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Modeling of A Continuously Variable Transmission and Clutching of A Snowmobile

Camerin Michael Seigars
University of Maine

MODELING OF A CONTINUOUSLY VARIABLE TRANSMISSION AND CLUTCHING OF A SNOWMOBILE

by

Camerin Michael Seigars

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Advisory Committee:

Michael L. Peterson, Professor of Mechanical Engineering, Advisor

Senthil Vel, Professor of Mechanical Engineering

Zhihe Jin, Associate Professor of Mechanical Engineering

Michael T. Boyle, Associate Professor of Mechanical Engineering

François G. Amar, Professor of Chemistry and Dean Honors College

Abstract

This thesis describes the conceptual operation of a continuously variable transmission (CVT) and develops a model of a CVT system. The purpose is to form a framework for understanding how CVTs work, what factors go into their design, why they are used on almost all modern snowmobiles, and how they can be tuned for better performance. By developing a model using rigid body analysis, computer modeling, and a list of structured equations, a CVT can be tuned more efficiently. The model is used to calculate values difficult or tedious to evaluate by hand with visual aide for clearer understanding of the motion. By combing the model results with a series of equations, that prescribe the dynamics of the CVT a methodological framework is developed that can be used to tune a snowmobile to save time, money, and tune more efficiently.

Dedication

I want to dedicate this Honors Thesis to two loving parents who are always there for me. Michael and Danielle Seigars, a son couldn't be any more blessed than to have you in his life, I love you both. I could not have been as successful in school and life without your guidance, and especially your love. The two of you remind me every day why I am so lucky to have you in my life. Thank you for being who you are, I wouldn't have changed a thing growing up with the two of you. I hope I can make you proud and repay your innumerable efforts raising me.

Love your son,

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Professor of Mechanical Engineering, University of Maine
Honors Thesis Advisor and SAE Clean Snowmobile Club Advisor

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Committee Member, Professor of Mechanical Engineering

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Committee Member, Professor of Mechanical Engineering

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Committee Member, Professor of Mechanical Engineering

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Committee Member, Dean of Honors College and Professor of Chemistry

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Introduction

1.1. Statement of Need

In the history of snowmobiles, it has been a continuous struggle to efficiently transfer power from the engine shaft to the track. The difficulty of this problem stems not only from the design constraints of a snowmobile due to size, weight, and power, but also from the diverse series of conditions it might experience from hard packed snow in a straight line to deep powder in sharp turns, and a variety of other conditions. How you best help the snowmobile deal with these conditions is by having a properly designed and tuned continuously variable transmission (CVT). The University of Maine Society of Automotive Engineers (SAE) Clean Snowmobile Project has an ongoing project to build a compressed natural gas (CNG) snowmobile to race in the SAE Clean Snowmobile Challenge. This presents a unique problem when it comes to the CVT because the conversion to CNG changes the behavior of the engine shifting the torque curve being fed to the CVT from the engine shaft, making it necessary to retune the clutch for new input parameters. In addition, the University purchased an Arctic Cat XF 1100 SnoPro for a snowmobile which is designed as trail sled, which has a completely different set of clutching objectives than a racing snowmobile, making it even more critical to retune the CVT. The ability to tune the CVT efficiently and accurately will be an ongoing need as teams from year to year change the characteristics of the snowmobile, making it necessary to retune the CVT.

1.2. Continuously Variable Transmissions and Clutching

1.2.1. Practical Concept

The concept of a CVT is complicated and includes a variety of complex engineering principles and moving parts. The details of how CVTs work will be covered in section 2.1.1. but in short the practical purpose of the CVT is to transfer the power of the engine from the shaft to the differential so it can be transferred to the track to move the snowmobile. The CVT does this while allowing for different gear ratios to be achieved by simply changing the diameter of the driving and driven clutch to achieved the desired results. This gives the sled the ability to idle without the clutch engaged and allows for the snowmobile to be tuned for different applications through the proper tuning of the CVT.

1.2.2. Importance of Computer Modelling

In order to properly tune a CVT for a specific purpose, several factors need to be taken into account which create a uniquely complicated problem traditionally solved by trial and error by an experienced mechanic who knows what to look for and how to change it in order to tune the CVT. With computer modeling to analyze the rigid body dynamics and the properties of certain components such as mass moment of inertia, tuning a CVT could be done with less trial and error because the factors that need to be changed can be narrowed down or even determined in some cases through the use of the models the and application of other known equations. This will save time, money, and effort in the process of producing a properly tuned CVT for the SAE Clean Snowmobile Project.

1.3. Objectives

The primary goal of this thesis is to define how CVTs work and how to properly tune them with a framework that uses a computer model and a series of defined equations for analysis. The list below outlines this goal with more detail.

Objectives:

- State the purpose of CVTs in snowmobiles.
- State the function and design of modern CVTs.
- Create a comprehensive explanation of how CVTs work theoretically and practically.
- Identify the type of CVT design used on the CNG Snowmobile.
- Identify important equations for CVT analysis.
- Create SolidWorks models of select components to find mass moments of inertia.
- Create MSC Adams models of the CVT in multiple stages of shifting.
- Create a procedure for how to tune the CVT using the computer program and identified equations for a specific purpose.
- Run the program to show how it works for assisting in tuning.
- Outline future work in relation to the CVT for the SAE Clean Snowmobile Project.

Materials

2.1. Continuously Variable Transmissions and Clutching

The CVT system used in vehicles is an important mechanism that allows the vehicle to achieve a wider band of speeds with the same engine by utilizing a series of gears to achieve variable speeds of the entire vehicle while maintaining the normal band of engine speed. Snowmobiles use CVTs specifically for the transmission of power to achieve this goal for a variety of reasons that will be explain in this section.

2.1.1. Concept of a Continuously Variable Transmission and Clutch

A continuously variable transmission (CVT) is, as described above, the system for transmitting power through a series of seamless gears from the shaft of the engine to the differential, which gears the power to the track to move the snowmobile. Figure 1 to the right shows a top-down view of a standard CVT design. It can be seen that the general layout of the system involves two pulleys, or clutches, connected by a rubber belt. The primary pulley, known as the driver pulley, is the clutch which is

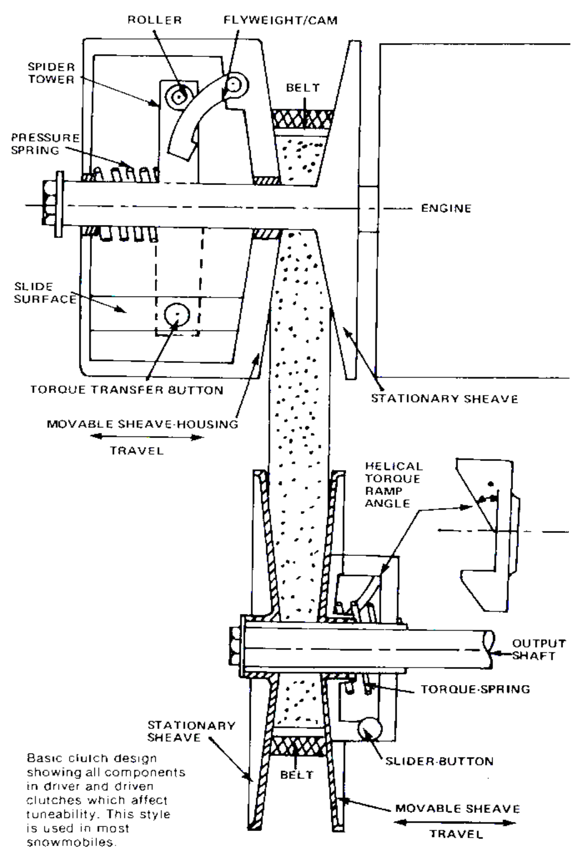

Figure 1: Top-down view of a standard CVT layout [1]

connected directly to the output shaft of the engine as seen in Figure 1. The belt is wrapped around the two sheaves on the driving clutch on one side and around the two sheaves of the secondary, or driven, clutch on the other side. This allows for the power of the engine to be transferred through torque in the shaft to the clutch or driver pulley which moves the belt, thereby driving the driven pulley using a seamless series of gears which transmits the torque into the output shaft shown, which goes to the differential or gearbox which transmits power to the track to propel the snowmobile.

The CVT travels through a seamless series of gears while transmitting power from the engine to the output shaft. This is achieved through changing the diameter of the pulleys that the belt experiences while it is being moved. This is accomplished by moving the outer sheaves on both the primary and secondary pulleys. Each pulley has two sheaves, one moveable sheave and one stationary sheave. The two types of sheaves are labelled in Figure 1 for visual reference.

In the next two subsections, the operation of first the primary pulley, and then the secondary pulley which is used for gearing and to transfer torque into the output shaft will be covered.

Primary (Driver) Clutch

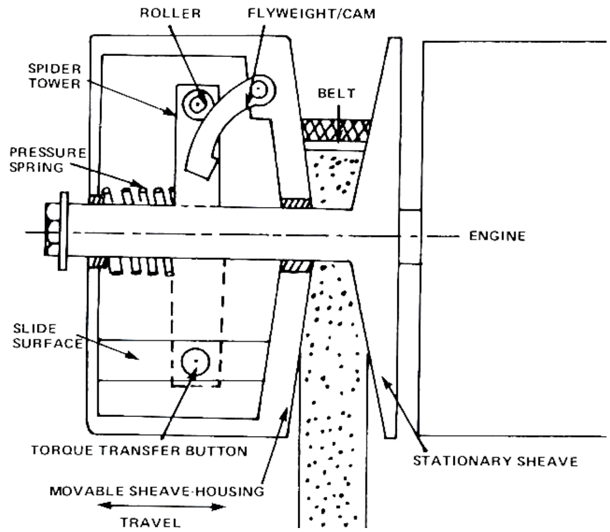

Figure 3: Top-down view of the primary clutch [1]

The main purpose of the primary pulley is to control engine speed and to ensure the engine runs on the power curve (the peak of the power band) throughout the entire shift cycle from lowest gear to highest (over-drive), which is considered the entire shift range of the CVT. Figure 2 on the right shows a blown up portion of Figure 1 which only includes the primary clutch. In this figure one can better see the inner workings of the mechanism. There are several factors that go into the operation of the primary clutch; however, its operation can be broken down into five phases; *free running (disengagement), engagement, running, shifting,* and *over-drive*. The first phase, commonly known as *free running*, is when the engine is running but the primary clutch is not engaged. The purpose of this phase is to allow the engine to run at idle without moving the snowmobile. The system moves into the *engagement* phase when the engine builds up enough RPMs to reach the engagement speed, which is when the flyweight force (centrifugal force) generated by the spinning of the

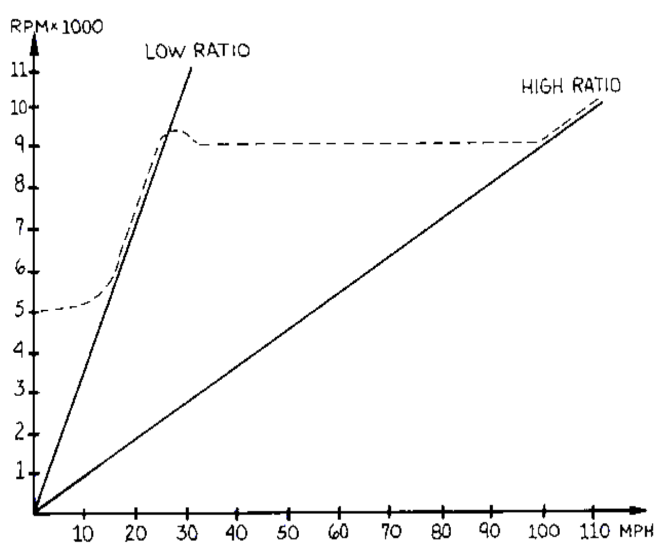

Figure 2: Engine Speed vs. Vehicle Speed for a Theoretical CVT System [1]

clutch is equal to the pretension load of the pressure spring shown in Figure 2. The third phase is the *running* phase which is when the centrifugal force of the flyweights is above the engagement point but the snowmobile hasn't built up enough RPMs to reach the shifting zone yet. Phase four, the *shifting* phase, is a critical part of having a well-tuned clutch. The most effective way to make use of the engine's complete HP and torque is to keep the engine on its power peak for as long as possible. The power peak is the maximum power the engine can produce which is found in the power band, which is a range of RPMs in which the engine is most powerful. The power peak is found inside the power band. Therefore, to make the most out of the snowmobile, the shifting phase should occur as close to the power peak inside of the power band as it is possible. This keeps the engine at its maximum power throughout the entire shift cycle of the sled making the most of its power. That means that *shifting* should occur at a constant RPM which corresponds to the power peak. Once shifting is complete, if the engine continues to build RPMs, it is off the power peak and out of the *shifting* phase and into what is called *over-drive*. *Over-drive* occurs at the CVTs maximum high end gear ratio. The danger of running a sled at *over-drive* is that as the engine builds RPMs past its power peak, it not only falls off the peak, but extremely quickly outside the power band and the engine begins to work harder with more RPMs but is generating less power. To assist in visualizing these phases Figure 3 above shows a plot of RPM x 1000 vs. MPH for a theoretical engine with a CVT. It is seen that in Figure 3, there are two solid lines and one dotted line. The first solid line, labelled low ratio, shows the points where the minimum ratio achievable by the CVT occur. The second solid line shows all the points where the maximum achievable ratio for the CVT occurs, also known as the high ratio, as it is

indicated. The dotted line represents the actual engine speed being experienced by the theoretical snowmobile in question. It can be seen that for this theoretical snowmobile, the *free running* phase occurs until 5000 RPMs when the snowmobile begins to accelerate and has a velocity. There is a brief period of time after the *engagement* phase at 5000 RPMs before the snowmobile is fully engaged and at the low ratio. Now the snowmobile is in the *running* phase as it builds RPMs and climbs up the low ratio line, indicating it is not shifting but simply building up more engine speed until the centrifugal forces being exerted by the flyweights is strong enough to enter the *shifting phase*, which can be seen as the straight dotted line as it travels from the low ratio line to the high ratio line. This indicated the snowmobile CVT moved through its entire shift cycle at a constant RPM, presumably the power peak if tuned correctly, which is the desired behavior. The last bit of the dotted line can be seen going up in RPMs along the high ratio line, this is what was described earlier as the *over-drive* phase.

Of the many components that make up the CVT system it is important to understand the function of the vast majority of them in order to best understand the system and how it functions. The following is a breakdown of some of the components in the driver clutch.

Driver Clutch Components:

- Stationary & Movable Sheave
- Pressure Spring
- Flyweights
- Sliding Buttons

The driver components work together to provide the engagement speed and dictate the speed of the engine. As discussed above, the primary role of the driver clutch is to control the RPMs of the engine through the entire shift range to provide the desired effect.

Two of the most important things in a CVT that are a part of the primary clutch were introduced above, the pressure spring and the flyweight system. More about these two components will be discussed in section 2.2.3. on how to tune a CVT since these are two of the major components for tuning the transmission system.

Secondary (Driven) Clutch

The main purpose of the secondary clutch is to deliver the power transmitted through the belt from the primary clutch to the output shaft as efficiently as possible. The primary clutch controls engine speed, but the secondary clutch dictates the efficiency of the system. In order to

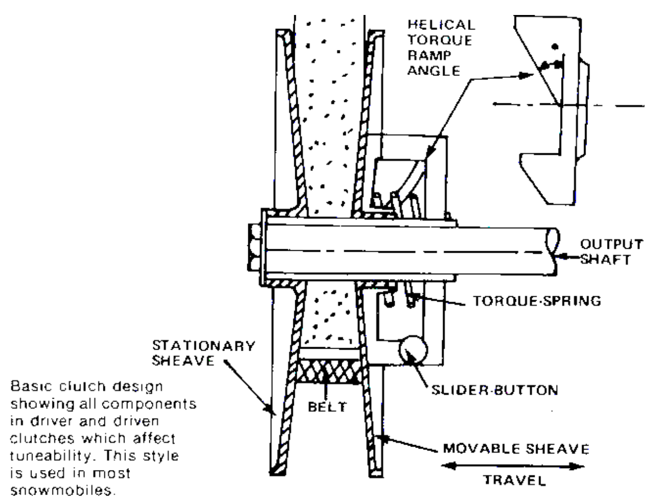

Figure 4: Top-down view of a secondary clutch [1]

transfer the power from the belt to the secondary clutch, there must be enough side pressure on the belt in any given shift ratio. This prevents the belt from slipping on the sheaves, which is a large source of inefficiency and belt wear. If the belt is slipping on

the sheaves, it is not effectively transmitting power to the clutch because of friction losses.

Horsepower is found by taking the torque and multiplying it by the speed of the pulley. This is shown as Equation 1 [1] below.

$$HP = T * V \tag{1}$$

HP is the horsepower, T is torque being applied, and V is the speed of the pulley.

Figure 4 on the previous page shows an enlarged portion of Figure 1 which highlights only the secondary clutch portion of the CVT. It can be seen that the secondary clutch transmits the engine horsepower from the torque of the belt into the output shaft which leads to the engine gearbox. It can also be seen that, like the primary clutch, the secondary clutch has two sheaves, one stationary and one movable like is shown in Figure 4. The movable sheave is controlled by the torque spring which can be used for tuning. The biggest obstacle to overcome with the secondary clutch is friction losses which can significantly lower the efficiency of the CVT.

There are many components that make the clutches in a CVT, and they differ from the driver to the driven. The following is a breakdown of some of the parts that make up the driven clutch similar to the list created previously for the driver clutch.

Driven Clutch Components:

- Stationary & Movable Sheave
- Torque Feedback Ramp
- Torsion Spring
- Sliding Buttons

The driven clutch needs to use those components to provide different things at different points in the shift cycle. At low ratio it needs to provide high side force on the sheave, but this force diminishes as the clutch shifts up because less side force is necessary to keep the belt from slipping at higher ratios. A general rule of thumb used by mechanics is that you need about twice as much belt pressure in low ratio than in over-drive if your CVT has an overall shift ratio of four. For a more accurate way to find the theoretical side force produced by the cam for any shift ratio Equation 2 [1] can be used.

$$SF = \frac{T*SR*12}{2*r*\tan\alpha} \tag{2}$$

SF is the side force, T is the torque in ft-lb, SR is the shift ratio, r is the ramp radius in inches, and α is the ramp angle. These are shown on Figure 5 to the right for assistance in visualization. In reality the side force involves other factors such as any side force generated by the torsional spring, the pretension being exerted by the torque load from the spring, sheave angle being experienced, and efficiency of the system which includes losses due to the amount of friction that is taking place.

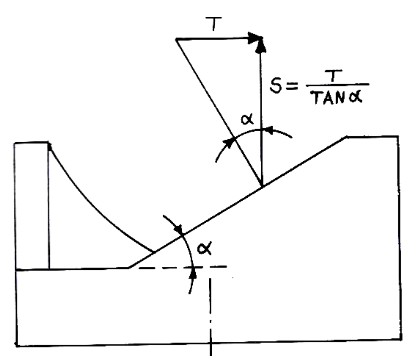

Figure 5: Visual Aide for Equation 2 [1]

Two critical components of the secondary clutch are the torque feedback ramp and pretension spring which were mentioned above. Figure 7 shows an example of an Arctic Cat pretension torsional spring that would be used in a secondary clutch. It can be seen the spring ends are turned to be nearly parallel with the spring travel path to hold the spring in place inside the clutch. This is necessary because the preload position of the spring is adjustable to change the spring tension which affects both initial belt pressure and the back shifting of the snowmobile. The higher the tension is made, the faster the CVT will backshift, and higher pretension is required for higher horsepower engines. Now that the torsional spring has been discussed it is important to introduce the specifics of the torque feedback ramp. Most torque ramps on modern snowmobiles, like the one shown in Figure 6, consist of three ramps and series of holes used to adjust the pretension of the torsional spring discussed above and shown in Figure 7. The ramps on the cam move against the sliding buttons on the movable sheave. This is how the amount of side force is controlled. Not only does the ramp angle have an effect on the side force, but the radius they work at does as well. As Equation 2 shows, the smaller the ramp angles, the smaller the side force

Figure 7: Torque Feedback Ramp [1]

Figure 6: Arctic Cat Pretension Spring [7]

will be that is produced.

There is a second style of driven clutch that is often times used by Arctic Cat which will be briefly introduced since the CNG snowmobile has this style of secondary clutch. It is called a reverse cam secondary. The biggest difference between this and normal secondary clutch is that on a reverse cam, the cam is on the outside of the pulley near the outside of the chassis. There is no huge operational advantage to this design, however it is easier to access since the cam and spring assembly is on the outside in the open as opposed to near the engine where it is more difficult to access.

Relationships Between the Clutches

When looking at the concept of CVTs there are many relationships that are essential to not only understanding how they work, but being able to analyze them for tuning purposes. The following series of concepts and equations highlight the most important engineering aspects of the CVT.

One of the most important relationships for a CVT is the effect of the ratio between the two clutches. The ratio between the two clutches dictates how the belt transfers speed from the primary clutch to the secondary clutch. The ratio dictates the torque transfer coefficient, and is traditionally given the following form *Driven:Driver,* where the *Driven* represents the multiplier for the torque for the driven or secondary pulley, and the

Driver represents the multiplier for the driver or primary pulley. There are three examples given with Figures 8 through 10 to illustrate each of the examples.

For example, a 4:1 ratio means the driven pulley is experiencing four times the torque of the driver pulley, this is considered a low end ratio because the diameter of the secondary is larger than the diameter of the primary. This is shown by Figure 8, which illustrates a side view of the primary and secondary clutches. The belt is shown inside of the primary clutch because the belt has not moved out on the sheaves yet, whereas for clarity, the secondary shows the belt wrapping around the outside which just shows the belts current position on the sheaves rather than the outside diameter of the pulley itself.

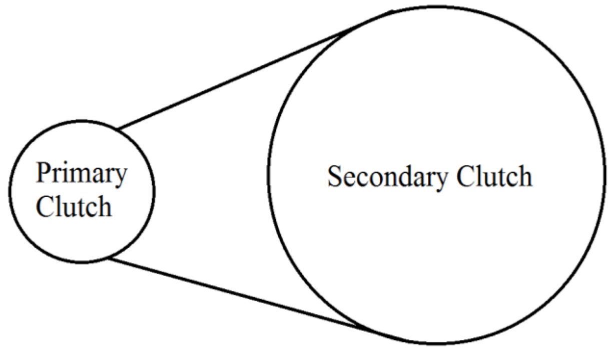

Figure 8: Side View of a CVT in Low-Ratio

As the CVT shifts out from low ratio, the belt will progress further out the sheaves on the primary clutch and further in the sheaves on the secondary clutch. This change in diameter ratio between the driven and driver side seamlessly shifts the sled. Figure 9 shows another example of a position in the shift range; it shows a 1:1 ratio,

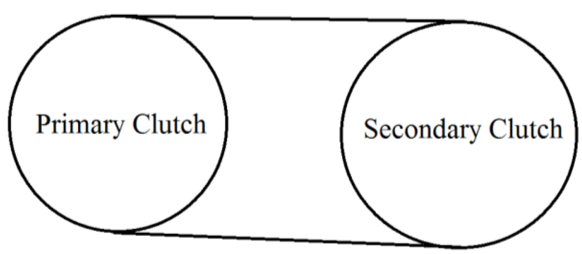

Figure 9: Side View of a CVT in Mid Ratio

which is the only time when the torque of the input shaft from the engine to the driver clutch equals the torque of the driven clutch to the output shaft.

Imagine if the ratio is 0.5:1 then the clutch is in or near over-drive and the driven is experiencing one half of the torque of the primary, meaning the diameter of the driven pulley is smaller than the diameter of the driver pulley, this is considered a high ratio. Figure 10 gives an example of the clutches in a high ratio position.

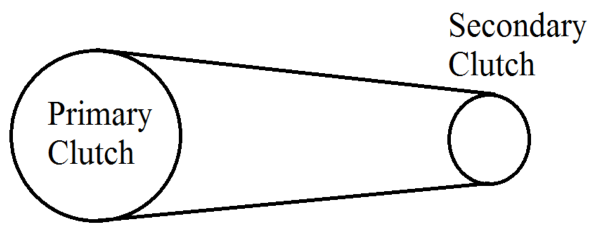

Figure 10: Side View of a CVT in High Ratio

The gearing relationship between the primary clutch and secondary clutch connected by the drive belt is the cornerstone of the rigid body motion of a CVT. More about the rigid body motion of this system will be discussed in section 3 which discusses rigid body analysis of a CVT system.

2.1.2. Why Snowmobiles Use CVTs

The purpose of the CVT on a snowmobile is to transmit power from the engine shaft to the differential which transmits the power to the track, essentially the ground, to propel the snowmobile forward. Why do snowmobiles choose CVTs instead a direct drive or a

gearbox like a car uses? There are several reasons for this choice, but first it is best to elaborate on some of the conditions a snowmobile will experience during operation then a CVT can be compared to a 4-Speed Gearbox to weigh the pros and cons of the two against each other.

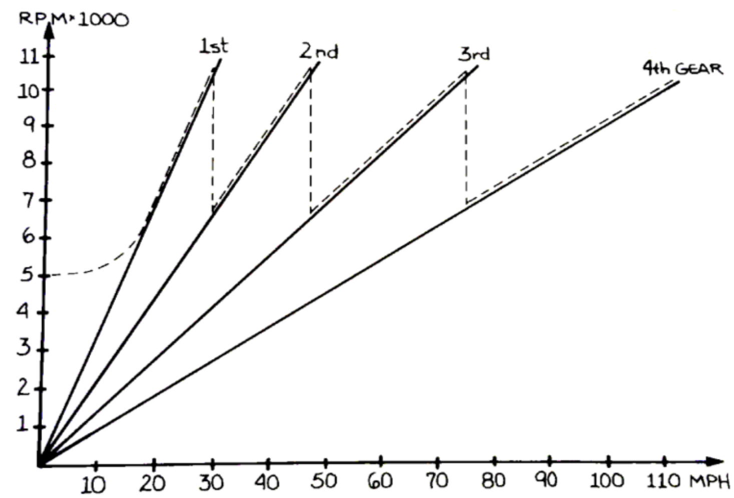

Figure 11: 4-Speed Gearbox Transmission Engine Speed vs. Vehicle Speed [1]

During operation a snowmobile undergoes a uniquely wide array of conditions compared to a traditional automobile. The best way to explain this is to take a couple of quotes from the *History* chapter of *Olav Aaen's Clutch Tuning Handbook* [1]. The first puts into perspective how some designers in industry view the abilities of the CVT, "A designer's dream: 'Smooth uninterrupted power delivery automatically adjusting itself to any load conditions.' A transmission like this would be a dream come true on most vehicles. It has already been a reality on snowmobiles for over 60 years." [1]. That really puts into perspective from someone who has worked with CVT systems on snowmobiles just how perfect the CVT is for snowmobile applications. The next quote concisely sums up how the CVT reacts to snowmobile load conditions, "Quick back shifting delivering lots of torque when you hit the soft powder, hard acceleration out

of tight corners combined with strong top end performance and a season on the same belt are now parameters almost taken for granted.” [1].

Now that some of the common conditions a snowmobile has to endure have been introduced along with a perspective on the ability of CVTs, the pros and cons of a CVT vs a traditional 4-Speed Gearbox can be laid out. As was introduced in section 2.1.1. and shown in Figure 3 a CVT works by building engine speed after engagement up the low ratio line until it ideally reaches the power peak of the engine and then makes a smooth shift through an infinite number of gear ratios from low too high in a straight line across the Engine Speed (RPMs) vs Snowmobile Speed (MPH), i.e, keeping the engine on the power peak through the entire shift range to high gear as is shown in Figure 3.This keeps the engine running where it has the most power for the longest, making performance the maximum that is possible after normal efficiency losses. A 4-Speed Gearbox, on the other hand, behaves in a very different manner. In a gearbox there are a series of predefined gears from low to high which are made by different combinations of mating parts inside the gearbox. Figure 11 shows an engine speed vs vehicle speed graph for a standard 4-Speed Gearbox style transmission. Unlike a CVT it can be seen that the gearbox has well defined gear lines for more than just low and high; since this is a 4-Speed transmission, it has four predetermined gears whose ratio line is shown as solid black line on the graph. The dotted line on the graph shows the path the theoretical engine takes as it builds up speed and shifts through the gears. Unlike a CVT the engine spends very little time on the power peak of the engine. In fact, the RPMs stray outside of the power band for much of the engine’s operation. The engine has to climb up first gear

past the power peak so that when it shifts into second and loses RPMs, the engine doesn't bog out or stall. That means not only does the engine have to rev past the power band before shifting, it also has to drop below the power band going into the next gear. This reduces efficiency greatly and also creates a delay between gears where there is no power being transferred through the transmission from the input shaft to the output shaft. This means the machine loses power output to the track during operation to shift and there is a high stress on the transmission and engine when it does engage into the next gear because the system has to shift back into gear against all the resistance of moving. In a gearbox the engine has to undergo this very inefficient process every time it shifts up or down in gears, as can be seen in Figure 11.

Given the conditions that were presented earlier for the operation of a snowmobile, it is an advantage is that a CVT remains on the power peak throughout its operation with the engine and a disadvantage is that the gearbox has to over rev and under rev, missing the power band for much of its operation. This behavior is closely illustrated in Figure 12 which shows a CVT operation as a bold dotted line and a gearbox operation as a lighter dotted line. In addition, it is a pro that the CVT can seamlessly switch between gears without having to drop power to the drive. This is essential for

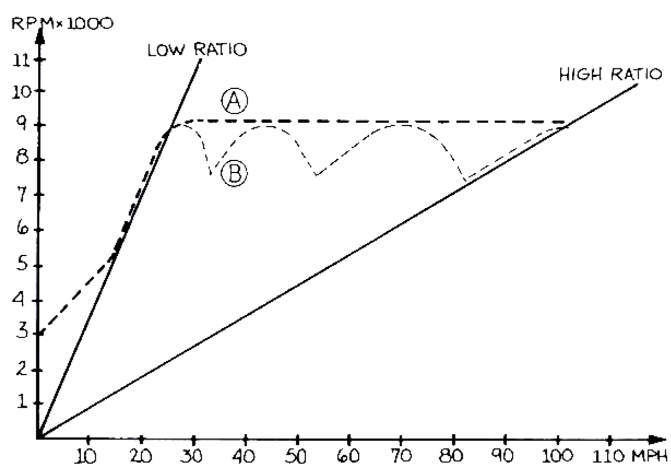

Figure 12: Comparison CVT & 4-Speed Gearbox Engine Speed vs. Vehicle Speed [1]

snowmobile operation where the system needs to quickly upshift as it accelerates on a straight stretch or out a tight corner or for a trail sled when the snowmobile hits deep powder snow and the system suddenly has drastically increased resistance on the track. In these conditions it would be detrimental to have a gearbox style transmission because in powder the inability to stay in the power band and the loss of power when the system would shift is more than enough to get the snowmobile stuck in the snow [1].

That is a concise introduction to why a CVT style transmission is almost exclusively chosen on all snowmobiles. There simply isn't a better way in modern engineering to deliver the seamless shifting under diverse and constantly changing loading conditions, and the clear advantage of being able to tune the CVT to keep the engine on the power peak for the best use of its maximum potential horsepower is without question.

2.2. Tuning a Continuously Variable Transmission

2.2.1. Why is Tuning Important?

When building a snowmobile for any application, it is essential to review the transmission setup. There are very specific parameters for setting up a CVT for specific applications. It is important to tune the CVT on a snowmobile because it can be the source of the most efficiency loss between the engine and the track of the machine, which directly translates to lost power at the track since the CVT is responsible for power transfer and gear ratios of the system. Tuning the system can eliminate almost all of the losses being experienced through the power transfer system. The only losses that cannot

be overcome through tuning are simple friction losses which can be drastically reduced with proper equipment but not eliminated because all moving parts experience friction and unavoidable mechanical losses from the operation of a system.

A simple example of the magnitude of the effect tuning the CVT can have on a snowmobile is a situation often talked about in the SAE Clean Snowmobile meetings where someone used the track dynamometer to measure the output power at the track and tune the CVT to increase efficiency [2]. Before tuning the snowmobile, the track was putting out 60 horsepower (HP) but the engine was putting out 120 HP, this is 50% loss of power from the input shaft of the engine to the track. Since the losses through the gearbox from the CVT output shaft to the track are minimal, most of this loss was coming from the CVT transmission system which was clearly not properly tuned. A significant amount of effort and a lot of trial and error over a large span of time (the traditional way to tune a CVT) was used to tune the CVT resulting in the track experiencing 100 HP at the track which is only a 16.7% loss in power through the system from the engine to the track. That increase of 40 HP is a significant difference, which highlights the extreme importance of properly tuning the CVT system. This is especially important on the SAE Clean Snowmobile project which is designing a more environmentally friendly snowmobile for racing. Any unnecessary loss of power burns more fuel than is necessary and could make the difference in winning or losing the race.

2.2.2. What does Tuning do?

Tuning a CVT allows for greater efficiency of power transfer, more or less back shifting depending on what is needed, control of take-off engine speed, and at what RPMs the CVT shifts and how fast. All of these make for a better running and more fine-tuned snowmobile, which is not only essential for maximized efficiency in regards to reducing excessive emissions and unnecessary power loss from the engine to the track from not utilizing engine power properly

The factors that are effected by the tuning of the snowmobile, i.e, what tuning a CVT does to the system, have been listed in a bulleted list below and will be further discussed in the following section on how to tune the CVT in a snowmobile.

Factors Effected by Tuning:

- Engagement speed of the primary clutch
- How long the engine stays on the power peak or in the power band if staying right on the power peak isn't possible or excessive for desired application
- Pressure spring in primary strength, rate of travel and starting position
- Flyweights properties including overall mass, position of COG, mass moments of inertia
- Reducing unnecessary belt pressure which leads to excessive power loss. This can come from many factors including the belt bending or stretching, and high belt heat

- Choosing the proper torque feedback ramp (correct ramp angles for desired effect)
- Pretension spring in secondary strength, rate of travel, and starting position
- Friction in the secondary. Steps can be taken to slightly reduce friction or to nearly eliminate it depending on cost and performance needs

In addition, one of the cardinal rules of tuning is as follows, “If you want to change engine speed, work on the driving clutch. If you want to improve efficiency or back shifting adjust the driven clutch.” [1].

2.2.3. How do you Tune a Snowmobile CVT?

The traditional method for tuning a CVT is experimental trial and error with a basic knowledge of how changing certain parameters will change the performance aspects of the CVT. This is a time consuming and expensive process for a variety of reasons. Using trial and error can easily lead to buying unnecessary components, making changes that decrease efficiency or bring the engine off the power band, and ruining parts by making improper modifications. As will be explained in section three the purpose of this thesis is to develop a process to making tuning more efficient both fiscally and with time spent. However, this section is going to focus on the process used for tuning a snowmobile and what changing certain factors due to the behavior of the CVT system. How to tune the CVT system is divided into two subsections, the first for how to tune the driver (primary) clutch and the second for how to tune the driven (secondary) clutch. Before the two

subsections are covered, Figure 13 shows the ideal Engine Speed vs. Vehicle Speed graph for a CVT with alphabetical labels for various sections of the figure. Note that this graphic will be referred to throughout this section for clarity and that the note on the figure does not have an accurate figure number for this thesis because it is from the source the figure was taken from, but the note itself still has valuable information in it about the figure.

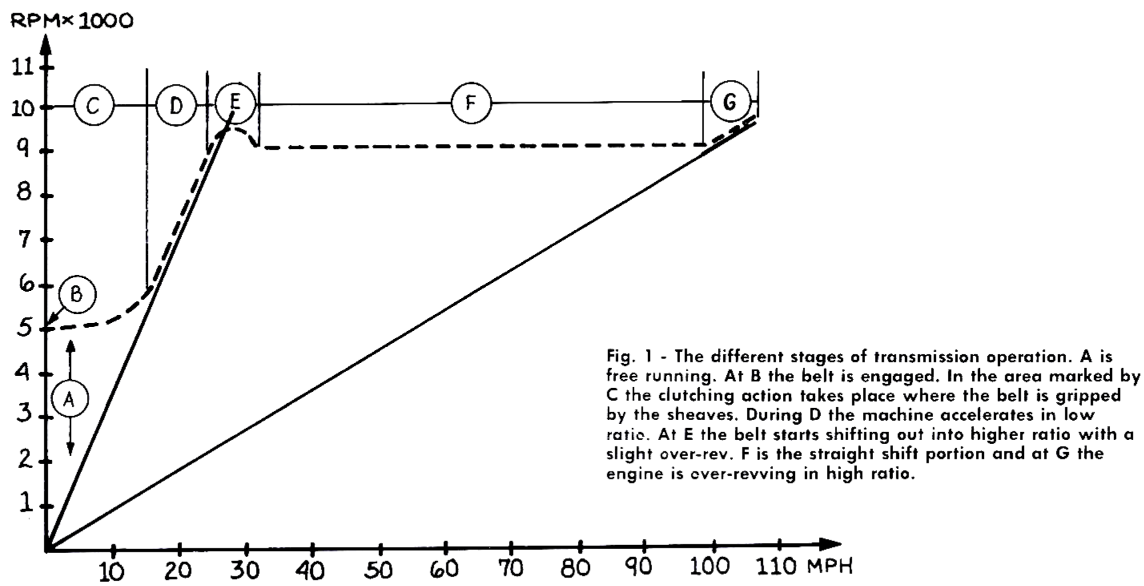

Figure 13: Ideal Engine Speed vs. Vehicle Speed for a CVT with Labels [1]

Tuning the Driver (Primary) Clutch

For the driver, tuning is all about controlling the engine speed properly for the entire shift range. As will be discussed in the next subsection on the driven clutch, that is where efficiency is tuned. The components adjusted in the driver for tuning are flyweights, pressure spring, and cam-mechanisms.

The first phase in the graph shown in Figure 13 is C, which is the clutching phase. This is the phase where the flyweights have generated enough force to overcome the pretension but haven't gotten enough side force to fully engage the belt. Clutching ends when enough side force is built up to fully engage the belt without slipping. This is the part as shown from the beginning of movement until the engine speed dotted line intersects the low ratio line shown as the line that leads to E. In tuning it is critical to match the flyweights and springs so as to minimize slippage which wears on the belt.

One of the most critical aspects of tuning is making sure the shift point, shown as point E on Figure 13 where the ratio moves off the low ratio line and begins to move horizontally on the path shown as F, is at the power peak of the engine or close to it inside the power band. For two-stroke engines especially, but also four-stroke engines, it is imperative that this is tuned correctly because these engines have a very narrow power band where the most HP can be generated by the engine, and that needs to be taken advantage of. A properly tuned CVT will keep the engine running at a constant RPM on the power peak through the entire shift range to best utilize maximum engine HP. The only exception is what is called over-run, which is shown as the slight dip up in the dotted line beneath E. The purpose of over-run is to make up for a known drop in RPMs that is going to occur as the engine begins to shift. So if the engine is slightly over-run in RPMs when it drops the known amount, it will drop down onto the power peak for the rest of operation. The question is how does tuning accomplish these goals? By adjusting two components of the primary clutch, the pressure spring and the flyweights.

Tuning the Pressure Spring

The first part of looking at the pressure spring in the driver is to know how the spring force relates to the belt pressure which is the most important concept in tuning the primary clutch. Equation 3 [1] shows the relationship between belt pressure force (Fbp), flyweight force (Ffw) and spring force from the pressure spring (Fps).

$$Fbp = Ffw - Fps \tag{3}$$

Now that the relationship essential to the primary objective of the driver clutches operation is known, how to best tune the pressure spring can be discussed first, then how to best tune the flyweights will be explored. The pressure spring has two major characteristics that can be changed and one limiting factor that is necessary to keep in mind.

First the limiting factor should be mentioned. This is the length of the spring and space constraints of the cavity it is installed in. This will vary from clutch to clutch and should be observed by the person tuning the snowmobile. Figure 14 shows a spring with common nomenclature that will be used in this section when talking about the spring.

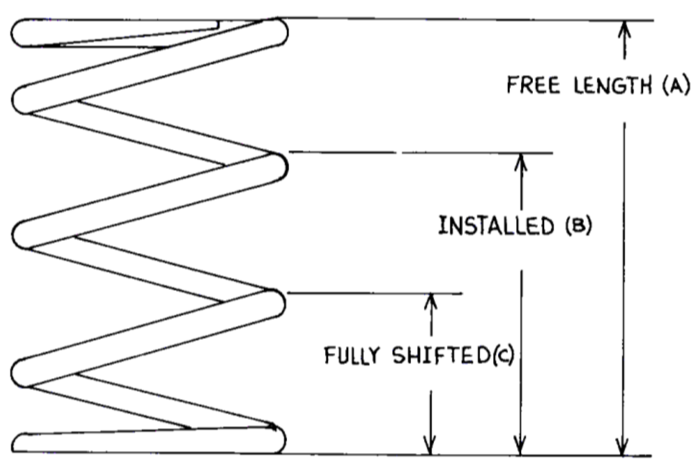

Figure 14: Spring with Common Nomenclature [1]

Now the two design criteria that can be changed on the pressure spring are its rate of travel and pretension. The rate of travel is a physical property of the spring chosen and should be a known value when the spring is purchased. Some manufacturers could refer to the rate of travel as just the rate, and in U.S. Customary units, it has units of pounds per inch (lb/in). When tuning a clutch, it is important to know what the effect of the rate is on the CVT so when tuning it is known whether a faster or slower rate per unit distance is required. The spring rate is usually used to effect shifting to maintain a straight shift from low-ratio to high-ratio. If, when shifting from low to high, the speed of the engine drops off the power peak, this can be corrected with a spring of faster (higher) rate. If the engine gains speed through the shift range, this can be compensated with a slower (smaller) rate. Note that when making decisions during tuning, it is often found that engagement speed is a secondary goal to having a straight shift along the power peak. The second design criteria are the pretension of the spring, which is dictated by the amount of compression the spring undergoes when installed in the clutch. From Figure 14 above, the position before installation would be seen as the free length, whereas the position B, the installed position is the state of the spring at pretension. The final position C (fully shifted) is the state of the spring when the clutch is shifted all the way out. Solving for the force of spring under pretension is easy; simply take the rate of travel and multiply it by the distance the spring is compressed (this is the difference between free length and the installed length). Now that pretension has been defined, during tuning it is important to know what higher and lower pretension does to the CVT. Pretension is used to control the engagement speed of the engine. It does this because the pretension force is

the force the centrifugal force of the flyweights needs to overcome in order to engage the primary clutch. More pretension means a higher engagement speed if flyweights are left the same. On the opposite end, less pretension means a lower engagement speed if flyweights remain the same. Note that shims can be stacked between the spring and the cover in order to increase the pretension by making the difference between the free length and installed length larger. The largest issue with shims is that if too many are installed, the spring may not have enough length to travel through the shifting phase and may bind, causing serious issues in performance.

The other design parameter that can be changed in the primary clutch is the flyweights. In a CVT the force that propels the system during shifting is all generated by the centrifugal force of the flyweights as the primary clutch spins at various angular velocities. To best understand the flyweight system and how to tune it, it’s important to understand centrifugal force which is what acts on the flyweights. Equation 4 [1] below can be used to calculate the centrifugal force Fc.

$$Fc = M * R * V^2 \tag{4}$$

Fc is the centrifugal force, M is the mass, R is the radius, and V is the velocity as shown in the Figure 15 to the right. Based on this formula, three relationships can be easily seen. The first is that centrifugal force increases proportionally with the weight. The second is that the centrifugal force increases proportionally

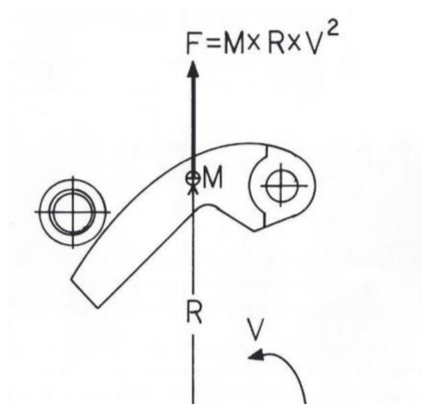

Figure 15: Flyweight Informative Diagram [1]

with radius. The third is that centrifugal force increases with the square of the speed. These relationships are extremely important in understanding and tuning the CVT based on the flyweights.

The first part of the flyweight system that can be used to tune the CVT is the cam curvature. This is the side shown on the top of the cam in Figure 16. The cam curvature can be changed to yield different engagement speeds, more or less aggressive shifting, over revving when shifting out, and smooth or aggressive shift patterns. Note that it is recommended that only small amounts are taken off the cam at a time because as little as twenty-five thousandths of an inch off the cam can change engine speeds by as much as 500 RPMs [1]. When tuning, the cam surface should only be grinded in the engagement area and the shift point area unless there are no other options for tuning. A typical flyweight is shown as Figure 16 which outlines the three areas that are of concern during tuning. It shows the engagement phase area as A, the transition phase area as B and the shift curve phase area as C [1].

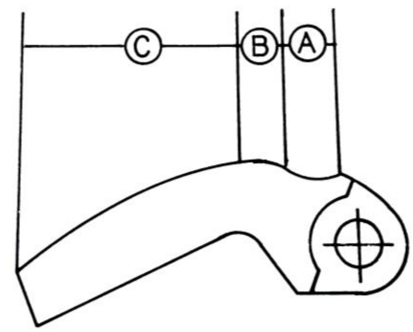

Figure 16: Flyweight with labelled phases [1]

Now that the concepts and geometry behind a flyweight have been fully introduced there are four more concepts to cover in relation to specific areas to tune using the flyweight.

These are as follows: the contact angle, engagement phase, transition phase, and the shift curve phase. These are outlined as italicized subsections below for clarity.

Contact Angle

The contact angle is the angle between the roller mechanism and the cam surface. This is best visualized by looking at the previously given Figure 15 to see the outer cam surface in question on the top of the cam and the roller which is shown on this surface near the left edge on the end of the cam. The contacted angle is defined in relationship to the shaft's center line, and at any given moment, determines how much of the centrifugal force being experienced by the flyweight is going to be converted to shift force. There are two ways to view the behavior in relationship to the contact angle. The first is that small angles convert a small percentage of the centrifugal force to shift force and large angles convert a large percentage. The second being that small angles need a faster engine speed (more RPMs) to produce the same force that a larger angle would produce at slower engine speeds (less RPMs).

Engagement Phase

When tuning there are two good ways to achieve a desired engagement speed; the first is to adjust the pretension of the pressure spring, and the second is to have the flyweight overcome the pretension at a certain engine speed. It is usually suggested that the pressure spring should be used to tune as much as possible without effecting the ability to keep the engine on the power peak while shifting before grinding the flyweight to effect

the engagement speed. If grinding the flyweight is necessary, there are two common practices for grinding the flyweight to change engagement speed, which are grinding notches and tucking under. Grinding notches or flat surfaces is the preferred method when grinding for engagement speed because it has less risk of creating undesired side effects. When grinding a notch or flat, the idea is either increase or decrease the engagement by changing the geometry of area A shown in Figure 16. To increase engagement less of an angle is desired, whereas to decrease engagement more of an angle is desired. Increasing engagement is done by grinding the flat or notch, a flat increase the engagement less than a notch because a notch as a far greater angle change to overcome. Now, the concept of tucking under is the relative position of the flyweight’s center of gravity (COG) to the pivot point for the cam. Tucking under should only be done as a last resort because it can create dangerous side effects, such as if the COG of the flyweight is directly under the pivot point the engagement speed becomes infinity meaning the engine could never build up enough speed to engage the primary clutch in reality [1].

Transition Phase

The transition phase is shown in Figure 16 as B and is essential to a well-tuned shift-out at the desired engine speed. The shape of the curve on this part of the cam depends highly on the desired effect for proper tuning of the CVT in question. One concept mentioned earlier can easily be achieved through manipulating the transition phase surface of the flyweight. This is creating an over-rev which can be done by extending the flat zone of the engagement phase so that it is further into the transition phase area of the flyweight.

This delays the change in angle from the engagement phase into the transition phase until higher engine speed, i.e, delays the transition until high revs.

Shift Curve Phase

The shift curve phase is shown in Figure 16 as C and is the largest phase on the flyweight as this controls how the engine shifts through the infinite number of gear ratios from low to high and in back shifting from high to low. As mentioned continuously throughout this thesis, an ideal shift curve is a straight one on the power peak for the engine, so the flyweight curve in this phase needs to be designed to obtain this straight behavior. Engine speed depends on the contact angle being experienced by the flyweight and the desired gear ratio at that time. The general behavior is, as mentioned under the contact angle subsection, the smaller the angle becomes with respect to the center of the shaft, the more engine speed it takes to create the same shift force. As a general rule, flyweights that have flatter surfaces need to have higher mass than flyweights with a more curved surface. In addition to the contact angle and cam surface geometry, the weight distribution of the flyweight itself has an effect on how shifting behaves. When thinking about weight distribution, it is all about the location of the flyweight’s COG. If you want a soft shift-out and aggressive top end, then have the flyweight be heavier on the tip. If a hard shift-out is needed and it is acceptable for the engine to rev more in the top end, then the flyweight should be heavier in the middle and light on the end.

Another option instead of grinding flyweights is to purchase new ones with a different shape. Manufacturers often times have several options for different flyweights which can be changed in and out of the primary clutch on the CVT.

Tuning the Driven (Secondary) Clutch

For the driven, efficiency of the transmission system is determined by having correct design for the components of proper tuning on the secondary sheaves and how they interact with the belt in the system. Four major factors can be changed in the secondary clutch in order to tune the CVT system: the torque feedback ramp, the pretension spring, the side force, and reducing the friction of the secondary clutch.

The torque feedback ramp consists of three ramps which acts as cams inside the secondary clutch. They work against the movable sheave by use of the sliding buttons. The ramp angle and radius the system is working on both affect the side force being used. The equation to solve for the theoretical side force is shown in section 3.3 as Equation 13 The general behavior for this is the smaller the ramp angles, the higher the side force. For the secondary, the belt pressure not only depends on the spring and its pretension, but also on the torque feedback of the ramps. Focusing on the spring and its effect on the CVT, it is important for the initial pressure of the belt in the system and back shifting through the gear ratios. When tuning for back shifting of the CVT, the higher the tension from the spring, the faster the CVT will back shift (go from higher to lower gear ratios). For the opposite, a lower tension spring makes response slower.

It is important to note that a lot of mechanics try to tune the speed of the engine (RPMs) by increasing the spring tension which makes it harder for the CVT to move the sheaves open, i.e, makes it harder for the CVT to shift. This is not the best way to accomplish this goal because there are some undesirable side effects of taking this tuning route. It changes the back shifting behavior of the CVT and lowers efficiency.

The last way to tune or improve the secondary clutch is less like tuning, but more of replacing parts to make an overall improvement in the system. One of two methods can be taken to reduce the friction in the secondary clutch. Friction is a problem in any mechanical system, it lowers efficiency by wasting energy, generating heat, slowing down the movement of parts, and wears out components faster than desired. The first and less effective method to reduce friction is closer to tuning the system than the second method. The idea is the reduce the friction experienced between the cams and the plastic sliding buttons that are common, which averages at about 10% in most stock modern CVTs. Reducing friction can be done by coating the cam surfaces in Teflon. This can reduce friction by up to 50% [1]. So after the Teflon coating, the friction drops from 10% to 5%, a significant difference in performance [1]. It should also be noted that most modern cams are made of aluminum to reduce weight and help with heat dissipation, but are not as durable as the old steel cams. This can be in part mitigated by coating the cams in a surface hard-coating to increase durability before applying the Teflon coating to reduce friction. The second method for reducing friction is far more effective than the first but is costlier and involves replacing the cams that slide against plastic buttons in the

clutch with a *Roller Action Clutch*. The roller action clutch design replaces the plastic buttons with needle bearing rollers that work against the clutch. The needle bearings get negligible friction when working in the CVT. It is said in *Olav Aaen's Clutch Tuning Handbook* [1] that this style of clutch eliminates friction. It is impossible to completely eliminate friction, but from the evidence presented in his book, it appears the friction drops to a negligible amount in comparison to other designs. No actual friction reduction statistics are directly given. There are two common types of roller action secondary clutches in use; the most common is one that uses aluminum cams with fiber rollers, the other is one that uses steel cams against steel rollers, which claims more reliable and consistent performance. With the introduction of this other style of clutch, it is important to elaborate on how to properly tune it once installed. On the left is an excerpt given as Figure 17 briefly explaining how the tuning process works.

Tuning A Roller Clutch

The first sign that your roller action clutch shifts out harder is a need to reduce the cam angles to keep the RPM. This clutch upshifts so fast into higher ratios that the standard cam angle will pull the engine down in RPM. It is therefore usual to drop the angles approximately 5° from stock on an Arctic roller (example: An Arctic ZR 580 drops from a 53° stock cam to a 48° roller cam) and approximately 2° on a Polaris. Since friction is eliminated, it is also necessary to run a stiffer spring and somewhat higher pretension because the friction forces added to the spring tension under shifting. This force component is now eliminated, and higher spring forces have to make up for it. On big displacement racing and trail engines a multiple angle is used. Racers want aggressive shift out, but an aggressive straight cam will pull the RPM down too much on top end. On a big Arctic the split for a stock machine may be 52°-48° (4°) while a 1000 cc open racer runs as much as a 56°-44° (10°). A greater split is needed in grass racing where tracks spin more, than on the ice. If you have a well made roller action clutch, you should end up with lower degree cams and stiffer springs to take advantage of the new friction-free environment where the track hooks up better.

Figure 17:Excerpt on Tuning a Roller Action Clutch [1]

The excerpt from the *Clutch Tuning Handbook* [1] was used due to the focus of this thesis being on the traditional style of secondary clutch. However, it was important to include some information on the basics of tuning the roller action style.

One concept not exclusive to either clutch but that pertains to the entire CVT is the concept of

Correct Inertia [1]. When an engine is designed and built, the crankshaft assembly has a natural torsional frequency which should be avoided to prevent the vibrational phenomenon known as *resonance* from occurring. Resonance is when the applied frequency of something matches the natural frequency of the system. When this occurs, the amplitudes of the frequencies stack, creating large vibrational disturbances which can greatly reduce efficiency or even destroy components involved. For a snowmobile, this is when the frequency of the crankshaft, which depends on the components mounted on it, occurs near the power peak where the system wants to operate. There is an easy way to fix this issue though: change the inertia of the components mounted on the crankshaft. The easiest component to change is the flywheel on the engine. This will shift the natural frequency away from the power peak which will reduce or nearly eliminate the problem.

Methods

3.1. Computer Modeling a CVT

3.1.1. Computer Model Concept

When tuning a CVT system there are a large number of factors influence the performance. An abundance of time is necessary due to trial and error that is used in the conventional process. The idea behind an MSC Adams computer model has the potential to make calculations needed for tuning a CVT to cut down on the number of iterations needed to tune the system. This will remove both time and money spent tuning the system while measuring the likelihood of a more optimal outcome.

3.1.2. Why use MSC Adams?

While the concept behind why a computer model is clear it is important to show how MSC Adams differs from other computer analysis programs. Before getting to far into why MSC Adams, it is important to point out the difference between Mechanics of Materials and Dynamics. In Mechanics of Materials the focus is on the study of internal forces and effects, whereas Dynamics focuses solving for loads being applied to systems that generate a certain behavior or motion of the entire system. Often times Dynamics is used first to solve for the loads that are then used in Mechanics of Materials to look at the internal behavior of an object. MSC Adams is a series of a computer modeling modules that work together to do rigid body analysis of systems. Adams is described as "The Multibody Dynamics Simulation Solution. Adams helps engineers to study the dynamics of moving parts, and how loads and forces are distributed throughout mechanical

systems." on the MSC webpage for Adams [3]. This sets Adams apart from other computer analysis software, such as FEA, which is used to find stresses and strains, or SolidWorks and AutoCAD which can be used for modeling objects and processes like flow analysis. In these other programs, you create a model and analyze specific components for aspects of those components structure or individual behavior. Adams, on the other hand, examines a series of components and how they are related to each other through position, joints, and their degrees of freedom and relative motion. This means Adams focuses on the dynamics of a system as a whole, not just the properties of any single component. Another difference between Adams and FEA or SolidWorks/AutoCAD is that Adams isn't just one program, rather it is a series of modules that do different things which work together to analyze a variety of aspects of the system which is modeled in Adams.

3.2. MSC Adams Model

3.2.1. Concept in MSC Adams

The concept behind the MSC Adams model, which was made using the module called Adams View, is to build a simplified version of a CVT which can simulate the rigid body behavior of the driver and driven clutches and the belt that connects them. Adams View was used to create two pulleys connected by a belt in order to be able to compute complex rigid body analysis of certain components without difficult and tedious hand calculations. This allows the user to run more permutations of ideas for how to better tune the CVT in a short amount of time.

3.2.2. Rigid Body Analysis

Since MSC Adams is built around Lagrangian rigid body dynamics for analysis, many of the concepts used behind the scenes in the program are not only useful for computations but for also for understanding the mechanics behind the motion of a CVT. There is a school of academic dynamics beyond that which is traditionally covered in a mechanical engineering undergraduate curriculum which is Lagrangian dynamics. This study of rigid body analysis uses kinetic and potential energy to be able to look at more advanced scenarios than traditional dynamics. Typically, rigid body dynamics is viewed as the study of external forces applied to bodies and their respective motion in global space. For example, the study of the momentum of a golf ball bouncing off the floor would be rigid body dynamics.

When learning about Advanced Dynamics, otherwise referred to as Lagrangian Dynamics, the following steps are helpful in setting up a scenario for analysis. These steps are taken from *Fundamentals of Applied Dynamics* by James H. Williams, Jr. [4].

Formulation of Equations of Motion via Lagrange's Equations for Holonomic Mechanical Systems [4]

Generalized Coordinates

$$\varepsilon_j \ (j = 1,2, \dots, n) \tag{5}$$

Admissible Variations

$$\delta\varepsilon_j \ (j = 1,2, \dots, n') \tag{6}$$

Generalized Forces

$$\delta W^{nc} = \sum_{i=1}^{N} \varphi_j * \delta\varepsilon_j \tag{7}$$

Lagrangian

$$L = T^* - V \tag{8}$$

Equation of Motion

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{\varepsilon_j}}\right) - \frac{\partial L}{\partial \varepsilon_j} = \varphi_j \tag{9}$$

These equations form a general process for using Lagrangian dynamics to analyze situations. The specifics of using these equations, and the difference between holonomic and non-holonomic constraints, can be further explored by reading the dynamics textbook cited as the reference for these equations, but will not be explored further here because it is not directly important to the scope of this thesis but was important to introduce.

3.2.3. Building the Model

The first part of building a CVT model in MSC Adams was having some geometry to base it on. The easiest way to do this was to literally take apart an existing CVT. This was done by removing components from a 2004 Arctic Cat 4-Stroke with similar components and design to the CNG snowmobile to build models that could be utilized with the CNG snowmobile with a fair amount of accuracy without having to remove components from

Figure 18: 2004 Arctic Cat 4-Stroke CVT System

the CNG snowmobile while it was under construction for capstone projects. The following figures show the CVT system on and off the snowmobile and provide the basic geometric dimensions that were taken to model the snowmobile in MSC Adams. Figure 18 shows an image of the CVT used as an example installed on the 2004 Arctic Cat 4-Stroke with the belt loosened off the primary clutch of the snowmobile.

The next image given as Figure 19 shows an image looking directly at the side of the CVT system when installed on the snowmobile. The primary clutch can be seen on the left, the secondary clutch on the right, and the belt is hanging

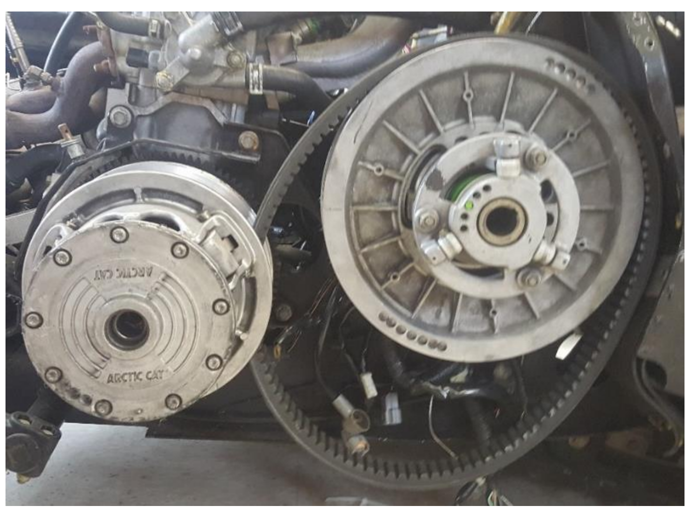

Figure 19: Side View of the CVT

around the secondary clutch. After removing these components from the snowmobile, they were taken apart to understand how they are put together, and some components were measured for parameters needed to build the MSC Adams models. It was found while measuring that when the belt was formed into a circle for convenience measuring, it had an outer diameter (OD) of 15.5 inches and an inner diameter (ID) of 14.5 inches, which means it has a thickness of 1 inch. For the primary clutch it was found that the minimum diameter of the sheaves was 2 inches and the maximum diameter of the sheaves was 8 inches. For the secondary clutch the minimum diameter of the sheaves was 6 inches and the maximum was 11 inches. These numbers were used when forming the geometry of the pulleys in the MSC Adams View models.

SolidWorks Computer Modeling

In addition, the flyweight from this clutching system was removed and modeled in SolidWorks in order to add a helpful visual for an actual clutch as well as to find the mass moments of inertia for the flyweight that was stock in the system. SolidWorks is a useful tool for analysis because it can easily find the mass and moments of inertia for any part or assembly modeled in it.

The flyweight from the 2004 Arctic Cat clutch is shown on the right as Figure 20 from the top to show the easiest profile of the part as it is seen removed from the CVT.

Figure 20: 2004 Arctic Cat 4-Stroke Flyweight

The part shown in Figure 20 came from the clutch used to build the Adams View models and was used as a basis to create the SolidWorks part shown as Figure 21 on the right in isometric view. Using this model, the SolidWorks mass properties function was used to generated the mass properties of the flyweight. These properties are given as output from SolidWorks as follows:

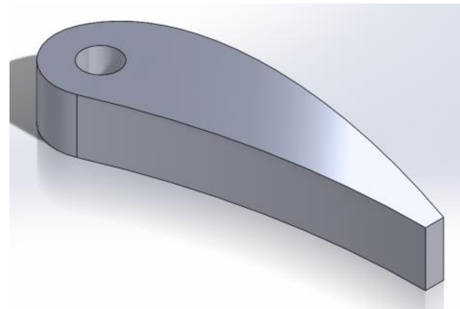

Figure 21: SolidWorks Flyweight Model

Mass properties of FlyWeight

Configuration: Default

Coordinate system: -- default --

Density = 0.28 pounds per cubic inch

Mass = 0.12 pounds

Volume = 0.43 cubic inches

Surface area = 4.68 square inches

Center of mass: (inches)

X = 0.71

Y = 0.19

Z = -0.19

Principal axes of inertia and principal moments of inertia: (pounds * square inches)

Taken at the center of mass.

Ix = (0.99, -0.00, -0.12)	Px = 0.01
Iy = (-0.12, -0.00, -0.99)	Py = 0.05
Iz = (0.00, 1.00, -0.00)	Pz = 0.05

Moments of inertia: (pounds * square inches)

Taken at the center of mass and aligned with the output coordinate system.

Lxx = 0.01	Lxy = -0.00	Lxz = -0.00
Lyx = -0.00	Lyy = 0.05	Lyz = 0.00
Lzx = -0.00	Lzy = 0.00	Lzz = 0.05

Moments of inertia: (pounds * square inches)

Taken at the output coordinate system.

Ixx = 0.01	Ixy = 0.02	Ixz = -0.02
Iyx = 0.02	Iyy = 0.12	Iyz = -0.00
Izx = -0.02	Izy = -0.00	Izz = 0.11

These values can be very useful during tuning to understand the parts being created and to simulate grinding off some of the part in SolidWorks before doing so to verify the significance of how much you would like to grind off the part. To get the most out of using this feature in SolidWorks, it is necessary to find a way to set the axis about which it takes the mass property values to be the pivot point which in the model is the center of

the hole through the flyweight. In addition, it should be noted that SolidWorks lists the mass moments of inertia as the last set of moments of inertia in the list provided.

MSC Adams Computer Modeling

Three MSC Adams computer models were made to demonstrate three positions in the shift range for a CVT. The first model being the low ratio for the beginning or take off of the snowmobile. The second being a mid-shift range showing the particular case of a 1:1 ratio. The third is a high ratio, possibly over-drive. The reason three models were constructed was to fully capture the shift range of a generic CVT system. The dimensions used are as mentioned in this case from a 2004 Arctic Cat 4-Stroke of similar design to the CNG snowmobile. As was explained above in section two, a CVT works by changing the dimensions of the pulley that the belt experiences based on the moveable sheave changing position in reference to the stationary sheave. MSC Adams cannot design a pulley that can change dimensions based on RPM input. This is a limitation which caused a restructuring of the models from original thoughts. Now the three models use an applied torque through a motor on the driver pulley, which means in order to simulate the various pulley dimensions that happen through shifting the three models previously mentioned were needed. The following nine figures show three different shift ratios with three views of each. The first view is an isometric view, then a side view, and then a top view for each of the shift ratios. Starting with a low-ratio, then a 1:1 middle-ratio, and

then a high-ratio.

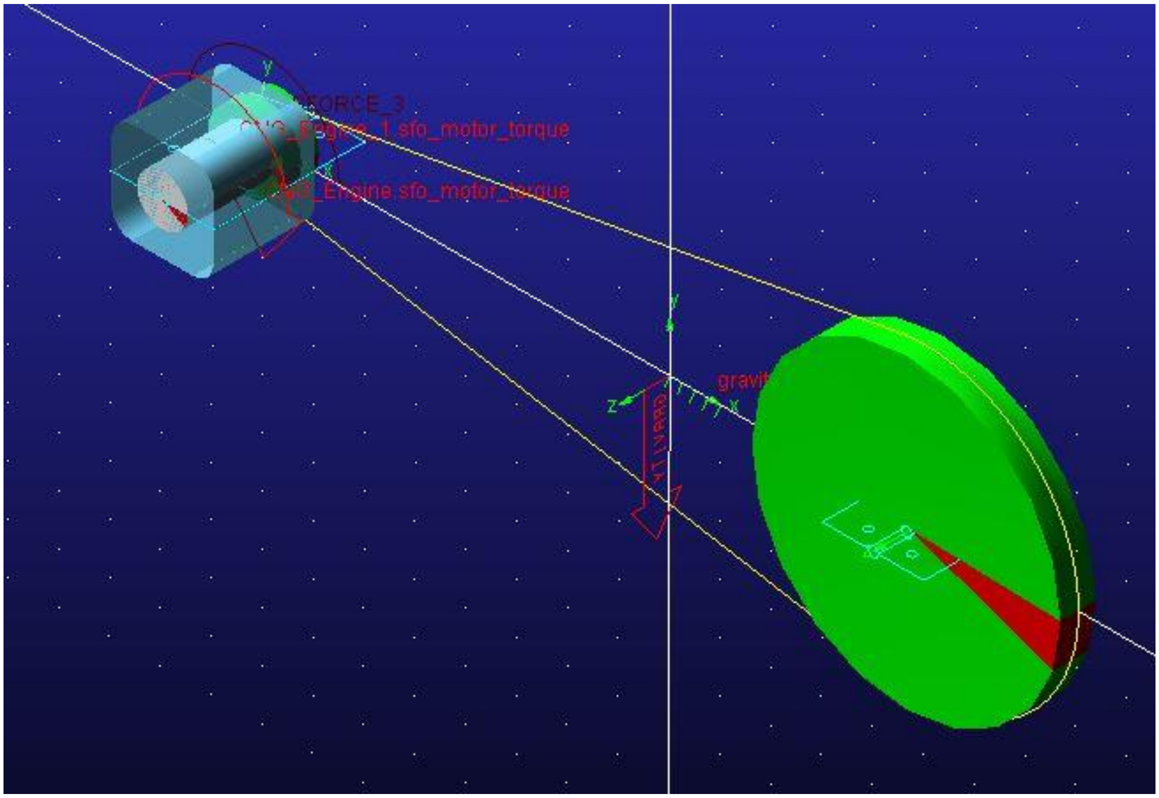

Figure 22: Isometric View of a CVT in low-ratio

The isometric view in Figure 22 above shows the low-ratio position of the CVT as it can be seen from the side of a snowmobile. Looking at the isometric view on a snowmobile, the outside of the chassis would be on the left, and the engine would be on the right. In the MSC Adams model the engine is shown on the left, even though on the actual snowmobile the input shaft would be coming in from the right. The reason for this is a limitation in the visual abilities of Adams View; this does not change the performance of the model. One advantage of the engine placed on the model is the level of control Adams View allows through this machinery component. Not only does it allow for a specified torque to be applied to the primary pulley which was used for the models

developed in this paper, it can also take what is called a spline file. A spline file is an RPM vs. torque file which can potentially be measured from the SAE Team dynamometer for the CNG snowmobiles engine. This would allow for running simulations in Adams that directly relate to the CNG snowmobile, which is the end goal of this thesis to help tune the snowmobile.

Figure 23 that follows shows the low-ratio of a CVT from a side view. On the left the engine can be seen which applies a simulated torque to the primary pulley which is shown as green behind the semi-transparent engine that is shown with the input shaft in gray. The pulley on the right side is the secondary pulley which takes the transferred torque through the belt shown in yellow and puts it into the output shaft which is not shown but takes the torque from the driven clutch and sends it to the differential to power the track.

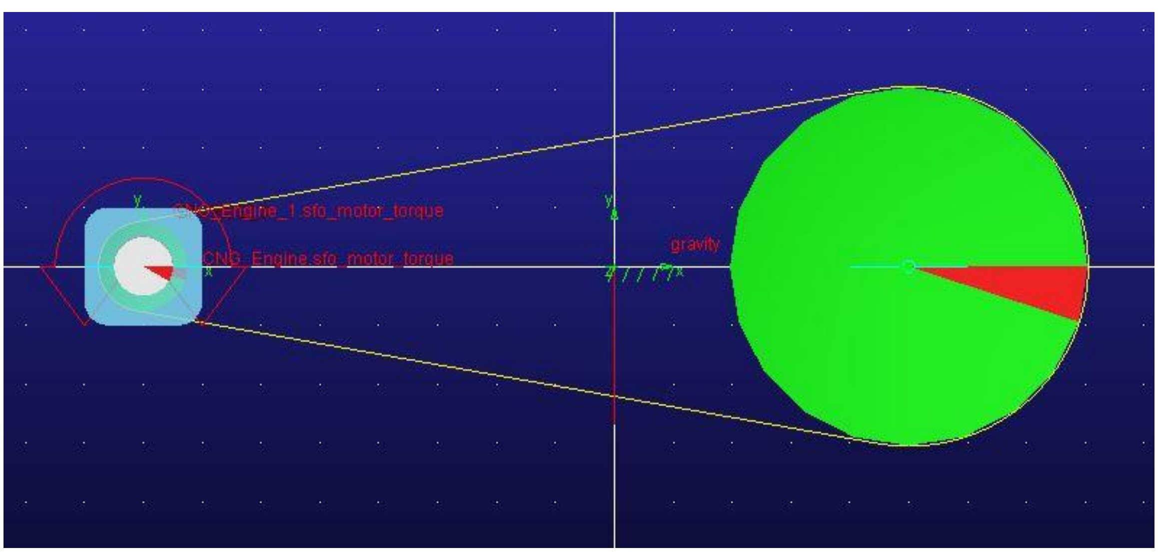

Figure 23: Side View of a CVT in low-ratio

The previously shown Figure 23 is a good example of a CVT being seen from the side before it has reached the shift phase. That means the primary clutch is still at its minimum diameter the belt can feel, but if the system is engaged, then there is enough pressure from the sheaves to spin the belt. As for the secondary clutch, it would still be out at its furthest diameter position because the sheaves haven't begun to separate and shrink the diameter the belt experiences which is done during the shifting phase.The next and final image of the CVT in low-ratio from Adams View is Figure 24, a top view of the model created. This view best shows the difference in diameter of the two clutches and how they are mounted on a joint created in Adams that is fixed in position in space, but allows the pulley to spin freely about this fixed position. This is the best choice for a joint that Adams offers because it simulates the motion of the pulley spinning about the axle on the snowmobile which has a fixed position in space but allows for free angular velocity and acceleration about that axle.

The last thing to note about the low ratio Adams View model shown in Figures 22

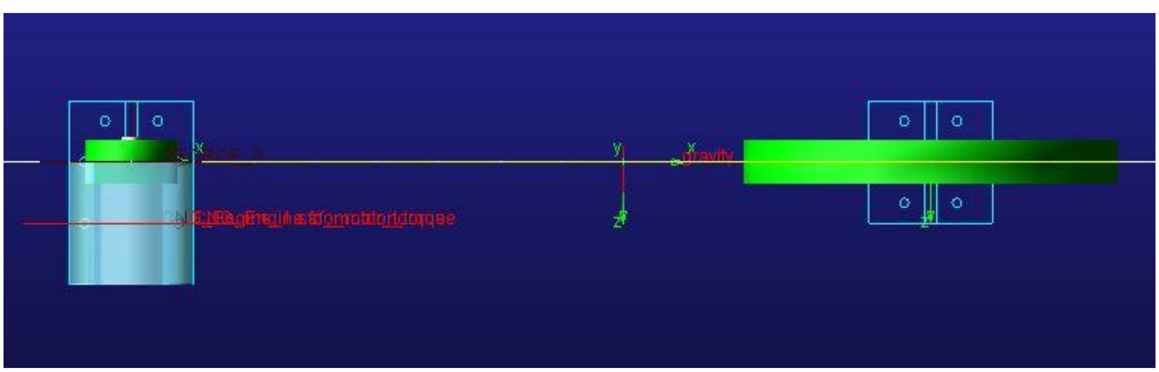

Figure 24: Top View of a CVT in low-ratio

through 24 is that it represents the absolute minimum position in terms of ratio that can be experienced for the geometry of the CVT that was used as an example. A low ratio doesn't mean it needs to be at this extreme position, it simply means that the diameter of the driver pulley needs to be smaller than the diameter of the driven pulley. For the Arctic Cat clutch used which follows a similar geometry and design to the CNG snowmobile the minimum ratio in low end is 4:1.

As the CVT experiences higher torque at the driver pulley from the input shaft being driven by increasing engine speed (climbing RPMs) the flyweights will continue to shift further outward due to increasing centrifugal forces. Once the RPMs are reached that push the flyweight into the shifting range of its geometry, the clutches will begin changing in diameter as the belt shifts out on the driver and the belt shifts in on the driven, in turn guiding the snowmobile transmission system through its shift range. The next model built was made for a mid-range position in the shift path. A special scenario was chosen for its unique dynamic properties; this is a 1:1 ratio.

The first view of the mid-range model is the isometric view, which lends the best overall visual perspective to this model and is shown as Figure 25. The model follows the same format as the low ratio model except for the fact that the diameter of the primary is the same as the diameter of the secondary, hence the 1:1 ratio as described in section two.

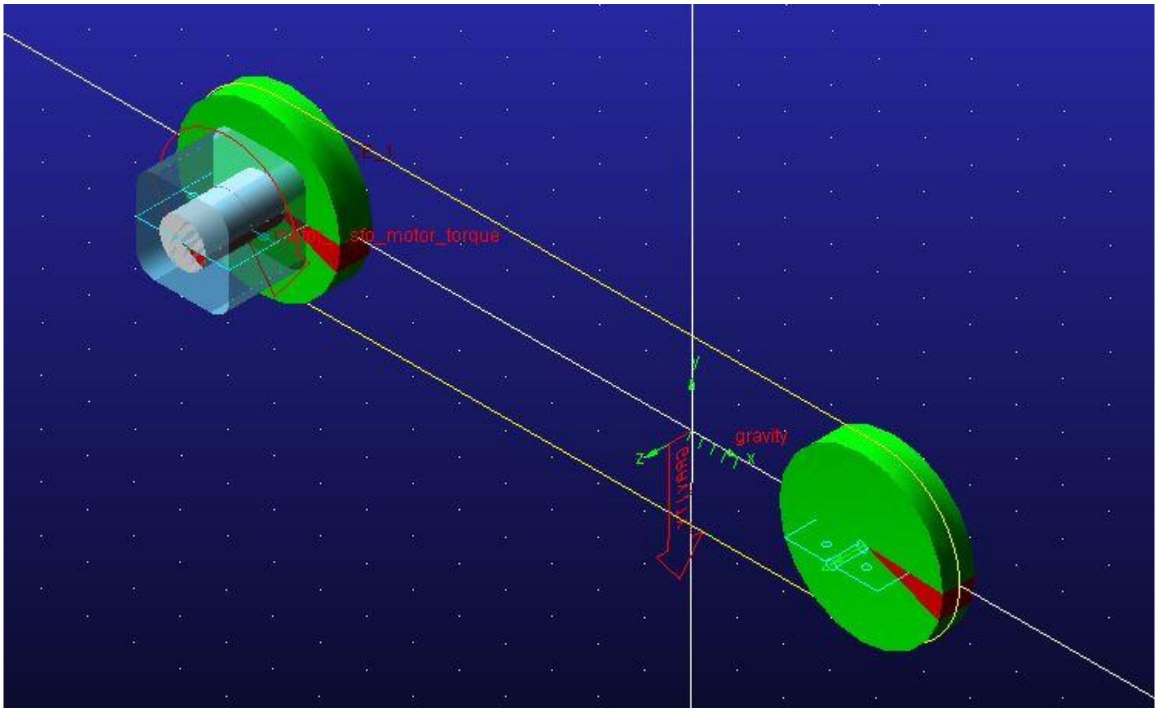

Figure 25: Isometric View of a CVT in mid-ratio

The uniqueness of this point in the CVT shift range is this is the only place where the torque of the input shaft equals the torque of the output shaft. Figure 26 that follows shows the mid-ratio model from the side further illustrating the equivalent diameters of the pulleys at this position.

Figure 27 that follows shows the mid-ratio model in top view, which is the last view, and

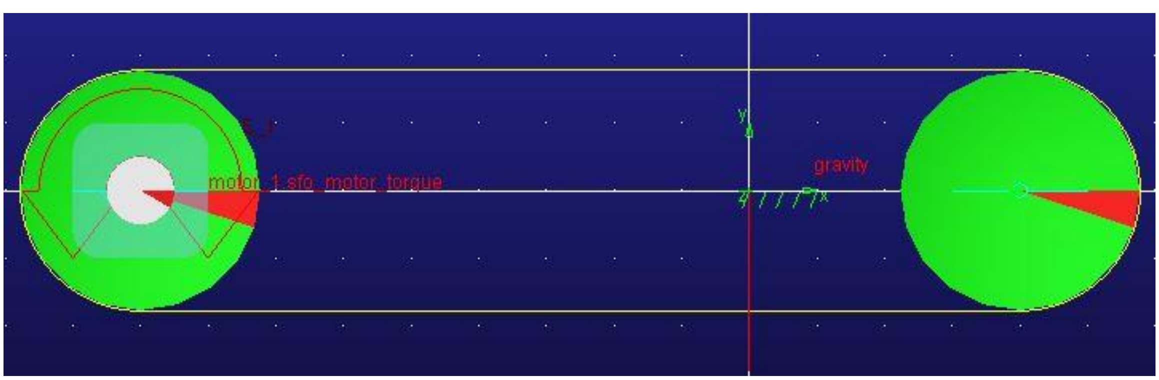

Figure 26: Side View of a CVT in a 1:1 mid-ratio

further illustrates that the two pulleys in the 1:1 ratio are in fact the same diameter as they are intended to be in this unique gear ratio.

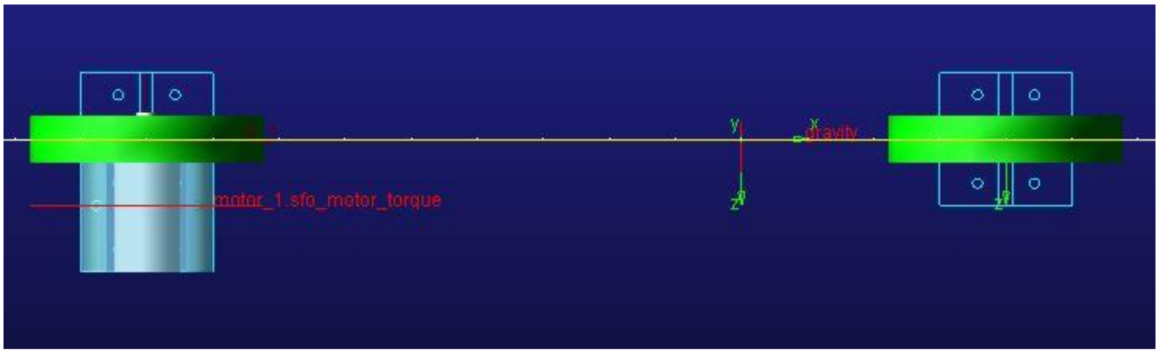

Figure 27: Top View of a CVT in a 1:1 mid-ratio

The third and final model created in Adams View is for a high ratio, which could also be considered over-drive for the snowmobile. This model follows the same basic structure as the previous two models except for that the two pulleys have a different diameter. At this point in the shift range of the CVT, the belt has moved all the way out on the driver clutch and moved all the way in on the driven clutch. A high ratio doesn't mean that the belt has to be at this extreme, a high ratio is when the driver pulley diameter is larger than the driven pulley diameter on a CVT. In high ratio the driver is actually putting out more torque than the driven is experiencing because the system is geared up, meaning the driven has a smaller diameter pulley, making the ratio between the two shrink the torque being transmitted.

Figure 28 that follows shows the high ratio Adams View model that was created in isometric view. This, like the other isometric views given in Figures 22 and 25, shows the best image of the model as a whole in high ratio. It can be clearly seen that the driver clutch attached to the engine on the left has a larger diameter than the driven clutch on the right which is connected to the output shaft.

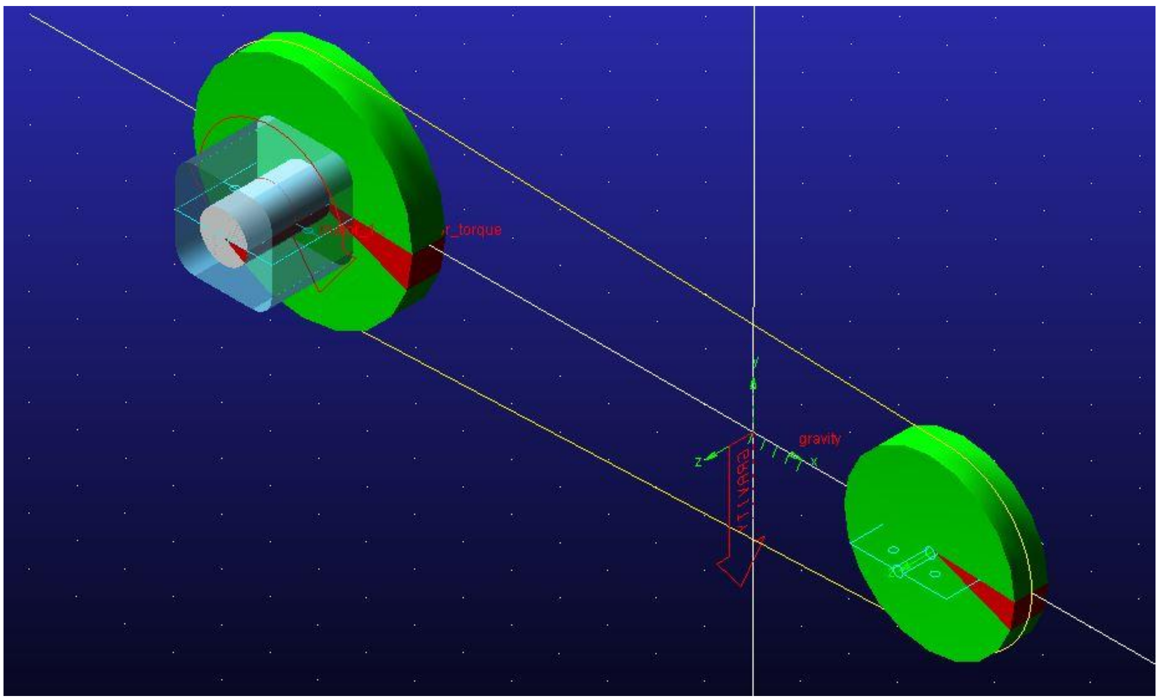

Figure 28: Isometric View of a CVT in high-ratio

The next graphic Figure 29 shows the high ratio CVT from the side which best illustrates how, in high ratio, the driver pulley has a larger diameter that the belt experiences than the driven pulley.

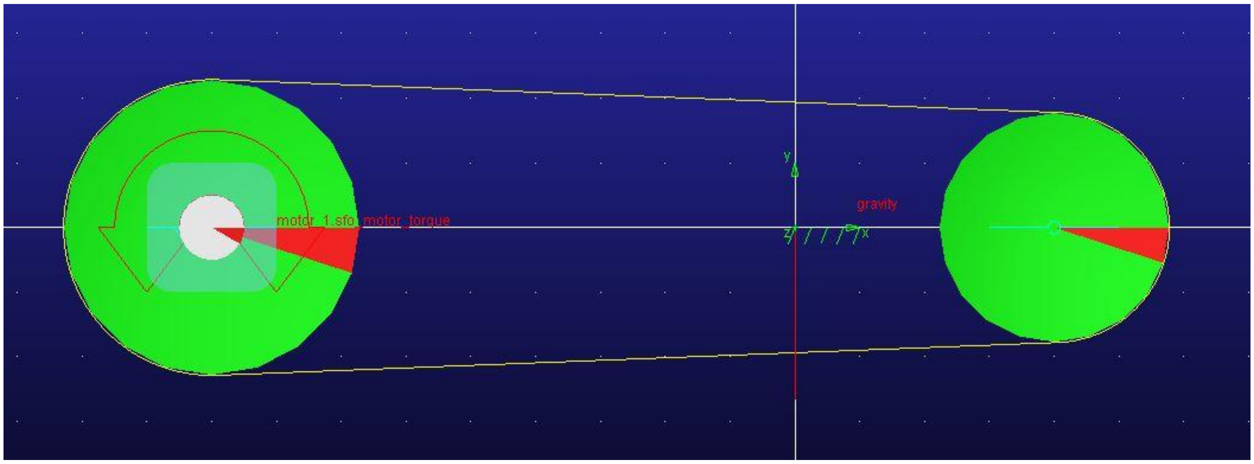

Figure 29: Side View of a CVT in high-ratio

Figure 30 that follows shows the high ratio CVT from the top to give a full picture of the model. It is easy to see here from another perspective the fact that once high gears have been achieved that the driver has a larger diameter than the driven.

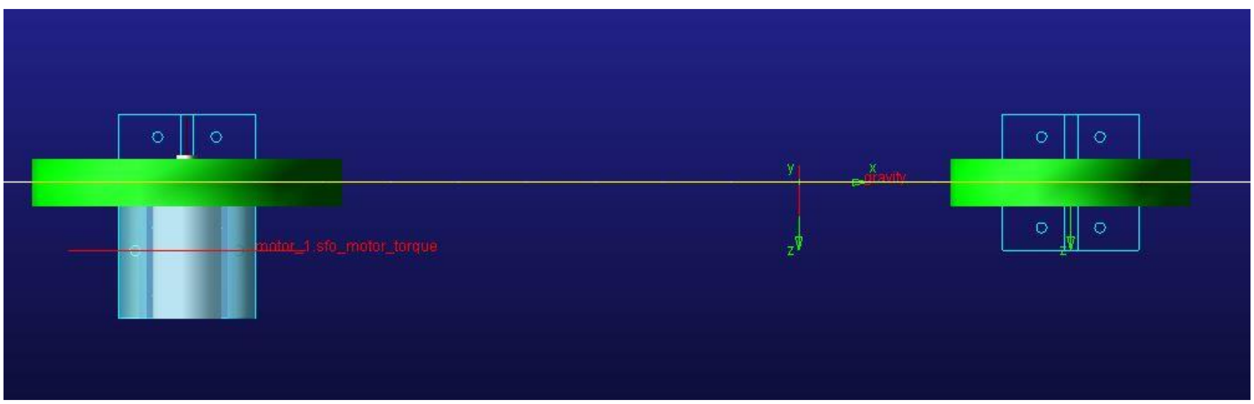

Figure 30: Top View of a CVT in high ratio

It should be noted that the high ratio, like the low ratio model made, was chosen as the high ratio model because for the particular clutch chosen, this is the other extreme in the shift range. The high ratio model shown in Figures 28 through 31 is the maximum shift position in high ratio for this Arctic Cat clutch.

It should be noted that for all three models generated in Adams View shown in Figures 22 through 30 the distance between the two joints that ground the driver and driven clutch in space are the same distance apart. This design decision was made in the modeling process because that is what is experienced on a snowmobile in reality. The position of the input shaft that the driver is mounted to and the position of the output shaft which the driven is mounted to do not change in space during the operation of the engine. Also, for the purposes of this thesis due to the current lack of an accurate spline file, a constant torque applied from the engine was used for analysis.

3.3. Equations Needed for Analysis

Due to the limitations of MSC Adams and the amount of time it takes to develop a model in such advanced commercial software, there are quantities that need to be calculated outside of what the current model is capable of. These equations will be listed and discussed in this section.

When thinking about CVTs it is important to know the relationship between horsepower (HP) and torque. Equation 10 [1] that follows shows the simple relationship between HP and torque.

$$HP = T * V \tag{10}$$

HP is the horsepower being generated by the torque T at the pulleys current speed V. This equation was previously shown in text under a different number, but was renumbered from a previous equation to Equation 10 here for clarity in this collected list of equations.

Like horsepower (HP), torque is an extremely important concept in any application of engines and transmissions, but it is especially important for CVTs that use torque consistently for the control and operation of its mechanical components. Equation 11 [5] below gives the basic definition of torque T given in pound-feet (lb-ft), r is the moment arm given in feet (ft), and P is the applied load given in pounds (lb).

$$T = P * r \tag{11}$$

For further explanation of torque (sometimes referred to as a moment in engineering) please refer to an external source. Also, the above equation assumes the force is being applied perpendicular to the pivot point, i.e, perpendicular to the r vector in space.

The general expression for the mass moment of inertia is shown below as Equation 12 [6] and can be used for reference when thinking about situations in tuning for a better understanding of the engineering principles at work.

$$I = k * m * R^2 \tag{12}$$

I is the mass moment of inertia, k is the inertial constant, m is the mass in pounds mass (lbm), and R is the distance between the defined axis (for example this would be the pivot point on a flyweight in the primary clutch when looking at that component), which is generally given in inches (in) or feet (ft) depending on the magnitude of the value of I. Further information for specific shapes or a better understanding can be found in many engineering textbooks or engineering websites.

Looking at a CVTs changing factors in the secondary clutch is how efficiency can be improved. One of the most important factors in that is proper strength of the side force applied. Too much side force leads to low efficiency. To find the theoretical side force produced by the cam for any shift ratio Equation 13 [1] can be used.

$$SF = \frac{T * SR * 12}{2 * r * \tan\alpha} \tag{13}$$

SF is the side force, T is the torque in ft-lb, SR is the shift ratio, r is the ramp radius in inches, α is the ramp angle. Equation 13 was renumbered from a previous equation for clarity in this list.

When looking at the pressure spring in the primary clutch it is necessary to know how the spring force relates to the belt pressure, which is the most important concept in tuning the primary clutch. Equation 14 [1] below is renumbered from a previous equation in the text, and shows the relationship between belt pressure force (Fbp), flyweight force (Ffw) and spring force from the pressure spring (Fps).

$$Fbp = Ffw - Fps \tag{14}$$

Another important force to know when tuning is the pretension force in the primary clutch. This is a simple value to compute but completely controls the engagement speed of the engine and is shown as Equation 15 [1] where the Fpt is the pretension force, Rs is the rate of travel for the spring, lf is the free length and li is the installed length. For clarity on the meaning of the lengths see Figure 14.

$$Fpt = Rs * (lf - li) \tag{15}$$

One of the important factors in tuning is knowing how efficient your current system setup is so you can determine if it is feasible to make it better by continuing to tune things. Equation 16 [1] that follows gives a simple equation to find the drive efficiency of a CVT.

$$\eta = \frac{Driven\ HP}{Driver\ HP} \tag{16}$$

η is the drive efficiency of the CVT, and the numerator is the HP of the driven (secondary) clutch and the denominator is the HP of the driver (primary) clutch.

3.4. Method for Using the Model and Equations

Since the MSC Adams model built cannot complete all of the calculations necessary for the complex process necessary to tune a CVT, both the model and a series of equations were outlined in the previous sections 3.2 and 3.3 which can be used to complete tuning a snowmobile. This section will bring the model and the equations together to create a process to assist in tuning the CVT on a snowmobile.

The following series of ten steps gives a framework for the recommended course of logic to take when using the model and equation set to make calculations for assistance in tuning a CVT system.

Step 1: Open the Adams View models and change the low ratio model dimensions to the minimum diameter for the primary pulley and maximum diameter of the secondary pulley experienced by the belt for the clutch model you are using. Now change the mid ratio model to be the diameters where your clutch meets for a 1:1 ratio. Then change the high ratio model to be the maximum diameter for the primary pulley and the minimum diameter for the secondary pulley.

Step 2: Run the Adams View models by editing the machinery engine module with a spline file taken from the engine you have or manually input torques for analysis. If a simpler analysis is wanted, a constant torque can be applied using the engine module.

Step 3: Use the Adams View simulation menu to run a simulation for the desired amount of time. Then use the plot tool to create graphs of the desired quantities for analysis. It is recommended that angular velocity of both the driver and driven be plotted as a function of time. This is easiest to analyze when a constant torque is applied from the engine module for simplicity.

Step 4: Now that angular velocities of the pulleys are known for a given torque input parameter of the primary pulley, the equations laid out in section 3.3 can be used to begin building the framework for tuning the CVT. Now the steps will move away from the computer model and focus first on tuning the driver clutch and then on tuning the driven clutch.

Step 5: Begin tuning the driver clutch by first thinking about what you need to change. The driver clutch is used primarily to control engine speed, i.e, engagement speed, and keeping the engine on the power band. This can be done by altering flyweights or changing the pressure spring. Equations to analyze these will be visited in the following steps 6 through 8.

Step 6: First looking at the easier hardware to tune in the primary clutch the pressure spring and the two most useful equations are equation 14 and 15. The pressure spring is

responsible for pretension to control engagement speed and for, in part, controlling belt pressure which is critical to shift control. First take equation 14 as given below with the unknown values of belt pressure force (Fbp), flyweight force (Ffw), and the spring force (Fps) and think about what parameter you are trying to change.

$$Fbp = Ffw - Fps \tag{14}$$

Equation 14 gives the relationship, but it takes intuition and an understanding of what you need to do to tune for certain outcomes to apply the equation. For more information on how to tune to achieve certain outcomes see section 2.2 of the report. Next, equation 15 gives more information about the spring which can be combined with the known spring rate from the spring manufacturers specifications to find the spring force needed for equation 14, or use equation 15 directly to figure out what is needed for a spring and preset distance to achieve a specific pretension force to work against known flyweight parameters.

$$Fpt = Rs * (lf - li) \tag{15}$$

Fpt is the pretension force, Rs is the rate of travel for the spring, lf is the free length and li is the installed length. For clarity on the meaning of the lengths see Figure 14. This step may not give a complete picture of how to apply equations to tune the pressure spring, but it gives a fundamental basis to work with.

Step 7: The focus in this step is to introduce equations useful for step 8 on the flyweights in the primary clutch. The flyweights are by far the most complicated and hard to tune part of the CVT system. Therefore, the following adheres to the framework for providing

a basis for tuning but not necessarily the complete picture. To start, equations 10 and 11 are given to help find horsepower and torque from other parameters, which in turn relates applied torque, power of the engine, and speed of the pulley.

$$HP = T * V \tag{10}$$

HP is the horsepower, torque is T, and the pulleys current speed is V. Equation 11 below gives the basic definition of torque T given in pound-feet (lb-ft), r is the moment arm given in feet (ft), and P is the applied load given in pounds (lb).

$$T = P * r \tag{11}$$

Now that the ability to find horsepower and torque have been introduced, how do they apply to tuning the flyweights? The behavior of the flyweights is thoroughly described in section 3.3 and isn't going to be replaced here. However, the general form of the moment of inertia given as equation 12 is shown and a brief discussion of how the computer model can be used to explore flyweight changes will be discussed in step 8.

$$I = k * m * R^2 \tag{12}$$

I is the mass moment of inertia, k is the inertial constant, m is the mass in pounds mass (lbm), and R is the distance between the defined axis (for example this would be the pivot point on a flyweight in the primary clutch when looking at that component) which is generally given in inches (in) or feet (ft) depending on the magnitude of the value of I.

Step 8: Due to time restrictions and limitations in the current Adams View model this step is not as extensive as it needs to be for complete information on tuning the clutch.

But a framework for understanding how to apply the equations given in step 7 to tuning the flyweights is laid out through a thought experiment.

Thought Experiment

Imagine one of the pulleys of fixed diameter and a constant applied torque. Now this pulley has a certain mass moment of inertia at any instant in time, however this is effected by the flyweights as they move from increasing centrifugal force. What would happen if the mass moment of inertia of the pulley increased but the torque and pulley diameter remained the same? It would be seen that since the mass moment of inertia is now higher that the disc would resist movement more than if it had a lower mass moment of inertia.

This thought experiment can be applied to the MSC Adams model by manually altering the mass moments of inertia of the pulleys in the pulley design options. Then, when plotting the angular velocity over time of the pulleys, the effects could be seen.

Step 9: Now the secondary clutch can be looked at for tuning purposes in both this step and step 10. The two most important concepts in the secondary clutch are the side force exerted on the belt and the efficiency. Equations 13 and 16 begin to outline how these parameters can be tuned in a more refined manner.

Side force is important because it effects shifting, belt wear, and efficiency. To find the theoretical side force produced by the cam for any shift ratio, Equation 13 can be used.

$$SF = \frac{T*SR*12}{2*r*\tan\alpha} \tag{13}$$

SF is the side force, T is the torque in ft-lb, SR is the shift ratio, r is the ramp radius in inches, α is the ramp angle. Equation 13 was renumbered from a previous equation for clarity in this list. Using this equation, known parameters from earlier analysis, and the model, the current side force can be found and from there parameters can be changed in the equation to explore ways to reduce or raise the theoretical side force.

Step 10: The final equation given in the framework for assisting in the tuning of a CVT is equation 16, which is the best way to check progress of tuning. It gives a simple relationship for how to find the efficiency of the CVT system.

$$\eta = \frac{Driven\ HP}{Driver\ HP} \tag{16}$$

η is the drive efficiency of the CVT, and the numerator is the HP of the driven (secondary) clutch and the denominator is the HP of the driver (primary) clutch. The horsepower values needed can be found theoretically using equation 10 or experimentally as described in the tuning section previously.

The framework given in this section may not complete the picture on how to tune a CVT system but does create a base for calculations and modeling in Adams View which can be

used by the SAE Clean Snowmobile project or anyone who needs to tune their CVT system using engineering rational to make educated decisions.

Discussion of Results

4.1. Final MSC Adams Model Discussion

4.1.1. Final Model Structure

As discussed extensively in section three, the MSC Adams model made in Adams View, coupled with the collection of analytical equations, makes the process for tuning a clutch significantly different from the traditional method of guess and check tuning in the sense that now these tools can be used methodically to critically review the CVT system and make educated decisions on how to tune the system on sound, well-established engineering rational.

The structure of the final rendering of the Adams View model used the program's built-in machinery tools and an understanding of the dynamics of the CVT system achieved from extensive research and the application of engineering principles. As was discussed in section 3.2 the final structure of the Adams View model is two cylindrical pulleys connected by a continuous belt that transfers the torque from the first pulley, representing the driver (primary clutch), to the driven (secondary clutch) through a belt relationship that simulates the role of the rubber belt in reality. Due to limitations in the Adams program, variable diameter pulleys could not be made to react to the changing centrifugal force the primary pulley would experience. So the three models were created to show different points in the shift range. The CVT_Low_Ratio Adams View model was created for the engagement geometry of the CVT which is the absolute minimum point in low gear for the system used. The CVT_Mid model was created for the unique position of the

1:1 ratio in the clutch for calculations in the middle range. Lastly the third model CVT_High_Ratio was built for analysis of the maximum shift ratio possible in high gear. This final structure in MSC Adams allows analysis over the entire shift range of the Arctic Cat CVT system that was used as an example.

Using the final structure of the model discussed above, the equations laid out in section 3.3, and the process explained in section 3.4, there is a solid framework for how to tune a CVT. With use of the process developed in this thesis the tedious but essential process for tuning a CVT has been laid out on a high technical level for application by engineering students in the SAE Clean Snowmobile Project for the further development of the CNG Snowmobile as well as anyone else who wishes to increase the performance of their snowmobile with a process to follow rooted in sound engineering principles.

4.1.2. Advantages of the Model

With the creation of any tool, it is important to weigh the pros and cons of that system to help establish its merit and its limitations for those who may wish to use it or improve upon it. This subsection outlines the advantages of the model.

The biggest advantage of the model is the ability to have a visual aide for understanding the function of a CVT at a given gear ratio. This cascades into the advantage of Adams View being able to easily calculate the increase or decrease in angular velocity from the primary clutch to the secondary clutch, which is necessary for finding the torque transfer.

This leads into the fact that the Adams model can accept a spline file for any engine which can dictate the torque at any given engine speed and translate that into the performance of the model for analysis. Then the results can be plotted for the user to see what is happening in the behavior of the CVT for that given loading condition. This saves the user hours of work guessing and checking because the model can run permutations instantly for any loading condition added with a few simple steps in Adams View. In addition, Adams has the added advantage of allowing the user to fully customize the mass moments of inertia for the pulleys in the model. This allows the user to easily simulate values in order to find a situation where the engine can be held on the power peak for a given set of parameters and then back out the flyweights necessary to achieve this parameter, making it easier to tune the CVT because the result that is necessary to improve things is a known quantity.

4.1.3. Shortcomings of the Model

Following the previous subsection on the merits of the Adams View model, this one outlines the disadvantages of the model created to help illustrate its limitations and areas that need improvement to achieve better results.

The biggest disadvantage to Adams View is its inability to have dynamically changing diameters on pulleys during motion. This made it impossible to build a conventional model for the full range of motion of a CVT shifting because the clutch pulley diameters need to change for the shifting of gears to occur. This made it necessary to rethink how to

model the CVT for tuning because the original method for completing the scope of the project was not possible. This really leads to the biggest disadvantage of MSC Adams, which is the extremely steep learning curve that is necessary to use the program coupled with almost no information readily available to learn how to use it. This lead to the inability of the model to directly compute all of the quantities needed for tuning. Time constraints, lack of formal training with the commercial software, and limitations of the software itself lead to the limited abilities of the model. However, it can be used for calculating essential values for tuning and greatly reduce time spent per permutation of input parameters in addition to building in the ability to easily define mass moments of inertia. Which leads to making educated decisions on how to grind or replace components like flywheels or change the pretension or entire spring in the clutches.

4.2. Numerical & Graphical Results

4.2.1. Numerical Results

Since the results taken from MSC Adams are exclusively shown in the next subsection for graphical results because this is the best way to extract meaningful information from the Adams View models, the Numerical Results subsection is going to focus on sample calculations. These calculations are done with the equations shown in section 3.3 which are used in the methodology for tuning presented in 3.4 in order to better illustrate how to properly use the information provided. The numbers used in these calculations were chosen for convenience and do not pertain specifically to any CVT system.

Each equation outlined in section 3.3 will be shown with units in the order they are introduced from equation 10 to equation 16.

Starting with equation 10. The value of torque and velocity were both chosen to be unity for simplicity and have the appropriate units in the calculation.

$$HP = 1\ lb - ft\ * 1\ RPM * \frac{1}{33{,}000} = \frac{1}{33{,}000}\ HP$$

Now the unit conversion used is 1 HP equals 33,000 lb-ft/min.

Now finding the torque T when a force and distance of unity are chosen is shown.

$$T = 1\ lb * 1\ ft = 1\ lb - ft$$

This application assumes the sine term in the torque equation drops out because the force is perpendicular to the moment arm making a ninety-degree angle so the sine of ninety is one eliminating the term. This decision was made because for pulleys in a CVT more often than not the sine term will drop out because of the nature of the application.

Using equation 12 will be shown to demonstrate finding a theoretical mass moment of inertia. The values of the mass m and the radial distance R are assumed to be unity, and it should be noted that the inertial constant k is dimensionless and depends on the shape of the body being analyzed. For a solid cylinder the value of k is 0.5 [6] which will be used for this calculation.

$$I = 0.5 * 1\ lb * (1\ in)^2 = 0.5\ lb - in^2$$

Applying equation 13, the side force can be found when assuming unity for the torque T, shift ratio SR, and ramp radius r and a value of 45 degrees for alpha.

$$SF = \frac{1\ lb - ft * 1 * 12}{2 * 1\ ft * \tan 45} = 6\ lb$$

Finding the belt pressure force with equation 14 is done as follows when assuming values of unity for both the flyweight force and the spring force.

$$Fbp = 1\ lb - 1\ lb = 0\ lb$$

Next the pretension force of the spring can be found by assuming unity of all values.

$$Fpt = 1\frac{lb}{in} * (1\ in - 1\ in) = 0\ lb$$

The value of zero pounds makes sense since the installed length is the same as the free length of the spring in the example calculation.

The last equation that needs to be demonstrated is equation 16 to find the efficiency of the CVT. A value of unity is chosen for the driver HP and a value of 0.5 is chosen for driven HP.

$$\eta = \frac{0.5\ HP}{1\ HP} = 0.5$$

That result means the CVT system is 50% efficient.

4.2.2. Graphical Results

One of the most powerful tools in MSC Adams is the simulation tool which allows for the user to not only run the model so they can see it in motion, but also to plot a variety of variables for different objects in the model so the rigid body analysis can be visualized on a graph.

After the construction of the models and developing a process for how to tune a CVT with the assistance of the tools collected in this thesis as it was presented in section three, it was determined that the most useful and informative graph was an RPM vs time graph with the units of RPMs being in revolutions i.e, rotations per minute, and the units of time being how long the simulation was running in minutes. The simulations shown on the following graphs are run for a short duration of time because a constant torque of 100 lb-ft is being applied to the input shaft for this simulation and the MSC Adams simulation spins the model up to full speed in less than 15 seconds in most cases, and since there is a constant torque from the simulated engine, the behavior is constant after that so there is no need for long durations to show the full behavior.

There are three graphs provided in this section, one for each of the Adams View models created, each being an RPM vs time graph. The first graph shown as Figure 31 for the low ratio computer model called CVT_Low_Ratio shows a behavior consistent with how the CVT shown behave in this low gear ratio. The blue dotted line shows the driver and the solid red line the driven.

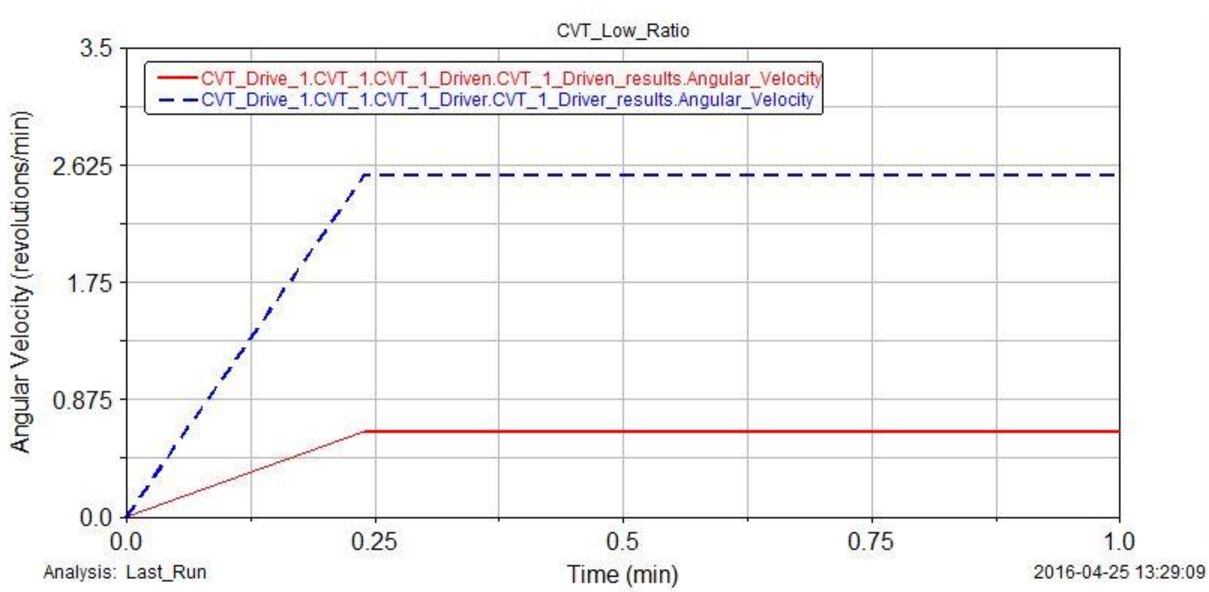

Figure 31: CVT_Low_Ratio Simulation Graph

It can be seen that in the low gear ratio the driver spins with a higher angular velocity than the driven because the driver has a much smaller diameter than the driven, meaning it needs to spin around multiple times to spin the driven one time around. This is consistent with behavior that should be seen for the relationship between driver and driven pulley in a CVT system.

The next graph created in Adams View is for the CVT_Mid model given as as Figure 32 which simulates the middle gear ratio of 1:1 that was created and shows the angular velocity of the pulleys using the same colors as Figure 31.

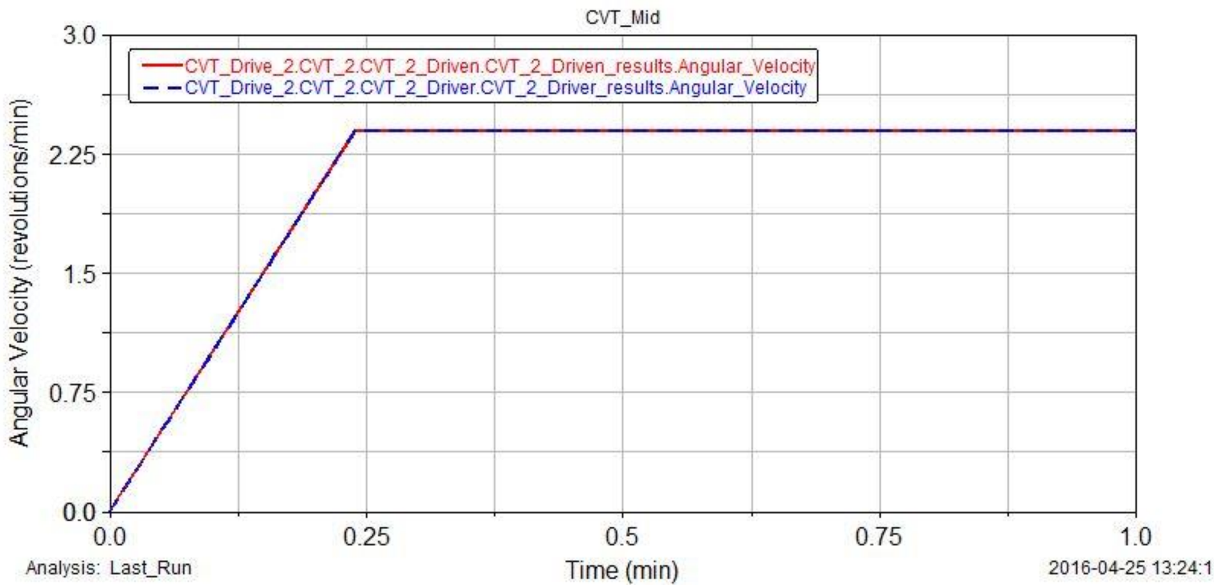

Figure 32: CVT_Mid Simulation Graph

This graph shows what is expected that when the gear ratio is 1:1 these is a 100% equivalent transfer between the primary and secondary clutches for angular velocity, and therefore torque, since the diameters of the pulleys are equal with the same angular velocity. The third graph taken from the simulation tool for the CVT_High_Ratio is shown as Figure 33 using the same color scheme as the previous two figures.

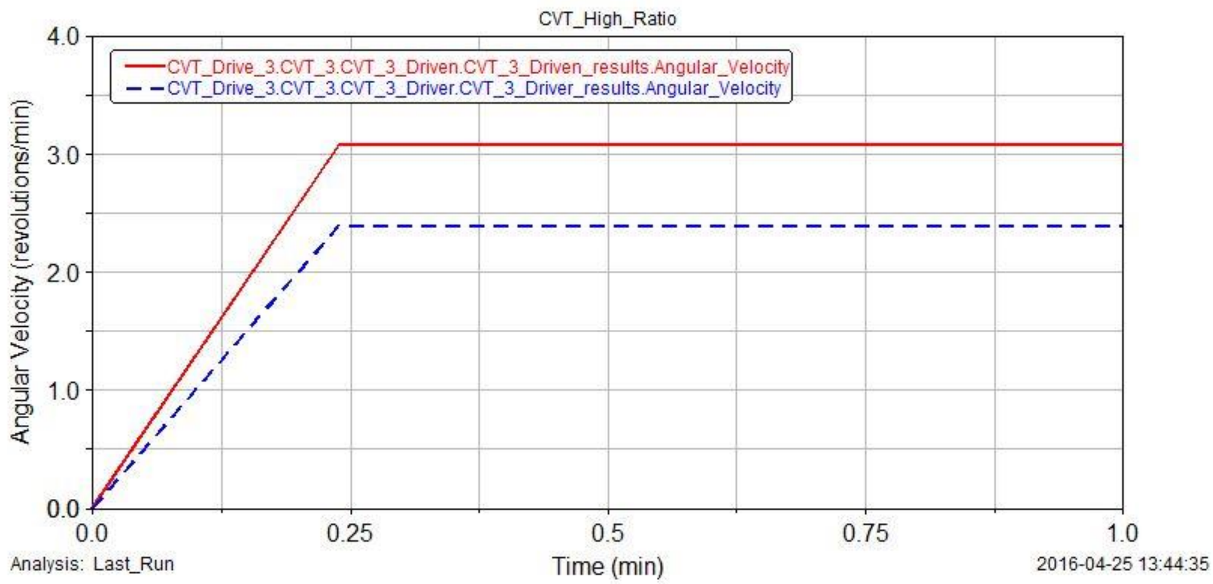

Figure 33: CVT_Hight_Ratio Simulation Graph

The behavior shown in Figure 33 corresponds to what intuitively should be the case for the CVT in high ratio, which is that the driver has now exceeded the angular velocity of the driven pulley because of the ratio of diameters (gear ratio) it has entered. Note that when the system has reached the state of maximum high gear ratio and continues to build engine speed past this point, it is often called over-drive.

All three models provide favorable results using the simulation tool to graph angular velocities of both the driver and driven pulley to compare results. These results make intuitive sense; however, a simple sample calculation is done below to verify the results for low ratio CVT model to further illustrate the validity of the model. It can be read from the graph for the low ratio shown in Figure 31 that the angular velocity of the driver pulley is 153 RPMs and 38.25 RPMs for the driven once the system has reached equilibrium for the constant applied torque. Based on the measurements taken for the

CVT used as an example for constructing the model in Adams View in this minimum low ratio position, the shift ratio has a value of 4.

$$38.25\ RPMs\ * 4 = 153\ RPMs$$

This simple calculation verifies based on the shift ratio for the known geometry used in the Adams View model that the angular velocities computed by Adams hold true to the geometric relationship between the pulleys.

4.3. Validation Discussion

4.3.1. Easy to Verify Model Parameters

There are many parameters in the Adams model that are easy to measure, these are listed as follows in a bulleted list for anyone wishing to validate this model in the future.

Easy to Verify Parameters

- Diameter of Primary (Driver) Clutch at any point in the shift range
- Diameter of the Secondary (Driven) Clutch at any point in the shift range
- Size of the belt used to connect the CVT system
- Pretension of springs in the system
- Rate of travel (rate) of springs in the system
- HP and torque generated at the crankshaft using the engine dynamometer
- HP and torque transmitted to the track using the track dynamometer

The most valuable parameters to measure when validating the model for assisting in tuning the clutch and seeing how effective this tuning process is would be to measure HP and torque at the crankshaft and then measure at the point it transmits to the track and find the efficiency. This will give an estimate of the power loss through the CVT because there are no other major factors between the engine and the track that will generate large losses of efficiency when the system is operating properly.

4.3.2. Difficult to Verify Model Parameters

As with any mechanical system, there are some parameters in the operation of a CVT that are simply difficult to measure, and therefore, difficult to verify with the model created and the methodology for tuning. These are listed as follows in a bulleted list.

Difficult to Verify Parameters

- Direct relationship between engine speed and position in shift range
- Centrifugal force of flyweights in CVT motion
- Belt pressure applied to the sheaves
- Angular velocity and torque of pulleys during operation

These values would be very useful to know but are extremely difficult to measure even with state of the art modern equipment when the CVT is in operation. Some just aren’t accessible or even visible when the system is moving. These are the types of values it is critical to be able to model in Adams or calculate using the provided equations for tuning.

4.3.3. Setup for Future Work

It is important for groups in the University to work together from year to year on long term projects. The following is information related to the validation of the MSC Adams computer models which has been brought together to allow for the future work section to best explain what could be done in the future to improve the work started in this thesis.

Future Work for Validation

- Measure and build models for CNG snowmobiles specific CVTs clutches.
- Measure CNG snowmobile HP and torque at crankshaft.
- Measure CNG snowmobile HP and torque transmitted to the track.

Future Work

In a project like the University of Maine Clean Snowmobile, which continues from year to year, it is necessary for groups to work together in order to preserve legacy work from previous years, which is critical to the most efficient continuation of the project and reducing rework for future groups. This section will outline suggestions and comments for future work teams could complete or spring board off from in relation to what has been done in this thesis.

The first suggestion for future work would be to replace the stock secondary clutch with a roller action clutch which will significantly reduce friction, or at least Teflon coat the current one as described in the tuning section in order to increase the efficiency of the secondary clutch before even beginning to tune other components of the system. This is an easy fix which will result in drastically improved results.

The second is to take a look at the suggestions for future work related to validation listed and measure the HP and torque of both the input shaft from the engine and being transmitted to the track to figure out the overall efficiency of the CVT in its current state. Also a spline file should also be taken from the engine dynamometer in order to better utilize the Adams View models for the specific parameters of the CNG engine. While getting the spline file, it is critical that the power band, and specifically the power peak, be found for the specific engine being used. These values will no longer be stock for the Arctic Cat 1100 4-Stroke two-cylinder engine because of the conversion to CNG.

The next step will be to work through the framework given in this report to properly tune the CVT system and find components that should be changed or modified accordingly in order to build a more complete picture of how to tune a CVT. The basis for numerical analysis and a start to the computer modeling has been done, but there is a lot more that could be done with further time and development.

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Author

Camerin Michael Seigars
University of Maine Mechanical Engineering Undergraduate
Honors College, Math Minor
President of UMaine SAE Clean Snowmobile Club

About the Author

A lifetime Maine resident born in Augusta, Maine in 1993, Camerin Seigars grew up in Gardiner, Maine where he attended all school prior to moving to Orono, Maine for college. He is now a senior in the Mechanical Engineering program at the University of Maine in Orono and a part of the Honors College. Camerin enjoys learning new things and spending much of his free time reading, whether it be an engineering text, something historical, a graphic novel, or an amazing piece of science fiction. He is eternally dedicated to learning, spending countless hours in college reviewing material, whether it be for a class or simply to expand his knowledge of a subject. Outside of academia, however, he enjoys backpacking through the Maine woods which he was first exposed to while working toward his achievement of Eagle Scout in the Boy Scouts of America, as well as fishing for landlocked trout and salmon, and hunting deer and water fowl. After undergraduate college he plans to continue his education at the University of Maine in the Master's program while working at the Advanced Structures and Composites Center on a project focusing on thermoplastic composites, which will be the focus of his Master's thesis.