth chapter outlines the control strategy behind the inline CVT. And finally in chapter five, vehicle testing is described, along with future work required for completion of this project.
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Introduction

The University of California in Davis (UCD) has been working on Hybrid Electric Vehicles for many years. The objective of their program are much less fuel consumption and a switch to the use of electric energy from other than oil resources. In addition, the objective is to be able to use renewable energy from the sun and wind, especially personally owned systems. A lot of knowledge has been acquired in many years and with this the research team tries to develop newer, better and more efficient vehicles which will be sustainable and environmentally friendly. One of the projects which UCD is currently working on, is the one which involves Sequoia, a Chevrolet Suburban, which was rebuilt as a Hybrid Electric Vehicle in the years 2000 and 2001. This car had a remarkable finish in the Future Truck competition and won first place, of all the competing teams. Later, the decision was made to replace the manual transmission in Sequoia with an Inline CVT in order to fully utilize the benefits of a hybrid powertrain and also meet U.S. market demands for automatic vehicles. The so called Inline CVT, which has its input axle inline with its output axle, and is highly desired for a rear wheel driven car, such as Sequoia. One of the benefits of this CVT is the use of a chain, which is able to transfer high amounts of torque through the drive train, so it is applicable for bigger vehicles like SUV’s and trucks. Until now, the replacement of the manual transmission with the Inline CVT has not yet been accomplish, within Sequoia. The main goal is the proof of concept of this CVT within an SUV. In the future, there will be future testing and designing to improve this transmission and reach higher efficiencies.

My internship report will outline all of the necessary steps that I had to make to complete this car, so it would be running with an Inline CVT in it. First of all, for the rebuilding of the drive train there were a lot of brackets needed. The design of all of the brackets are explained in the beginning. All the design specifications are outlined and the steps that have been taken are explained, for each of the necessary brackets. Because there was a very close deadline to reach, the weight optimization of the complete drive train, has not been a significant priority. Only the stiffness and volume have been optimized as much as possible in the time available. Once the brackets were completed, all the shafts to connect the drive train with the power train and the wheels had to be designed and fabricated. Additionally, the hydraulic system of the CVT had to be built into the car. Also, a box had to be designed and fabricated for the CVT controller amplifiers. Furthermore the CVT controls were implemented.
into Sequoia and this is outlined towards the end of the report. All future work required in order to finish this project is outlined at the end of this report.
Chapter 1

History of the Project

A structure outline is given to have a clear view about the organization in this research group and their history outline within the different challenges, in which they compete. Furthermore an outline of the history of this Inline-CVT project is given to make a better view of what has been done in the past.

1.1 Structure of HEV Center Group

Within the Department of Mechanical and Aeronautical Engineering at UC-Davis there are several Research Areas, Centers and Laboratories. One of the Centers is the Hybrid Electric Vehicle Center which is completely run by graduate students and undergraduate students. The structure of the HEV Center in the year 2006 is outlined in Figure 1.1.

The vehicle design group of graduate and undergraduate students which run the HEV Center is called Team Fate. This name is derived from the 1960’s feature film The Great Race, in which Professor Fate continuously strives to thwart his opposition never giving up, never surrendering. Team Fate, Prof. Frank’s vehicle design group for more than 25 years, has produced a number of advanced concept vehicles and control designs. For a more detailed description, there will be refer to Team Fate’s website: http://www.team-fate.net/

1.1.1 Challenge X (2004-2007)

Currently Team Fate is competing in the Challenge X competition and it is one of the seventeen teams who will re-engineer a Chevrolet (GM) Equinox sport utility vehicle to minimize energy consumption, emissions, and greenhouse gases while maintaining or exceeding the vehicle’s utility and performance.
1.1.2 Yosemite (2002-2003)

Yosemite is a battery dominant hybrid electric vehicle based off of a 2002 Ford Explorer. Yosemite uses a 1.9L Saturn engine and a 75kW UQN electric motor as its primary drive system. It also incorporates a 60kW Enova electric drive system to further enhance drivability and efficiency.
1.1.3 Sequoia (2000-2001)

Sequoia is a modified 2000 Chevrolet Suburban. A Parallel Hybrid Powertrain is implemented with a DC brushless electric motor (EM) and an internal combustion engine (ICE). A second electric motor is used on the front wheels to provide 4WD. Both the engine and electric motor can operate simultaneously or independently, allowing a flexible and efficient control strategy.

![Figure 1.3: Sequoia (2000-2001)](image)

1.1.4 Coulomb (1998-1999)

Coulomb is a hybrid-electric Mercury Sable AIV. Coulomb features a prototype all aluminum body, composite body panels, and an advanced CVT-based powertrain. Combined with a 75Kw electric motor, a 659cc Subaru engine, and an 18.6Kwh battery pack Coulomb is one of Team Fate’s most advanced vehicles yet.

1.1.5 Joule (1996-1997)

Joule is a 1996 Ford Taurus converted to a hybrid electric vehicle by students of the UC Davis FutureCar Project. The twelve schools selected to compete in the 1996 FutureCar Challenge were assigned the task of matching government and industry (PNGV) efforts to create a mid-size family sedan that attains 80mpg while maintaining the performance, utility, and cost of a conventional car. At the 1997 FutureCar Challenge, Joule won many individual awards on its way to winning the competition.
1.1.6 AfterShock (1993-1995)

AfterShock is a parallel hybrid designed from the ground-up. The aluminum frame and composite body were designed and manufactured to be lightweight and extremely aerodynamic. Aftershock achieves an incredible 73 mpg.

1.2 History of the inline CVT project

UC-Davis and GCI developed a new concept [Fra04] for a CVT, which can be used as a direct displacement for transmissions in rear wheel driven SUV’s or small trucks. And the first prototype was built and shown at the 2004 International Continuously Variable and Hybrid Transmission Congress [Bro].
Furthermore the working principle of the inline-CVT is also outlined in a TU/e Master Thesis Report [Oud05] in more detail. After the prototype was built there has been developed a test rig to analyse the Inline CVT and prove that it was a working principle. This is outlined in [Sch05] and there it is also explained why it was necessary to make a test rig and do some tests for example to determine it’s efficiency.

After all these things were done it was necessary to prove its concept into a SUV. And here it was decided to convert one of the previous cars in the Team Fate History, Sequoia, into a Hybrid Vehicle with inline CVT. After disassembling the originally used manual transmission and some other parts like the drive shaft, there could be started by designing and building the brackets to put in the inline CVT into Sequoia, which is further explained in the rest of this report.
Chapter 2

Drive train brackets and shafts

After an inspection of Sequoia, it was very clear that it was necessary to make several brackets to connect all of the necessary parts of the drive train. In the beginning, only the engine, the electric motor and the empty Inline CVT case were hanging under Sequoia. Still, there was some space left, in which originally the manual transmission was placed. Additional components that needed to be installed included, the CVT case bracket, the CVT slip clutch bracket, the slip clutch itself, the slip clutch drive shaft bracket and the first part of the drive shaft with it’s u-joint. In Figure 2.1, it can be seen how much space was available to install these parts. The slip clutch, which is necessary to make sure that the Inline CVT will not be damaged by very high torque that could occur in the drive train, was donated by Borg Warner. This lead to a specific volume which would be taken by the slip clutch which needed to be used. The space left was available for the three brackets, but this was not much.

2.1 Overall design specification

One of the design specifications was, that the original manual transmission, had to be able to fit in the car after the Inline CVT was installed. It is necessary to demonstrate, that the Inline CVT is directly exchangeable with the manual transmission. It is desirable to demonstrate to the car manufacturers that the Inline CVT can directly replace the manual transmission, and is therefore a good viable alternative requiring few significant alterations. This led to the design specification of not adjusting the chassis of Sequoia. This would mean that the original bracket for the manual transmission, which is shown in Figure 2.2, could be replaced by a re-designed CVT chassis bracket. Also this way the original mounting locations on the chassis can still be used for the new bracket.
Figure 2.1: The space left in the transmission compartment

Figure 2.2: Original manual transmission bracket
2.2 The CVT slip clutch bracket

2.2.1 Design

The CVT slip clutch bracket, which is necessary to mount the slip clutch onto the CVT case, needed to be as short as possible. This way the slip clutch is positioned as close as possible to the CVT case and there would be enough space available for the rest of the drive train, such as the drive shaft U-joint. Also the bracket needs to align the slip clutch axle with the CVT axle, to make sure the bearings will have a good lifetime. And because there was not much space left between the bottom floor of the car and the slip clutch case it was necessary to design the bracket in such a way that the slip clutch electric connector was still reachable. The first design which was made is shown in Figure 2.3. Here the length of the cylinder would be too long and there would not be any space left to put the slip clutch drive axle bracket. This bracket was based on an aluminium design and required a great thickness on the cylinder wall to allow for proper bolt threading without compromising strength. Afterwards it was decided to make the bracket of steel, so that the thickness of the cylinder wall could be less and this would lead to a much shorter bracket design. The second version would be 0.11m long instead of 0.17m and also the weight would be around 4kg instead of 6.25kg. The second and final design is shown in Figure 2.4. Here is also seen that difference between the two designs lead to the use of regular bolts with nuts, instead of a thread in the aluminium, which meant a thick cylinder wall. Some stiffness calculations have been made to compare the two versions, but unfortunately the computer capacity which was available could only do the first version and was not able to complete the second one, so no accurate comparison could be made of the two designs. In Appendix A.1 the technical drawing from the final design can be found.
2.2.2 Fabrication

When the fabrication of the CVT slip clutch bracket was started, the decision was made to use material that was still in stock. This led to the use of scrap material which was not of the correct size. So the final fabrication became a little bit different and this can be seen in Figure 2.21. In this picture it can be seen that the use of nuts was not necessary anymore. Instead there is created a thread in the cylinder wall of the bracket. The thread was not completely full at the end, but it would be strong enough to mount the slip clutch safely. The amount of weight that the bracket would be able to carry was not much in the vertical direction. Only the slip clutch itself and the drive shaft with it’s U-joints constitute the forces in this direction. This drive shaft is also supported by the differential gear in the rear. Also, the momentum in the bracket would never reach that level, because then the slip clutch would slip and the force would no longer be applied to the bracket.

2.3 CVT chassis bracket

2.3.1 Design of chassis bracket

In the manual transmission configuration, the bracket to support the drive train, which is in picture 2.2, was bolted to the chassis. Because the two transmissions need to be interchangeable, the bracket which would support the drive train with the CVT in it needed to be designed completely new. But some of the parts would be the same as the old bracket, because they both would use the same mounting locations on the chassis. After a lot of measuring and hand drawings, the first design was presented. It is shown in Figure 2.5 and it can be seen that the support bracket is one piece of a hollow rectangular beam. On one side there is a support welded to it, on which the rubber dampener would be placed and the small CVT bracket from Figure 2.16 would be bolted
Unfortunately this design would have some disadvantages. First of all, the forces in the bracket are vertical and shown in Figure 2.5 and it shows that there would be a torsion moment in the bracket. This is not preferable and the best thing would be to make a design where the force would be in line with the symmetric axle of the beam, so no torsion would occur. The other disadvantage would be, the free displacement in vertical direction, by the small CVT bracket. It would hit the beam when the forces would become too large.

Because of the disadvantages a second design was made, which is shown in Figure 2.6. Here the small CVT bracket, which is explained in more detail below, is made smaller. This way the vertical force is almost in line with the axle of the beam and only a small torsion will occur. Also the free displacement in the vertical direction has become more and can be adjusted later, if necessary, by milling a bit more of the bracket, without losing that much of it’s stiffness.

To compare both designs a stiffness analysis was done on the brackets. With
the use of a software packet called Ansys version 10.0 the calculations were done. This packet was chosen because it could load the part files, which were made in Unigraphics, immediately without making any conversions to them. In Figures 2.7 and 2.8, it can be seen how the fixed points and forces are acting on the bracket. The vertical load, which acts on the bracket, is taken by estimation with relation to the weight presented in [Roo]. In Figures 2.9 and 2.10 the results of the total displacement can be seen and in Figures 2.11 and 2.12 the results of the strain test can be seen. Also the stresses in the bracket are shown in Figures 2.13 and 2.14. From these figures can be concluded that the maximum deformation goes down from 0.131mm to 0.115mm. Both levels of deformation are tolerable, so this was not a reason to go with the second design. Also, the strain in both designs fell within an allowable tolerance, and therefore was not a deciding factor for design selection. The same can be said for the stress measured on the members. The shear stress for aluminum is around 25Gpa and as the results show, the max stress was around 23Mpa for the first design and 18Mpa for the second design. Therefore, the selection of the second design was a result of the criteria previously discussed, including the displacement tolerance between chassis bracket and CVT bracket and the smaller CVT bracket. The fact that the weight of the second design was slightly higher than the first design, respectively 7.12kg and 6.68kg, was not considered to be a significant reason for selecting the first design.
Figure 2.9: Deformation on first design of the CVT chassis bracket

Figure 2.10: Deformation on second design of the CVT chassis bracket

Figure 2.11: Strain in first design of the CVT chassis bracket
Figure 2.12: Strain in second design of the CVT chassis bracket

Figure 2.13: Stress in first design of the CVT chassis bracket

Figure 2.14: Stress in second design of the CVT chassis bracket
2.3.2 Fabrication of chassis bracket

To make this bracket, a certain series of steps had to be followed. At first, the parts would be welded together and the holes for mounting them to the chassis would be drilled at the end. This is done, because when the aluminium part is welded together, the part will undergo plastic deformation as a result of the stresses that occur when heating it by welding. When the holes are drilled at the end, the bracket will fit nice into the chassis, without making any adjustments afterwards. But because of the small available budget, it was decided to bolt the pieces together. The aluminium welding would be quite expensive and to weld it ourselves was not possible because the welding machine didn’t have enough power for such a welding job. By bolting the parts together, the beams would plastically deform, if the bolts and nuts are over-tightened. For this reason, spacer blocks (aluminium) were placed between the beams to prevent this from happening.

2.3.3 Design CVT case bracket

The small CVT case bracket, which connects the drive train to the bracket on the chassis, through a rubber damper, is also designed in two steps. The first design looked like the picture in Figure 2.15. But to gain more stiffness a second design was made as can be seen in the picture of Figure 2.16. The problem which had to be solved in this design was the fact that all the hoses and lines connected to the CVT case on this side, still had to be connected. This also was the case with the design of the big CVT bracket. With some adjustments and the use of different connectors for these oil lines, it would all be possible.

Again some stiffness analyses were performed to check if the brackets were structurally sound. In Figures 2.17 and 2.18 the forces are shown, which apply on the bracket and in Figures 2.19 and 2.20 the total deformation are shown.
As can be seen, the brackets should both be okay and the second design is far better than the first one with respect to the stiffness of it. The second design is about four times as stiff as the first design.

All the different parts that are used to complete the CVT bracket are presented by technical drawings in Appendix A.2.

2.3.4 Fabrication CVT case bracket

The fabrication of the CVT chassis bracket was done by writing a CNC program for the milling machines which were available in the workshop at the department of MAE. After milling the parts, they were welded together. It can be seen that the small CVT bracket is not exactly the same as the design. This is due to the fact that a slight miss measurement was made and during the welding a misalignment occurred. Luckily these mistakes had no big impact on the assembly, because the mistakes could be taken out by installing an extra plate. The misalignment was in the direction in which a little bit of space was left and
Figure 2.18: Forces working on the second design of the small CVT bracket

Figure 2.19: Total deformation on first design of the small CVT bracket

Figure 2.20: Total deformation on second design of the small CVT bracket
also occur due to the effect of stresses in the aluminum after welding, which cause bending. Also the parts of the chassis bracket were made on the milling machines in the workshop. Afterwards all the parts were bolted together as explained previously. The only difference between the design and fabrication was the squared blocs, instead of the 4 cylindrical busses.

2.4 The shaft connections

To connect all of the components in the drive train three shafts had to be made. They were all made with spline fittings, by a company named Gear Enterprises, which is located in Stockton, California. For contact information see Appendix D. All the shafts were drawn in Unigraphics in sketch style, because it would take too long to do this perfectly. Gear Enterprises does not use graphic software to make their splines and they only needed the spline geometry and related dimensions, in order to be able to fabricate these shafts.

2.4.1 The electric motor CVT shaft connection

The electric motor connects to the CVT by a shaft which consists of an internal and external spline. The internal spline fits onto the electric motor site and has a standard spline geometry, which is given in Appendix B.4.1. The external spline fits into the CVT site and has a standard DIN 5480 spline. The drawing of the shaft is shown in Figure 2.22 and the technical drawing used for Gear Enterprises can be found in Appendix B.1. The final shaft is shown on the right hand site in Figure 2.26.
2.4.2 The CVT slip clutch shaft connection

The CVT connects to the slip clutch by a shaft which consists of two external splines. The external spline which fits into the CVT is again the standard DIN 5480 spline. The external spline which fits into the slip clutch was also a standard spline, of which the specs are given in Appendix B.4.2. The drawing of the shaft is shown in Figure 2.23 and the technical drawing used by Gear Enterprises can be found in Appendix B.2. The final shaft is shown in the middle of Figure 2.26.

2.4.3 The slip clutch drive shaft connection

The bracket that connects the slip clutch to the drive shaft also needs to be as small as possible, because otherwise there will not be any space left for the u-joint of the drive shaft. Also it is not the intent to change the slip clutch, since the team has been donated two slip clutches, one for the Challenge X
car and one for Sequoia. If one breaks, the other one needs to be replaceable between both cars. And furthermore the coupling needs to be moveable in the axial direction of the shaft, due to the suspension of the car. In Figure 2.24 it is explained how much in worst case scenario the displacement would be, by using the Pythagorean theorem $a^2 + b^2 = c^2$. The question mark stands for 0.019 m in the approximation of about 0.25 m suspension displacement. The drive shaft is about 1.625 m long between the u-joints. In Figure 2.25 the bracket design is shown.

\[ +/\ -0.25\text{m} \]
\[ 1.625\text{m} \]

Figure 2.24: Drive shaft displacement

Unfortunately the old drive shaft which was used in the manual transmission configuration was a bit too long. And so this drive shaft had to be made shorter, by 155 mm. Because this is still quite a bit shorter, the shaft was brought to a company named Drive Shaft Service in Sacramento, California, where they made it shorter and balanced it. For the contact information of Drive Shaft Service, you’ll be referred to Appendix D.

\section{2.5 Total assembly design of the drive train}

With all the above described brackets the total assembly of the drive train looks like in the picture of Figure 2.27. To have a better idea of the internal workings, an exploded view is shown in Figure 2.28.
2.6 Total assembly of the drive train

A good picture of the complete drive train assembly can be found in Figure 2.30. However, before the total drive train could be assembled several things had to be done. These are described in the sections below.
2.6.1 CVT case adjustments

According to report [Sch05] there still were some problems with the inline CVT. One of them was the fact that in the lower ratio range, around 0.5, the CVT chain would hit the CVT case. As can be seen in Figure 2.29 there are some scratches on the CVT case. It was decided to grind more material out of the CVT case so this would not occur anymore. Also a blue paint was inserted again, just in case it would still hit the CVT case after the grinding. Then it would be more easily observed which areas of the case still required removal of material.

2.6.2 Oil sump pump and oil level gauge placement

Before the complete transmission could be placed in the car, it was first necessary to install the oil sump pump and the oil level gauge somewhere near the
transmission. The oil sump pump was taken off the test rig and mounted on the chassis bracket, which can be seen in Figure 2.30. Also the circuit of the oil sump was adjusted by adding an extra output towards the primary CVT pump. The reason this was necessary is explained in the next chapter of this report. For the oil level gauge the same principle was taken as in report [Sch05], but some couplings were bought to place it near the CVT case, where some space was available, as can be seen in Figure 2.31.
Chapter 3

Hydraulic system of the CVT

After completing the hardware of the drive train the hydraulic system to operate the Inline CVT had to be built into the car. And some adjustments, in comparison to the system used in the report [Sch05], had to be made.

3.1 CVT pump bracket

To place the primary and ratio pump into the car, the original mounting bracket of the test rig, as described in the report [Sch05], was disassembled. And the plates with the electric motor, the pump and the gear set on it were directly mounted in the car. The variables which had to be taken into account before doing this were the space available in the car and the position of the pressure gauges. Not everywhere was enough space in the car to place the pumps. And with that came the fact, that the pressure gauges had to be visually accessible to be able to control the CVT. The plan was to place the car on the dyno stand and try to control the CVT while the car was running. To do this, the gauges had to be located somewhere where they would be visible while controlling the transmission. This lead to the placement of the pumps quite high in the car, as can be seen in Figure 3.1. This would have some disadvantages regarding the supply of oil to the primary pump, which is explained in the next section.

3.2 Operating system adjustments

The operating conditions of the primary pump are very low RPM, whenever the pressure has been built up in the system, because it only has to compensate for leakage of oil. Since the primary pump is located far above the level of the oil sump in the CVT (approximately 1 m), this will lead to a back flow of the oil to the oil sump. And no pressure can build/hold anymore in the system. To compensate for this altitude difference, an extra oil line from the outgoing oil sump pump is connected to the beginning of the ingoing line of the primary
pump, as can be seen in Figure 2.30 and Figure 2.31. So the oil sump pump, which is primarily used for filtering the oil and supplying the oil to the plain bearings, is now also used to supply enough oil in the intake line of the primary pressure pump.
Chapter 4

CVT control components

After completing the hardware of the drive train the hardware of the controls for the drive train had to be designed and made.

4.1 CVT pump amplifier box design

To operate the electric motors for the primary and ratio pump some amplifiers are necessary. Since these are electronic devices, it was preferred to have them in a box which was as watertight as possible. First of all two designs were made with different arrangements of the amplifier boxes, capacitors and connectors in it, as can be seen in Figure 4.1 and Figure 4.2. The design of Figure 4.2 was the most compact design and so this design was chosen over the other. A thermodynamic analysis was then performed to figure out if there was some extra cooling needed for these amplifiers. Because, out of experience, it was not clearly known how much the amplifiers would need to be cooled. One of the previous cars did not have any cooling at all and operates well. At another car there was liquid cooling applied to it, but this was more overkill towards the cooling needed. The decision was made to do an analysis of it, in order to ensure safe operation of the amplifiers.

In Appendix C the specifications of the amplifier are outlined, and it can be seen that it dissipates 300W maximum at a continuous current. The electric motors will never be running in such a way that they need continuous current from the amplifiers for a long time. Also they are operating at a lower current than what maximum is allowed by the amplifiers. So the 300W will never have to be dissipated at all. The decision was also made to check how much heat could be dissipated by integrating a heat sink on the sides of the amplifier box. First of all, there was a check if it would be helpful to integrate a heat sink so more heat could be dissipated, by using the theorem of [Inc96], pages 117 and 118. A fin is used to increase the heat transfer from a surface by increasing the effective surface area, but the fin itself represents a conduction resistance to heat transfer from the original surface. So the use of a fin will not always
Figure 4.1: First design of arranging parts in amplifier box

Figure 4.2: Second design of arranging parts in amplifier box
increase the heat transfer rate. By evaluating the fin effectiveness $\varepsilon_f$ it can be checked if it is useful to integrate a heat sink. If $\varepsilon_f > 2$ the implementation of a fin is useful.

$$\varepsilon_f = \left( \frac{kP}{hA_c} \right)^{1/2}$$  \hspace{1cm} (4.1)

Where:

- $k =$ thermal conductivity
- $P =$ fin perimeter
- $h =$ convection heat transfer coefficient
- $A_c =$ fin cross-sectional area

Because only simple tools for the milling machine were available, we had to stick to rectangular shaped fins. With the help of Figure 4.3 and the material properties of aluminum, the fin effectiveness could be calculated. Aluminum has a thermal conductivity range between 168\,W/m\cdot K and 237\,W/m\cdot K, 180\,W/m\cdot K was taken as assumption. The convection heat transfer coefficient for a forced convection gas would be in the range of 25\,W/m^2\cdot K and 250\,W/m^2\cdot K. We assumed a slightly forced convection, because the car is moving most of the time, but the box is placed under the hood, so 50\,W/m^2\cdot K was assumed. The width ($w$) of the box is 0.27\,m and the fin thickness ($t$) was chosen to be 3\,mm.

Substitution of the parameters into 4.1 gives:
\[ \varepsilon_f = \left( \frac{180 \cdot (2 \cdot 0.27 + 2 \cdot 0.003)}{50 \cdot (0.27 \cdot 0.003)} \right)^{1/2} = 49.26 \]

This means that the use of a heat sink is useful, since 49.26 \( \gg \) 2.

The next step was to calculate how much heat would be dissipated by using a heat sink in the amplifier box. Therefore we first have to obtain the temperature distribution along the fin. We begin by performing an energy balance on an appropriate differential element within the fin. To simplify the analysis, certain assumptions are made. Only one-dimensional conditions in the longitudinal \( (x) \) direction are assumed, because the fins we create are relatively thin. This means that the temperature change in the longitudinal direction will be much larger than that in the transverse direction. Furthermore, the fin is considered to be at a steady-state condition and the thermal conductivity is assumed constant within the aluminum. Also radiation from the surface is assumed negligible and the convection heat transfer coefficient \( h \) is uniform over the surface. By doing this, we can follow the [Inc96] on page 113 and 114. This will give us the general form of the energy equation for one-dimensional conditions.

\[
\frac{d^2T}{dx^2} + \left( \frac{1}{A_c} \frac{dA_c}{dx} \right) \frac{dT}{dx} - \left( \frac{1}{A_c} \frac{h}{k} \frac{dA_s}{dx} \right) (T - T_\infty) = 0 \tag{4.2}
\]

To solve Equation 4.2 for a straight rectangular fin we say that the base surface temperature is \( T(0) = T_b \) and extends into a gas (air) of temperature \( T_\infty \). For the prescribed fin in Figure 4.3, \( A_c \) is a constant and \( A_s = P x \), where \( A_s \) is the surface area measured from the base to \( x \). Accordingly, with \( dA_c/dx = 0 \) and \( dA_s/dx = P \), Equation 4.2 reduces to

\[
\frac{d^2T}{dx^2} - \frac{hP}{kA_c} (T - T_\infty) = 0 \tag{4.3}
\]

A simplification of the form of this equation is made, by transforming the dependent variable by defining an excess temperature \( \theta \) as

\[
\theta(x) \equiv T(x) - T_\infty \tag{4.4}
\]

which gives, \( d\theta/dx = dT/dx \), because \( T_\infty \) is a constant. By substituting Equation 4.4 into Equation 4.3, we obtain

\[
\frac{d^2\theta}{dx^2} - m^2 \theta = 0 \tag{4.5}
\]

with

\[
m^2 \equiv \frac{hP}{kA_c} \tag{4.6}
\]
Equation 4.5 is a linear, homogeneous, second-order differential equation with constant coefficients. Its general solution is given by the form

$$\theta(x) = C_1 e^{mx} + C_2 e^{-mx}$$

To evaluate the constants $C_1$ and $C_2$, it is necessary to specify boundary conditions. The first boundary condition is specified by the temperature at the base of the fin ($x = 0$). This $T_b$ is taken with a safety factor from the specification of the amplifiers in Appendix C. Here it can be found that the amplifiers will shut down if they exceed a temperature above $650^\circ C$. So $T_b = 600^\circ C$ is chosen and together with an approximation of the gas temperature $T_\infty = 250^\circ C$ under the hood. In driving conditions, we now get

$$\theta(0) = T_b - T_\infty = 60 - 25 = 35 \equiv \theta_b$$

The second boundary condition is specified at the fin tip ($x = L$), and there are four different possible physical outcomes, as outlined in [Inc96], pages 114 to 117.

- Option A, a convection heat transfer: $h\theta(L) = -k\frac{d\theta}{dx}|_{x=L}$
- Option B, adiabatic: $\frac{d\theta}{dx}|_{x=L} = 0$
- Option C, prescribed temperature: $\theta(L) = \theta_L$
- Option D, infinite fin ($L \to \infty$): $\theta(L) = 0$

Option D is not the case in our situation and also we don’t know if options B and C would apply to our situation. So there is chosen to go by option A. By applying an energy balance to a control surface about the tip (Figure 4.4), we obtain


\[ hA_c[T(L) - T_\infty] = -kA_c \left. \frac{dT}{dx} \right|_{x=L} \]

or

\[ h\theta(L) = -k \left. \frac{d\theta}{dx} \right|_{x=L} \tag{4.9} \]

This means, the rate at which energy is transferred to the fluid by convection from the tip must equal the rate at which energy reaches the tip by conduction through the fin. And by substituting Equation 4.7 into Equations 4.8 and 4.9, we obtain, respectively,

\[ \theta_b = C_1 + C_2 \tag{4.10} \]

and

\[ h(C_1 e^{mL} + C_2 e^{-mL}) = km(C_2 e^{-mL} - C_1 e^{mL}) \]

Solving for \( C_1 \) and \( C_2 \), it can be shown that

\[ \frac{\theta}{\theta_b} = \frac{\cosh m(L - x) + (h/mk) \sinh m(L - x)}{\cosh mL + (h/mk) \sinh mL} \tag{4.11} \]

And the form of this temperature distribution is shown schematically in Figure 4.4. By having the temperature distribution within the fin, the total heat transfer rate \( q_f \) of the fin can be calculated. By applying Fourier’s law at the fin base

\[ q_f = q_b = -kA_c \left. \frac{dT}{dx} \right|_{x=0} = -kA_c \left. \frac{d\theta}{dx} \right|_{x=0} \tag{4.12} \]

and knowing the temperature distribution, \( \theta(x) \), \( q_f \) may be evaluated, giving

\[ q_f = \sqrt{hPkA_c\theta_b} \frac{\sinh mL + (h/mk) \cosh mL}{\cosh mL + (h/mk) \sinh mL} \tag{4.13} \]

So filling in all the parameters in Equation 4.13 we get the heat transfer rate of one fin

\[ q_f = \sqrt{50 \cdot 0.546 \cdot 180 \cdot 8.1 \times 10^{-4}} \cdot 35 \cdot \frac{\sinh(13.68 \cdot 0.012) + \left( \frac{50}{8.1 \times 10^{-4}} \right) \cosh(13.68 \cdot 0.012)}{\cosh(13.68 \cdot 0.012) + \left( \frac{50}{8.1 \times 10^{-4}} \right) \sinh(13.68 \cdot 0.012)} = 422 \text{W} \]

with
\[ L = 0.012m \]
\[ P = 2w + 2t = 2 \cdot 0.27 + 2 \cdot 0.003 = 0.546 \]
\[ A_c = wt = 0.27 \cdot 0.003 = 8.1 \cdot 10^{-4} \]
\[ m = \sqrt{\frac{hP}{kA_c}} = \sqrt{\frac{50 \cdot 0.546}{180 \cdot 8.1 \cdot 10^{-4}}} = \sqrt{\frac{27.3}{0.1458}} = 13.68 \]

If we would look at the worst case scenario, in which the temperature under the hood would be \( T_{\infty} = 55^0C \) and \( \theta_b \) would become 5, the \( q_f \) would be around 63W per fin. With 16 fins for each amplifier this would mean \( q_f \) will be around 1000W for each amplifier, which is more than necessary, compared to the maximum dissipation of 300W per amplifier. Within this analysis there has to be taken into account that we assumed the base temperature homogenous with the amplifier, which is not correct, but reasonable.

So after all we could conclude that implementing a heat sink in the amplifier box would be helpful.

### 4.2 CVT pump amplifier box fabrication and placement in the vehicle

Now that it would be useful to integrate a heat sink into the amplifier box, this was done by milling it in the side plates. The direction of the fins were chosen to be vertical after placing it in the car. This was done because there will be only a slight airflow under the hood of the car and so we would expect to have a airflow of rising warmer air upwards. In Figure 4.5 the final fabrication and placement in the car can be seen.
Chapter 5

CVT controls

The CVT software controls will be taken from another project which is running at the same time, within Team Fate. The idea is to partly integrate the existing vehicle controls of the Challenge X car with which they are competing in the competition at the moment. This part, the CVT controls used for this car, should be able to be implemented in Sequoia in order to control the inline CVT while only changing some parameters. Because of the time required to complete the fabrication of all the parts for the drive train, there was very little time remaining to fully develop the complete controls system and integrate it into Sequoia. That’s why no real controls have yet been integrated. Also we are not yet sure that it can be done this way, because the vehicle controls of Sequoia, have been done differently than in the Challenge X car. Since both controls are written in C code, it is expected that the CVT controls can be integrated into the vehicle controls of Sequoia. Some software problems will likely have to be overcome. The strategy behind the controls used can be explained briefly, so there is an idea of what still has to be done to complete this part of the project.

5.1 Control strategy

The general control strategy of the Challenge X car is outlined in report [Wil04], which is similar to what will go into Sequoia. A continuously variable transmission is comparable to a transmission with infinite gears. According to the report, the equation governing a powertrain that incorporates a CVT is:

\[ \alpha_{driveshaft} = \frac{-\dot{R}I_e \omega_e + T_e R - T_{losses} - T_{drag}}{I_e R^2 + I_{driveshaft}} \]  

(5.1)

in which:
\[ \alpha = \text{angular acceleration} \]
\[ \omega = \text{angular velocity} \]
\[ R = \text{transmission ratio} \]
\[ I = \text{polar moment of inertia} \]
\[ T_e = \text{torque from the engine} \]
\[ T_{\text{drag}} = \text{vehicle drag transferred to driveshaft} \]
\[ T_{\text{losses}} = \text{losses including all driveline losses} \]

The subscription \( e \) refers to the input shaft of the CVT, while losses, drag and driveshaft correspond to variables reflecting the state of the output shaft of the CVT.

In 5.1 it can be seen that a negative torque is produced, that is proportional to the shift rate, \( \dot{R} \). This implies that for rapid shift rates, the vehicle may momentarily decelerate when the operator requests acceleration. In a hybrid powertrain this effect can be compensated for, by using the electric motor to accommodate transients and CVT dynamics.

In Figure 5.1 it is shown how the CVT will interface with the associated onboard control hardware. This part needs to be implemented into Sequoia, which could cause some integration problems with the old vehicle controller, as mentioned before.

To operate the vehicle most efficiently, the CVT ratio rate of change and clamping pressure are commanded by the Powertrain Control Module (PCM). The accelerator command is translated to torque required from the CVT. From the CVT equations and models, a clamping pressure is determined and sent to the CVT. Once the clamping pressure has been achieved, the CVT controller sends this information to the PCM, which then commands the Internal Combustion Engine (ICE) or Electric Motor (EM) to provide that torque.

### 5.2 Development of the plant model of the inline CVT

The first step to develop the plant model of the inline CVT is to research the components and those component dynamics that make up the transmission. The inline CVT is a pulley type transmission in which hydraulic pressures are applied to the primary, the intermediate axle and secondary pulleys changing their displacement. The primary pulley receives input torques and transmits that torque through the drive chains and the intermediate axle to the secondary pulley and then to the driveshaft. The difference in pulley displacement results in a specific gear ratio that transmits input torque to the driveshaft. There has to be a displacement sensor installed on the primary or secondary pulley, to measure the transmission ratio. Within report [Oud05] the relation between the displacement (stroke) and the transmission ratio for the inline CVT is explained.
Figure 5.1: The figure shows how the CVT will interface with the onboard controls hardware
The second step to develop the plant model of the inline CVT is to determine the relationship between the applied voltages to the servomotors and the developed pressures by the pumps. This is also outlined within report [Wil04] where this is done for a single chain CVT. For the inline CVT this should be the same, because the higher pressures necessary for controlling an inline CVT, are created by the gear ratio between the servomotors and primary and secondary pumps. The pressures go about two times as high as a single chain CVT and the applied gear ratio was 2:1 according to report [Sch05].

As can be read in report [Wil04], there has already been made a controller in Simulink. Herein the inputs received by the model are the voltages to be applied to the pumps. The pumps develop pressure which is then applied to the pulleys and on their turn, the pulleys displace under this pressure and a ratio is created. The values for ratio and clamping pressure are returned to the CVT controller for control feedback.

Inputs are:

\[
\begin{align*}
R_{\text{dotmax}} & \quad \text{Maximum rate of change of ratio} \\
R_{\text{desired}} & \quad \text{Ratio commanded} \\
R_{\text{clamping}} & \quad \text{Clamping pressure commanded}
\end{align*}
\]

Sensors are:

\[
\begin{align*}
W_m & \quad \text{Speed of the input shaft} \\
W_{ds} & \quad \text{Speed of the drive shaft} \\
W_{ia} & \quad \text{Speed of the intermediate axle} \\
R_{\text{sensor}} & \quad \text{Position sensor of primary pulley} \\
P_{\text{change}} & \quad \text{Pressure in ratio circuit} \\
P_{\text{clamping}} & \quad \text{Clamping pressure}
\end{align*}
\]

Outputs are:

\[
\begin{align*}
V_1 & \quad \text{Voltage signal to servo that controls the ratio pressure} \\
V_2 & \quad \text{Voltage signal to servo that controls the clamping pressure}
\end{align*}
\]

And as can be seen in comparison to the controller already built, the extra sensor which has to be integrated in the controller, is the \( W_{ia} \)

A ratio request, a rate of change of ratio, and necessary clamping pressure must be received from the PCM in order to control the transmission. To handle the applied torque, the transmission must provide pressure to the pulleys. Once the necessary pressure is developed, the PCM commands torque form the EM and/or ICE and then sends a ratio and maximum rate of change of ratio to the transmission. The maximum rate of change of ratio is dependent on the input torques to the transmission and the vehicle speed. Therefore it is supplied as
a variable calculated by the PCM for monitoring of the EM, ICE and vehicle speed.

The primary pulley position as well as input and output shaft speed sensors are used to calculate the state of the current transmission ratio both geometrically and via angular velocity. The CVT controller constantly compares these ratios with the ratio requested as it is making the ratio change. The controller considers these values and applies a voltage to the servo pumps to make the necessary changes. The voltage supplied to the amplifier circuit is a signal between -5 volts and 5 volts, the higher the value the faster the servomotor pumps fluid through the line.

Because there is not yet implemented a displacement sensor at the primary pulley the transmission ratio can not yet be determined geometrically. The question is if there can be found a proper displacement sensor which fits in the current configuration of the drive line, or if the ratio will only be calculated through the angular velocity? We have previously searched for an appropriate inductive proximity sensor, but we have not yet found one that is appropriate for measuring the displacement of the primary pulley.

Next steps to take in the development of the plant and controller models are the use of individually sampled inputs and to compare the results. Furthermore it is still necessary to verify simulation of the plant model and controller model in a loop. Then the entire CVT model can be used to operate the vehicle on a dynamometer stand.

5.3 Possible problems

Some problems that need to be considered are the following. First of all the sensor signals from the throttle pedal and brake pedal are most likely in a higher voltage range than the range which is used for this controller. A voltage divider needs to be integrated into the wiring towards the PCM. The integration problems of this controller into the vehicle controls of Sequoia are not yet known.
Chapter 6

Vehicle testing and completing the powertrain

At the end of this internship the building of the complete drive train was not yet accomplished completely. Some of the vehicle components still need to be fabricated or adjusted at the time this report was written. Also due to the fact that the controls were not yet implemented and that this would take a certain amount of time, real vehicle testing did not yet take place.

6.1 Pressurizing hydraulic system of inline CVT

One thing we tested was the pressurizing of the transmission within the car. By putting an extra pump in between the supply line of the primary pump, and operate this pump with a hand drill, we were able to build up pressure within the transmission. By testing it for a short time, we already had 100 bar (1500 Psi) reading from the optical gauges. This is the operating pressure of the inline CVT, according to [Broj]. In Figure 6.1 it can be seen how the testing was set up and during the test no serious leaking appeared.

6.2 Completing the powertrain

The set up for complete vehicle testing is now done by putting the complete car on the dynamo stand in the lab and try to get it operating. Before this can be done, certain things have to be taken care of.

6.2.1 Exhaust adjustment

Due to the new transmission a new engine exhaust pipe needs to be made, because the old one would not fit anymore. Team Fate members have taken
Figure 6.1: Pressure test setup
this job on them at the end of this internship, because of the time period available for this internship it was not possible to do this anymore.

6.2.2 High voltage box adjustments

The high voltage box which originally was used in Sequoia was located near the transmission. After inspection it was decided to replace the box because of safety reasons. The box was located so close to the transmission, that it would be too dangerous, because of possible motion of the CVT case. This could lead to a broken box or connectors, which were located on that side. The high voltage box also needs to be adjusted for some connectors which are necessary for operating the CVT. At the end of this internship, also this has been taken over by members of Team Fate. There is the possibility of completely rebuilding a new high voltage box and locating it at a better position within the car.

6.2.3 CVT controls wiring

There was no time available to complete the wiring of the controls of the new transmission. So this is another thing which first has to be done before real vehicle testing can take place. A thing which has to be taken into account is the operating voltage of the controls. Mototron uses sensor signals which operates from -5 to 5 volts and not from 0 to 10 volts.
Conclusion and Recommendations

In conclusion it can be said that the design brackets are useful for demonstration of the inline CVT in Sequoia, but can be optimized for weight reduction at a later date. It can also be concluded that the hydraulic system to operate the inline CVT is functioning. Furthermore, the use of a heatsink for the amplifiers is not explicitly necessary due to an overkill towards cooling, but provides a safe design. Regarding the controls for the CVT, it is not yet proven that this will work, since there is not yet an example shown. However, significant amounts of work has already been completed by Team Fate members to finish this job in the near future.

Now having nearly completed this internship, it still can not yet be concluded that there is proof of ability to drive Sequoia with an inline CVT. But seeing the progress made, it can be concluded that it is not far away in the future to accomplish this. The problems to overcome are of a level which can be solved, so the car will be operational. Some of the problems still to overcome were outlined in 6.
Appendix A

Technical drawings brackets
A.1 CVT slip clutch bracket
A.2 CVT chassis bracket

Figure A.1: The total assembly of the chassis bracket
Little bracket for on the Chassie bracket Aluminum
A.2.1 Small CVT bracket
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B.1 Electric motor CVT shaft
B.2 CVT slip clutch shaft
B.3 Slip clutch drive shaft bracket
B.4 Spline specifications

B.4.1 Spline specification Electric motor

Figure B.1: Spline specification Elektromotor
B.4.2 Spline specification BorgWarner slip clutch

Figure B.2: Spline specification BorgWarner slip clutch
Appendix C

Amplifier specifications

**B30A40 Series**

**DESCRIPTION:** The B30A40 Series PWM tacho amplifiers are designed to drive brushless DC motors at a high switching frequency. They are fully protected against over-voltage, under-voltage, over-current, over-heating and shortcircuits. All models interface with digital controllers or can be used as stand-alone drivers. They require only a single unregulated DC power supply. A simple red/green LED indicates operating status. Loop gain, current limit, input gain and offset can be adjusted using 144-pin potentiometers. The offset adjusting potentiometer can also be used as an onboard input signal for testing purposes when SW=1. (CE,PE,SW) is CN.

**SPECIFICATIONS:**

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<th>MODEL</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>POWER STAGE SPECIFICATIONS</strong></td>
<td>B30A40</td>
</tr>
<tr>
<td>DC SUPPLY VOLTAGE</td>
<td>0.3 - 400 VDC</td>
</tr>
<tr>
<td>PEAK CURRENT (2 sec max., internally limited)</td>
<td>≤ 30 A</td>
</tr>
<tr>
<td>MAXIMUM CONTINUOUS CURRENT (internally limited)</td>
<td>≤ 15 A</td>
</tr>
<tr>
<td>MINIMUM LOAD INDUCTANCE*</td>
<td>600 µH</td>
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<tr>
<td>SWITCHING FREQUENCY</td>
<td>20 kHz ± 15 %</td>
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<tr>
<td>HEATSINK BASE TEMPERATURE RANGE</td>
<td>0° to 48°C, disables if &gt; 55°C</td>
</tr>
<tr>
<td>POWER DISSIPATION AT CONT. CURRENT</td>
<td>300 W</td>
</tr>
<tr>
<td>OVER-VOLTAGE (SHUT-DOWN; self reset)</td>
<td>425 V nominal</td>
</tr>
<tr>
<td>BANDWIDTH (load dependent)</td>
<td>2.5 kHz</td>
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**MECHANICAL SPECIFICATIONS**

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<tr>
<th>SPECIFICATION</th>
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<tr>
<td>POWER CONNECTOR: P1, P2</td>
<td>Screw terminals</td>
</tr>
<tr>
<td>SIGNAL CONNECTORS: P1, P2</td>
<td>AMP 749100-6, AMP 749101-6</td>
</tr>
<tr>
<td></td>
<td>P1 is a 25-pin high density female D-sub connector and P2 is a 16-pin high density female D-sub connector</td>
</tr>
<tr>
<td>SIZE</td>
<td>8.00 x 5.02 x 1.90 inches</td>
</tr>
<tr>
<td>WEIGHT</td>
<td>2.12 lbs</td>
</tr>
</tbody>
</table>

* Low inductance motors require external inductors.
Appendix D

Contact information

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Gear Chain Industrial B.V. (CVT chain)  
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Phone: 0031 40 2833763/ 0031 40 2906801  
Email: gci@wxs.nl
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Acknowledgement

Initially my internship would focus on the controlling of the Inline CVT and determine the efficiency. Unfortunately at the beginning of my internship the test rig was not operational and the priority within the Hybrid Electric Vehicle Center, was to finish the implementation of the Inline CVT into Sequoia, a Chevrolet Suburban Truck. This meant, that I had to design and fabricate every part necessary to accomplish this. And afterwards the vehicle would be controlled at a dynamo stand in the lab, to make it completely operational for demonstration. Instead of going into the controls, my task was more designing and fabricating. This was still very enjoyable and educational to me. Unfortunately I was not able to get it all done in time to see Sequoia running through the streets of Davis, because of the amount of work it all was to do this.

I would like to thank Prof. Frank for giving me the opportunity to work at the HEV Center at UCDavis, within Team Fate and for the support he gave to me within my assignment. Also I would like to thank my supervisor at the TU/e, dr. Veenhuizen, for letting me go to UCDavis and his advisement towards my updates. Furthermore I would like to thank the team leaders and members of Team Fate and my colleges within the MAE department, who worked with me and helped me out with this project. Also thanks to Leo and Mike from the Workshop at the MAE department, for helping me and advising me with all the practical difficulties I had to overcome. And thanks to Tom "the welding guy", for helping me out with the aluminum welding!