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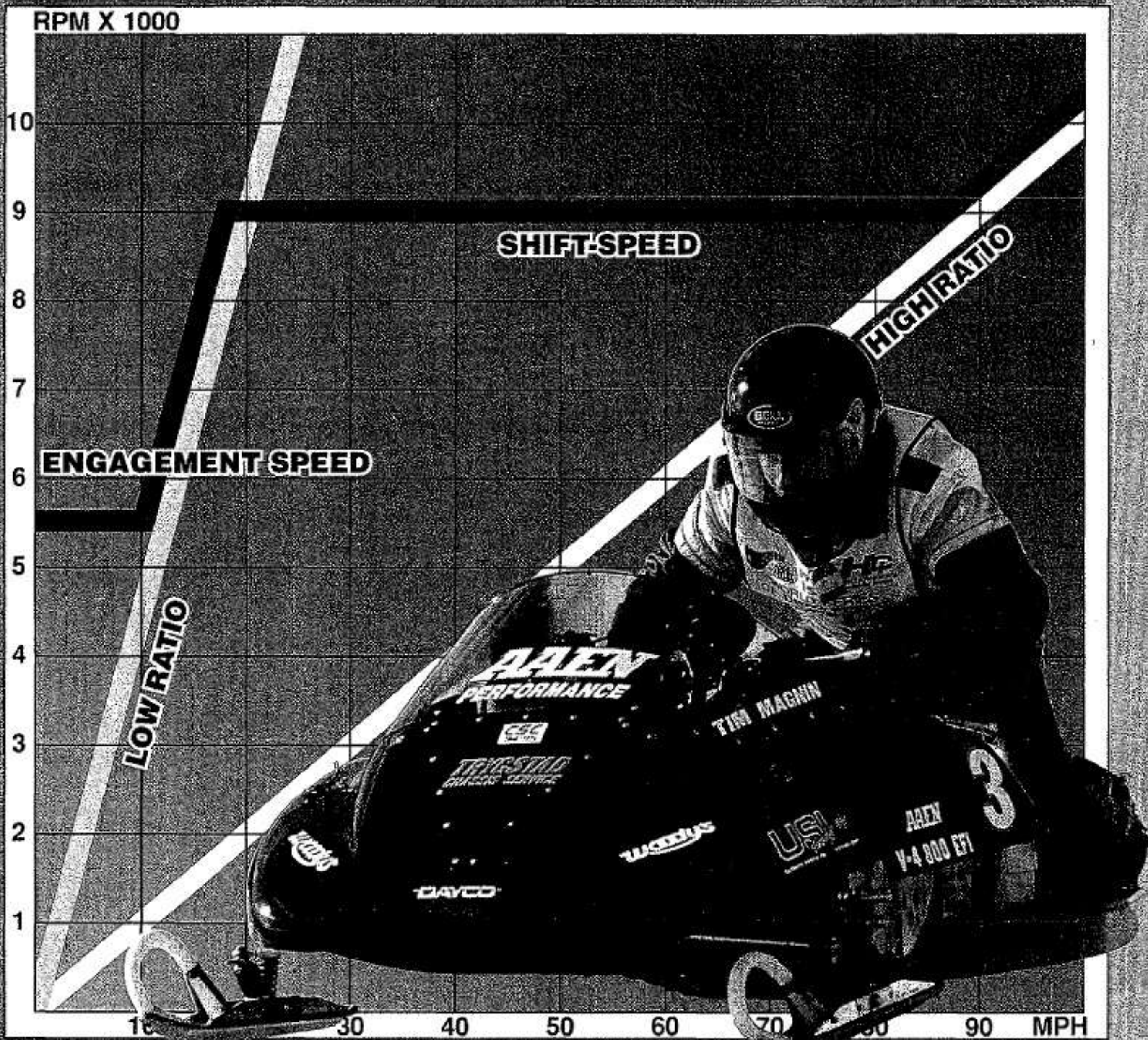
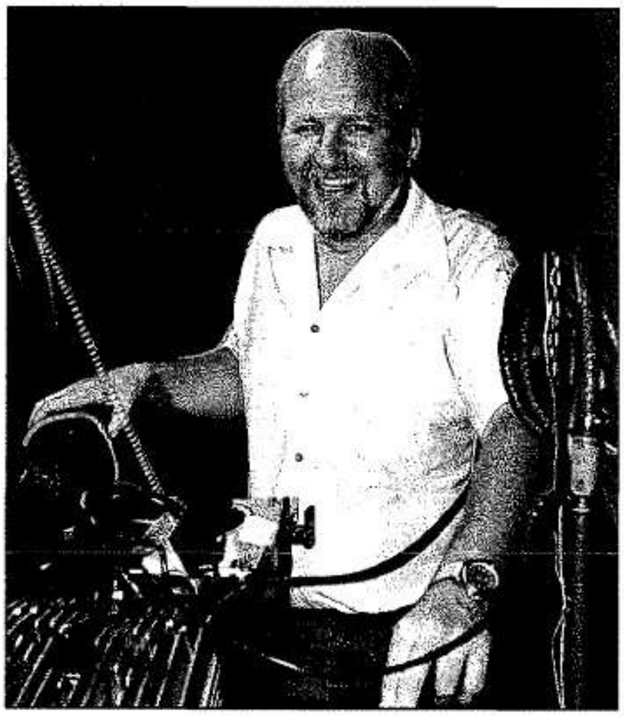
OLAV AAEN'S CLUTCH TUNING HANDBOOK



FOR SERIOUS RACERS AND ANYONE WHO WANTS MORE PERFORMANCE FROM THEIR VARIABLE RATIO BELT-TRANSMISSION.



About Our Involvement

When I first rode a snowmobile in the Black Hills in 1968, I had no idea I was to be involved with this vehicle for most of the rest of my adult working life. Looking back 38 years later it has been quite a trip. I was then a mechanical engineering student at the South Dakota School of Mines in Rapid City. After finishing my Masters at Purdue, I was recruited by OMC to work with the Evinrude and Johnson Snowmobiles, and just happend to land square in the middle of snowmobilings first "Golden" age.

The snowmobile was a crude vehicle with plenty of room for improvement, and clutches and belts were in great need of developments. Racing kept the pace of progress up, and new inventions popped up left and right. Progress was rapid and the Comet and Polaris clutches that now are the benchmark developed quickly. New and better flyweight curvatures increased power output and belt life.

During this time I worked on new clutch designs, including a unit with two selective and distinct shift patterns that could be used for trail riding or hot dogging. One of the projects was to develop a useful computer program to design and predict clutch performance. This was not completed before OMC left the snowmobile industry in 1975, but a lot of the engineering research has been carried forward and is presented in this book.

When we started Aaen Performance in 1975, it was clear that there was a need for information on clutch tuning, and our original Clutch Tuning Book was a small type written book produced on a copy machine. Since then we have gone through lots of new, updated and expanded editions of this book. You are reading the ninth edition of this handbook which has now been published for 31 years. Today, AAEN Performance is one of the major manufacturers of high performance snowmobile products.

The variable speed belt transmission is an integral part of the modern snowmobile. Most performance increases necessitate a recalibration of the transmission, and continuous clutch development is therefore a natural part of the business. AAEN Performance also distributes a full line of Comet and Polaris clutches as well as all tuning components and tools. With extensive and continuous experience in all design aspects of the variable speed belt transmission, we have contributed many articles on clutch tuning to the snowmobile press. We also publish the "Carb Tuning Handbook" for Mikuni Carbs.

AAEN Performance sponsors several snowmobile race teams in Oval, SnoCross, Drag Racing and Speed Runs along with F-500 cars to test the latest ideas in transmission design.

Our latest "test bed" is a "D Sports" race car running on the SCCA circuit with a continuously variable ratio belt transmission. With a 200 HP engine in a 1,000 lb. car, the belt transmission is tested to its limits. This book represents our 37 years of engineering design and development experience with the variable ratio belt drive transmission.

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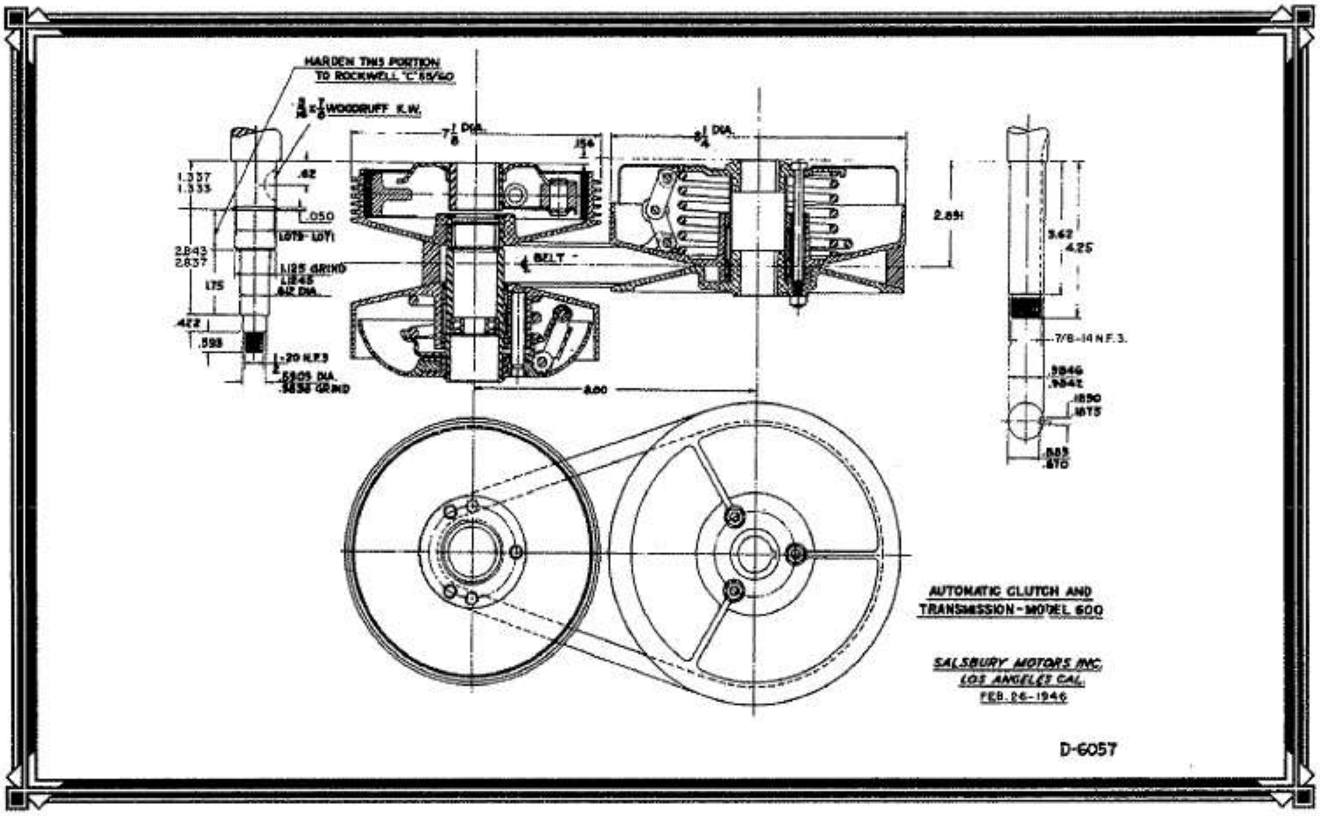
Revised 1989, 1993, 1995, 1996, Nov. 98, Dec. 01, June 06

Thanks to Terri Aaen, Elana Tarwid and Jeanette Seiberlich

The author testing theory against reality.



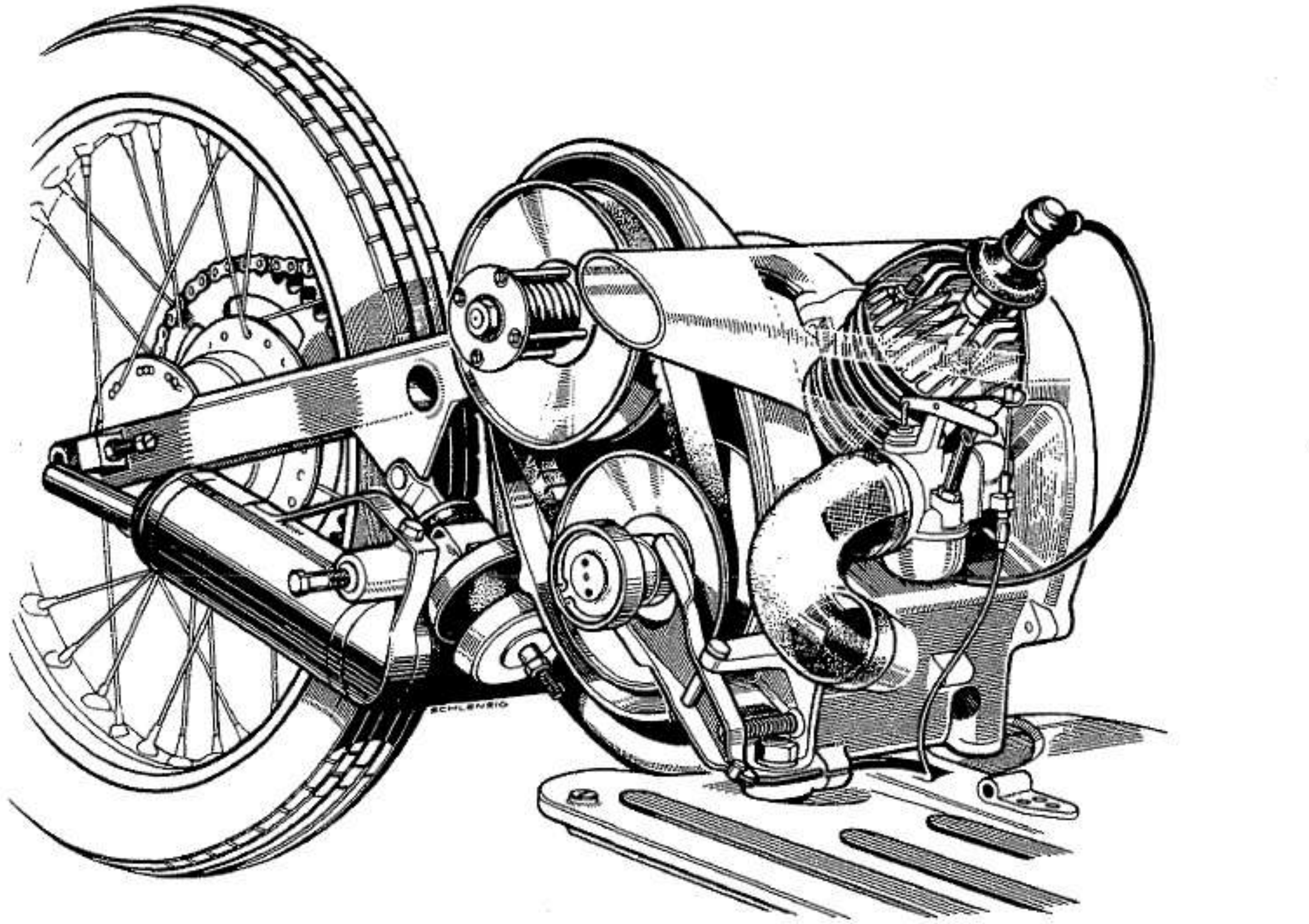
The first known belt transmission to be offered on a commercial snowmobile was this Salsbury model 600 found on the Eliason "Kitchener Series" K-10 in 1951. This first clutch had the belt under constant pressure, and the engagement was performed by a centrifugal clutch. Instead of a helix cam in the secondary, side forces were controlled by a spring and a set of flyweights.

INTRODUCTION

The continuously variable transmission (CVT) plays a most important part in the performance of you racer. It is the vital link in a vehicle that constantly changes speed powered by an engine which ideally should operated at a constant speed. With the narrow power band of a modern two-stroke racing engine it is important that the engine is kept on the power peak and the power transmitted in the most efficient manner for maximum performance.

The modern V-belt transmission is, in spite of its mechanical simplicity, controlled by a number of interdependent variables. It is only by matching these variables that the optimum performance is obtained from the vehicle. The purpose of this handbook is to explain the function of the transmission and the variables that influence its performance and efficiency, and to give you a testing procedure that will enable you to match your transmission to your racer's requirements. There are very few things the factory racers do that you could not do yourself with a good tachometer and a good testing procedure. The two ingredients necessary to obtain an efficiently matched clutch are an understanding of the mechanism and plenty of testing.

In this ninth and expanded edition we will go into further detail in the technical theory areas than in previous editions. If you are in a hurry and don't have the time to read all the chapters, we have a new appendix with basic tuning information on pages 75 & 76. As you get further into the tuning of clutches and start looking for those extra percentages that make a winner, the theory section will make more sense. We are going into extensive detail in tuning popular clutches in the appendix and this should give the answers to most of the important questions relating to these models.



Belt drive CVT transmissions were also used in scooters in the early fifties. This is a DKW "hobby" scooter showing a cable operated belt release (Audi Historical)

CHAPTER I

History

Variable Speed Belt Transmissions: How They Developed

A designer's dream: "Smooth uninterrupted power delivery automatically adjusting itself to any load conditions." A transmission accomplishing this would be a dream come true on most vehicles. It has already been a reality on snowmobiles for 40 years.

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.

The variable ratio snowmobile belt transmission has achieved a high level of technical sophistication that is far from the sometimes crude devices used in snowmobiling's infancy.

Variable ratio belt transmissions were an integral part of the snowmobile's success. Snowmobiles encounter high-drag load conditions where the use of a gear box is impractical. Interrupting the power delivery to change ratios would result in the machine losing too much momentum and getting stuck. In the early days there were few groomed trails and the snowmobilers were faced with breaking their own trails most of the time. Incorporating a variable ratio belt transmission made the machine easy to operate and practical in the snow.

In the early stages, manufacturers were mostly interested in whether the transmission would work at all.

The first transmissions came directly from industrial applications as did the engines which powered the machines. 20 HP meant a high performance, top of the line machine and designers were happy they had found a solution to uninterrupted power delivery.

Early transmissions were all simple in design and operation. The driven clutch consisted of two stamped steel sheaves with the fixed sheave often duplicating in function as a brake disk. A straight pressure spring pushed against the movable sheave and controlled the belt movement.

The driving clutch also consisted of two stamped steel sheaves, a fixed and a movable. A pressure spring pushed the movable sheave away from the belt, and a centrifugal mechanism worked against the spring. This centrifugal mechanism first had to overcome the pretension of the pressure spring to engage the belt, and then rev higher until it overcame the spring in the driven sheaves and started to shift the belt into higher ratios. The spring in the engine clutch controls the engagement speed. The spring in the driven clutch controls the shift speed which should coincide with the engine's power peak for best performance.

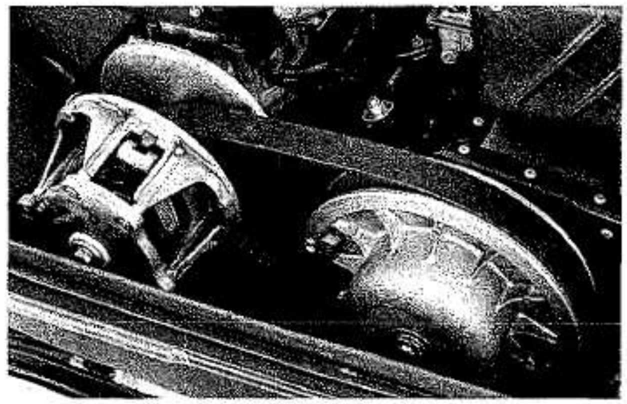
Early centrifugal weight systems included balls, shoes or garter springs working against the outside cover, or the more "sophisticated" kidney-shaped flyweights. The movable sheave usually traveled on a greased spline or on square or hex bushings.

Problems started to occur with more power and higher engine speeds. At 25 HP it seemed like belts just disintegrated. Belt design was in its infancy and while the belt manufacturers set out to find better materials and designs, engineers worked on the transmissions to find a solution to the problem.

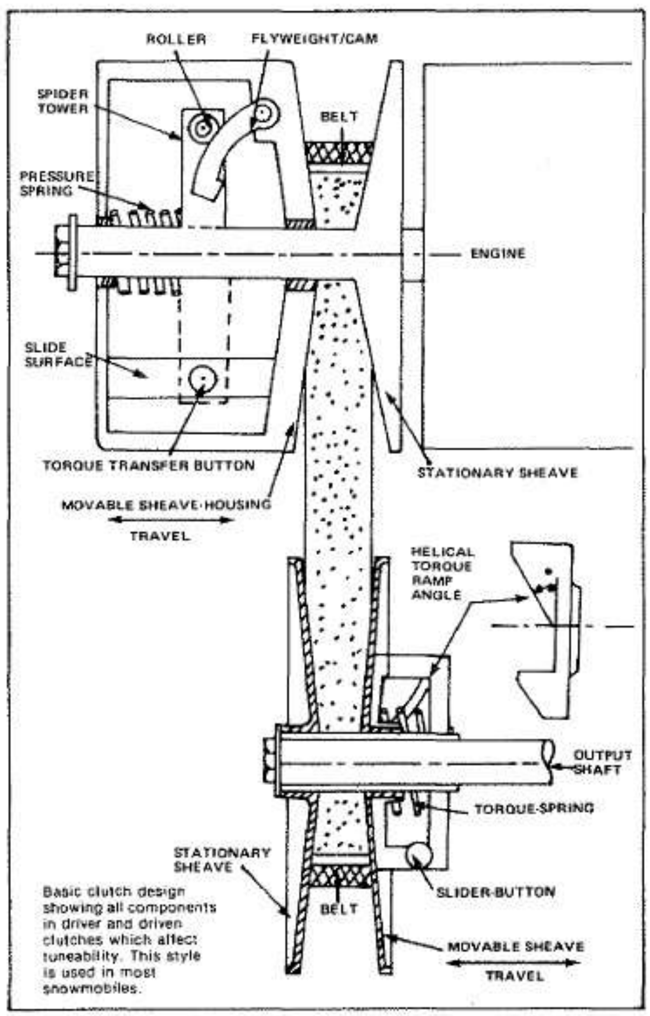
When analyzing the driven clutch, engineers found that the pressure spring by itself actually put undue stress on the belt. The pressure spring had to be set for this high torque condition, otherwise the belt would spin on takeoff or in heavy snow conditions. The spring got tighter as the movable sheave shifted out against it, and the belt became subject to loads many times more than required.

The solution was to install a weaker spring and complement it with a torque feedback cam. When high torque was transmitted, side forces were high. In 1:1 ratio where the torque was less, less side force was applied on the belt by the cam.

Belt life and back-shifting improved dramatically with this design, but it also opened up new design problems in the front clutch.



The modern snowmobile transmission as we know it today. Open-faced die cast aluminum primary clutch, driving an aluminum secondary clutch via Kevlar reinforced wide belt.



Basic clutch design showing all components in driver and driven clutches which affect tuneability. This style is used in most snowmobiles.

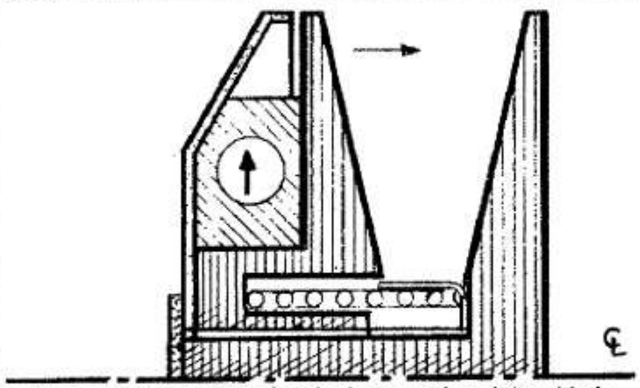


Fig. 1 - Representative of early design with a sliding block working against the cover and movable sheave. This design had high friction and limited tunability.

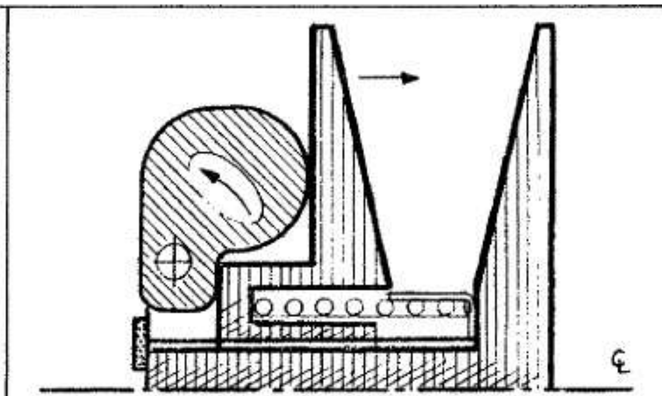


Fig. 2 - The Kidney Weight Clutch reduced the friction and introduced a flyweight system, but there was no cam to tune and the movable sheave still had its torque transferred through a spline.

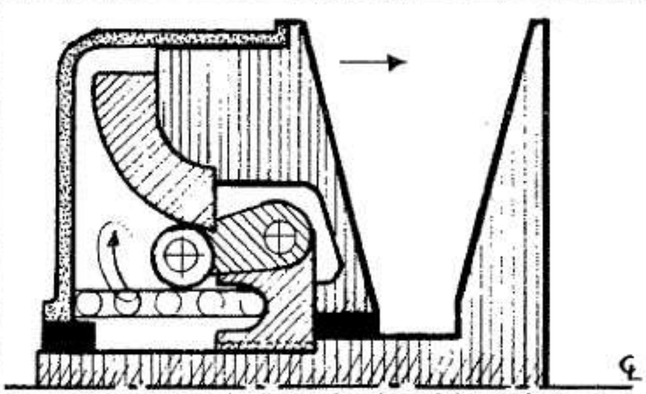


Fig. 3 - Arctic's hex clutch introduced tunability with separate flyweight and roller. Torque transfer was improved by use of a hex fiber bushing instead of splines.

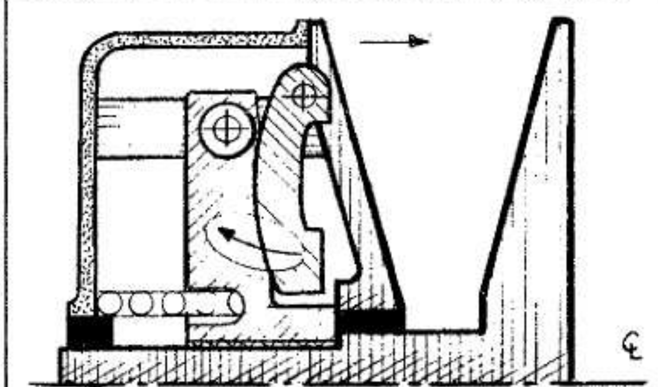


Fig. 4 - Today's state-of-the-art with flyweight cam against spider mounted roller, and torque transfer point moved out to the spider tower.

A completely new centrifugal mechanism would be necessary to take full advantage of the new driven clutch torque feedback cam.

A new generation of front clutches started to arrive. They had many improvements in common. The sheaves were cast in aluminum to reduce rotating weight and remove heat more efficiently from the belt surface. To improve back shifting response, the splines on the movable sheave were eliminated and sliding surfaces were instead provided as far out toward the outside diameter as possible. The further away from the center the sliding surface is located, the less force is required on the surface to transfer the torque as the sheave is moving. When the force is reduced, the friction is reduced and the clutch can respond quicker to load changes. New high performance clutches all have the sliding surfaces located on the outer surface usually as buttons or sliders working on the spider towers.

The largest improvements came in the design of the centrifugal mechanism. To match the belt force of the engine clutch to the new torque feedback driven clutch, a cam surface was needed that could be designed with curvatures to deliver the correct side force.

Several designs emerged and they all had a roller working against a cam surface. The most common design used today by Polaris, Comet and Yamaha has a flyweight with the curvature as part of it, working against a roller.

This design has proven most successful for efficient power delivery. The movement itself gives the approximate side force requirement, and only a slight curvature is necessary to give the correct side force for most engines. By changing the curvature in midrange and engagement area, flyweights can be tailored to any combination of horsepower and vehicle weight.

The Arctic hex clutch and the John Deere TR800 both had the same flyweight setup. The flyweight itself consisted of two arms with the roller bolted in between. Different weights could be added to the roller assembly. The roller assembly pivoted and worked against a stationary cam. Weight could be adjusted independently of the cam curvature and made fine tuning easy. Drag racers loved this clutch as it was easy to set up accurately. Arctic's drawback was relying on the hex bushing to transfer torque. It quickly wore out and started binding up during prolonged use on performance machines. John Deere's TR800 is an improved version of the Arctic. It has the same flyweight system, but the hex bushing has been replaced with torque transfer buttons on the spider towers. John Deere's engineers also paid close attention to cooling by designing a number of

Snowmobile racing in all forms speeded up clutch development. Racing requires clutches that can respond quickly to changing load conditions.

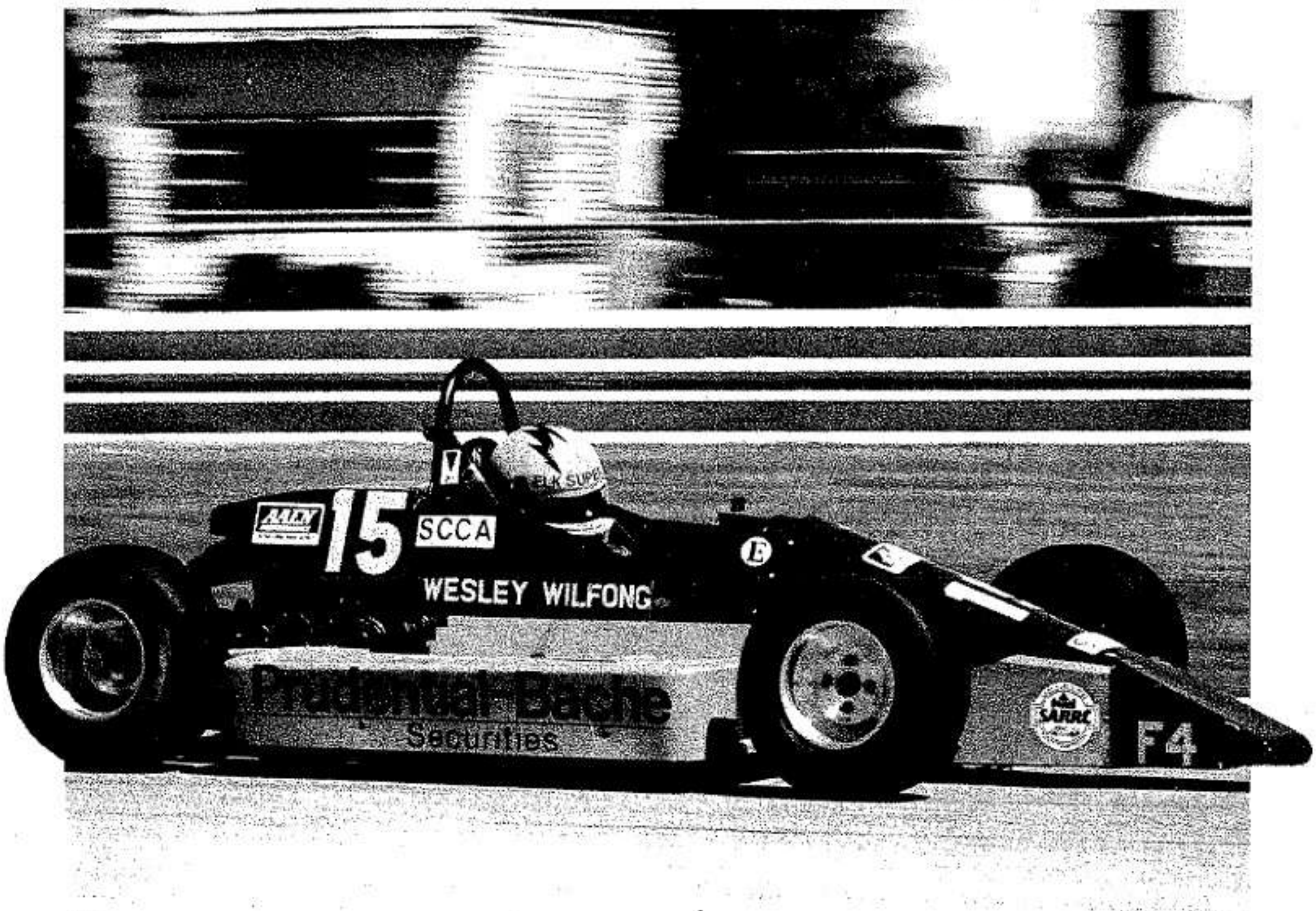
holes and lugs in the sheave that acted to cool the belt.

The variable speed belt transmission is now best known as "the snowmobile clutch," due to its universal acceptance on snowmobiles. Snowmobiling as we know it would have been impossible without this simple but efficient transmission delivering uninterrupted power to the track. Clutches and belts have gone through a tremendous development in 37 years of snowmobiling, and racing has been one of the main catalysts to rapid development. "Getting it to the ground" and "it's all in the clutches" are two terms often used by racers to emphasize the importance of a well-tuned and efficient transmission.

Variable speed belt transmissions are also showing up in motorcycles, ATV and F-500 race cars. Rokon was the first to use a belt setup on a motorcycle, and although the transmission was a success, the bike is no longer produced. F-500 or mini-Indy cars have been around for over 20 years, first on the ovals in the midwest, and now as a highly successful national class in SCCA road racing. Honda uses a belt transmission on their Odyssey Dirt Buggey. Polaris introduced its new 3 and 4 wheel ATV's with a totally enclosed transmission using an ingenious air circulation system to keep the clutches cool but the dust and mud out.

The simple design, high efficiency, low cost and easy operation of the variable speed belt-transmission is making it increasingly popular for numerous new applications. Whatever the application, the basics are the same for best efficiency and good belt life. Understanding how it all works will give you better use out of your machine.

One of the fastest growing classes in SCCA Road Racing is Formula 500 which uses a snowmobile engine and a continuously variable belt transmission (CVT).



Speed and Power: This hill-climb racer needs both to make it up the hill.

CHAPTER 2

Speed and Power

SPEED DIAGRAMS

Speed diagrams are important in understanding what happens to the transmission in your vehicle. In order to grasp what's going on, you have to be thoroughly familiar with the speed diagram, and be able to design your own.

By knowing what your engine power curve is, you can design the ideal shift curve diagram for your engine, transmission, and gearing. By watching your tach, you can determine how far from this ideal picture the transmission is actually performing. By comparing the ideal diagram with the actual diagram, you can then make intelligent decisions about choice of springs and flyweights to achieve the ideal shift curve. These are the basics of clutch tuning, and we will constantly refer to the ideal shift curve as we explain the effect of changing components in the transmission.

You only have two reference points when you work with a transmission. Unlike a car transmission where you would know what gear you are in, you have to compare engine speed (RPM) and vehicle speed (MPH) with your gearing to know which ratio you are in.

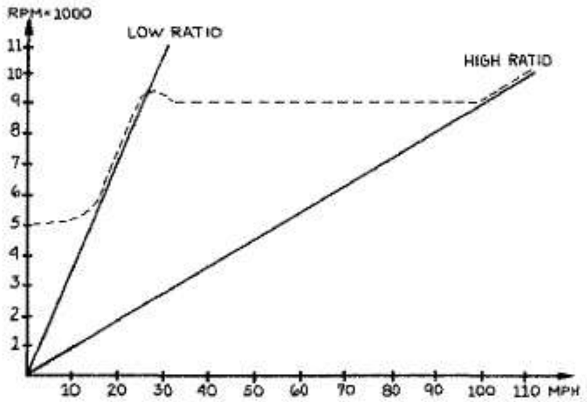


Fig. 1 - Speed diagram of variable speed transmission. The dotted line represents the engine speed as the vehicle speed increases. The big advantage of the snowmobile transmission is its ability to hold the engine RPM on the power curve while it continually changes ratios.

A speed diagram (Fig. 1) has the engine speed (RPM) on the vertical axis and the vehicle speed (MPH) on the horizontal axis. Two lines at different angles start at the zero point. These lines represent low and high ratio of your transmission. The steepest line represents the low ratio where the belt is all the way into the engine driving clutch and all the way out on the driven clutch. The low ratio is usually close to 3:1.

To the right of the low ratio is a second line at a flatter angle. This line represents the high ratio which occurs when the belt is shifted all the way out on the front driving clutch, and all the way in on the rear driven clutch. High ratio is usually in overdrive close to .75:1. The overall shift ratio of this variable belt transmission is the low ratio divided by the high ratio (3/.75 = 4) which gives us 4. An overall shift ratio of 4 is typical of most transmissions presently in use.

The dotted line represents the ideal shift curve of an engine with a power peak at 9000 RPM. Two points are of importance, the engagement speed and the shift speed.

ENGAGEMENT SPEED

In order to start the vehicle moving, the engine must be engaged at a speed that has sufficient power to start accelerating the machine.

Engagement speeds are dependent on a number of variables which will be discussed more extensively later. With a power peak at 9000, our sample engine can be comfortably engaged at 5000 RPM. To obtain a 5000 RPM engagement speed, the flyweights have to overcome the pretension in the driving clutch and start moving the sheaves together until enough force is exerted on the belt to start the vehicle moving. As the force increases, the belt pressure will go from zero to 100% of the required force to transfer the torque. This is called the "clutching phase."

When the belt is fully engaged the speed will continue to increase in low ratio until 9000 RPM is reached. This is called the shift point.

SHIFT SPEED

Our engine is developing maximum horsepower at 9000 RPM, and we want to hold the engine at the power peak as the transmission is shifting out and the vehicle speed is increasing. When the engine reaches 9000 RPM, enough force has been built up in the flyweights to overcome the belt tension provided by the driven clutch. The belt will now start moving into the driven clutch and outward on the engine clutch.

The sole purpose of clutch tuning is to have the flyweights and springs matched to the belt pressure of the driven clutch in such a way that the engine is held at its power peak (9000 RPM) all the way from low ratio to high ratio. This will give us the maximum power delivery. When the transmission is all the way shifted out, and vehicle speed still increases, the engine speed will increase following the high ratio line.

CONVENTIONAL 4 SPEED TRANSMISSION

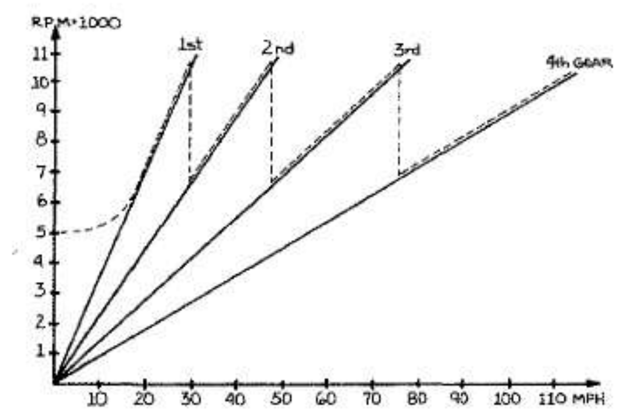


Fig. 2 - Speed diagram of a four speed conventional transmission. The dotted line represents the speed variations of the engine as the vehicle is accelerated from first to fourth gear.

A conventional 4 speed transmission would have a different speed diagram. Low ratio would correspond to first gear, and the high ratio would be 4th gear. In between the two lines we would have the 2nd and 3rd gear ratios.

As the clutch is released in 1st gear and the vehicle starts moving, the RPM is climbing along the 1st gear ratio line.

The driver must rev the engine to 10,500 RPM and then shift into 2nd gear. When he shifts the engine speed will drop to 6000 RPM and then climb up the 2nd gear ratio line. At 10,500 he shifts into

3rd gear and the speed drops to 6500 RPM. In 3rd gear the speed builds slower along the 3rd gear ratio until the vehicle again is shifted into 4th gear at 10,500 RPM, and the speed then drops down to 7000 RPM. With 4th gear engaged the vehicle continues to slowly accelerate until top speed is reached in 4th gear.

When comparing the 4 speed gear box to the variable speed transmission a couple of things become apparent. With the 4 speed box, very little time is spent at 9000 RPM where the power is. With the variable speed transmission, all the time is spent at the power peak from low ratio to high ratio. Efficiency of a belt drive and a gear box is about the same, and it becomes quite apparent that a lot more power is delivered to the ground with the variable speed transmission. By following the power curve in fig. 4, and plotting it with the 4 speed and variable drive transmission the difference in power delivery can be easily seen.

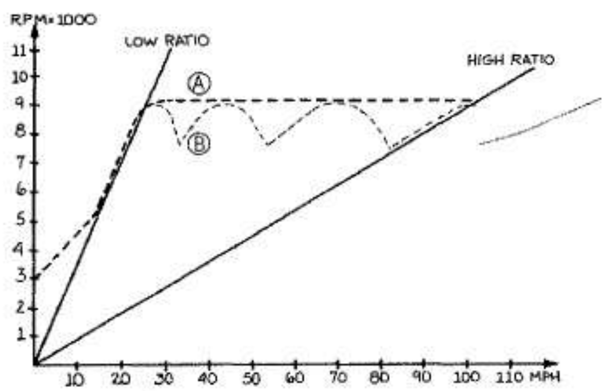


Fig. 3 - Power delivery of a variable speed transmission compared to a four speed gear box. Line A shows the constant delivery of power from the belt transmission, while line B represents the power available to the vehicle through a four speed gear box. Given equal power and weight ratios, the variable belt transmission will easily outperform the conventional gear box.

Our purpose in tuning is to keep the transmission shifting "straight" at the power peak and deliver this power as efficiently as possible through the driven clutch.

POWER CURVES

The practical shape of a power curve is dependent on the transmission used in the vehicle. If we look back at our examples using a 4 speed gear box compared to a variable speed transmission, the width of the power band becomes important. With the 4 speed gear box a wide flat power band with good torque from 6000 RPM to 10,500 is necessary to make the vehicle work. If we had a 6 speed gear box we could get away with a somewhat weaker power band that worked well from 8000 RPM to 10,500 RPM. As we add more gears the requirements become narrower. In the case of variable speed transmissions, there are no defined steps as in a gear box. A higher output, narrower power band can be used.

With the transmission holding the engine at the 9000 RPM power peak, the only limitation on the power band is what makes the clutch perform. A certain spread between the torque peak and the horsepower peak is necessary to make the transmission perform well. The spread between the peaks has a "self adjusting" purpose. As the vehicle encounters resistance in the form of the hard cornering, hill climbing, loose snow, etc., the tendency is for the RPM to drop. If the engine speed drops and the torque decreases at the same time, there will be no torque feed back to the driven clutch to help backshifting, and the transmission drops off the power band and will take some time to recover.

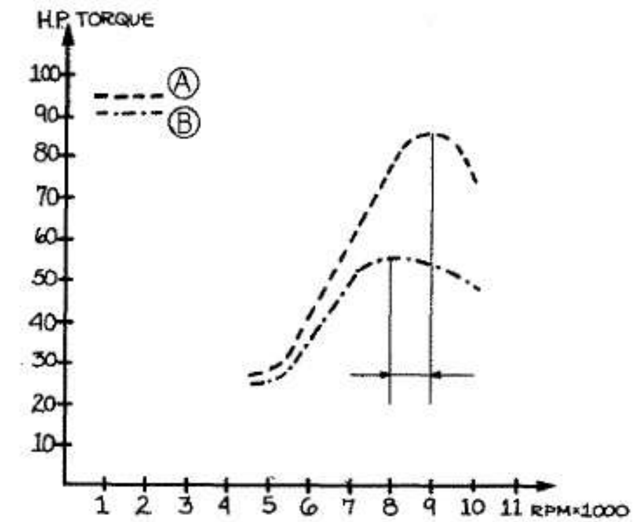


Fig. 4 - Horsepower and torque of a snowmobile engine have to meet certain criteria to work well with the transmission. The torque should peak as much as 1000 RPM earlier than the horsepower to give good torque feed back for back shifting when the RPM drops. Line A represents horsepower and line B represents torque.

If the RPM drops but the torque increases at the same time, the increased torque feed back will help the clutches to back shift and bring the RPM back up to the peak of the power band. This spread in the relationship between the torque peak and horsepower peak may vary from 250 RPM on a high strung racer on flat tracks, to over 1000 RPM on a heavy trail machine which runs in deep snow and also has to compensate for belt wear. Trail machines must also have good low end power for an engagement speed in the 4000-5000 RPM range, while a racer can engage at 6000 RPM or over. What you can get away with depends on the use, and this will be discussed in greater detail in later chapters.

To tune your clutches, you have to know where

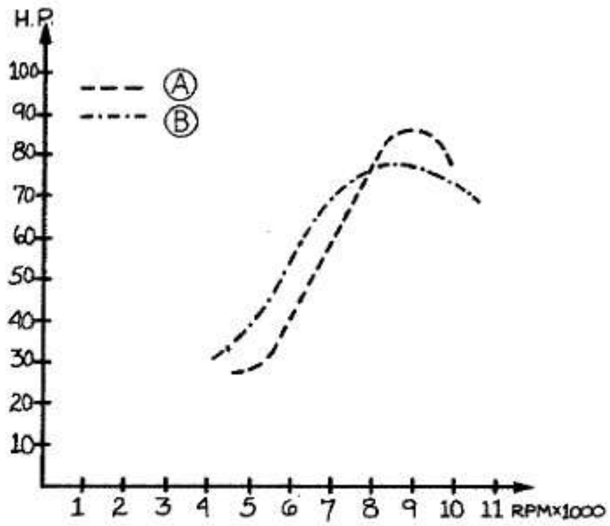


Fig. 5 - Because the variable speed transmission has no spacing between ratios, it can use a peakier power curve than an engine powering a four speed gear box. Line A represents a usable power curve for a belt transmission, while line B shows the curve needed for a gear box. For engines of equal displacement, more power can be used with a variable speed transmission.

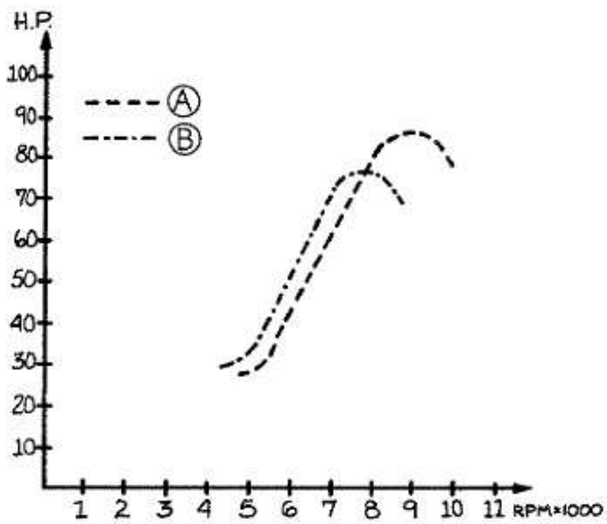


Fig. 6 - When an engine has been modified to develop more power at higher engine speeds, the whole power curve moves over. Line A represents the modified engine, while line B represents the standard engine. At 8000 RPM they both have the same power. Unless the clutch is recalibrated to run at 9000 RPM to take advantage of the new power curve, you would probably only get equal or worse performance from your machine.

the power peak of your engine is. You must also have a reliable tach for reference while testing.

When the engine has been modified, it is important that you know how the power curve has been changed so that you can tune to the new power peak. If the engine used to produce power at 8000 RPM, and you modified it with porting, pipes and bigger carbs to pull more power at 9000 RPM, the complete clutch set up must be changed to work with the new power curve. If the transmission is not shifting at the new 9000 RPM peak, all the money you spent on modifications means nothing. With the old flyweights and springs in the clutch you may end up with worse performance than your original motor. You are falling way short of the new power peak and you are also on the down side of the torque curve which makes it hard to do any back shifting.

Any internal combustion engine is an air pump, and the amount of power you obtain is directly proportional to the weight of air and gas you can burn in a given time. The density of the air has a direct effect on the power output. Air density is controlled by pressure, temperature and humidity. Pressure and temperature have the largest influence on your power. When you operate at higher altitudes the air gets thinner and your engine loses power. When the temperature goes up, the engine loses power. An engine will require heavier weights, in the driver, on a cold winter's day at sea level. On a warm spring ride in the mountains, lighter weights are required to keep the engine speed up.

Know your power curve and its limitations when you work on your clutch. Your job as a clutch tuner is to match the clutch to your engine curve and obtain the best possible performance for the conditions your vehicle is experiencing, whether it is racing or trail riding.

CHAPTER 3

Drive Ratio and Efficiency

DRIVE RATIO

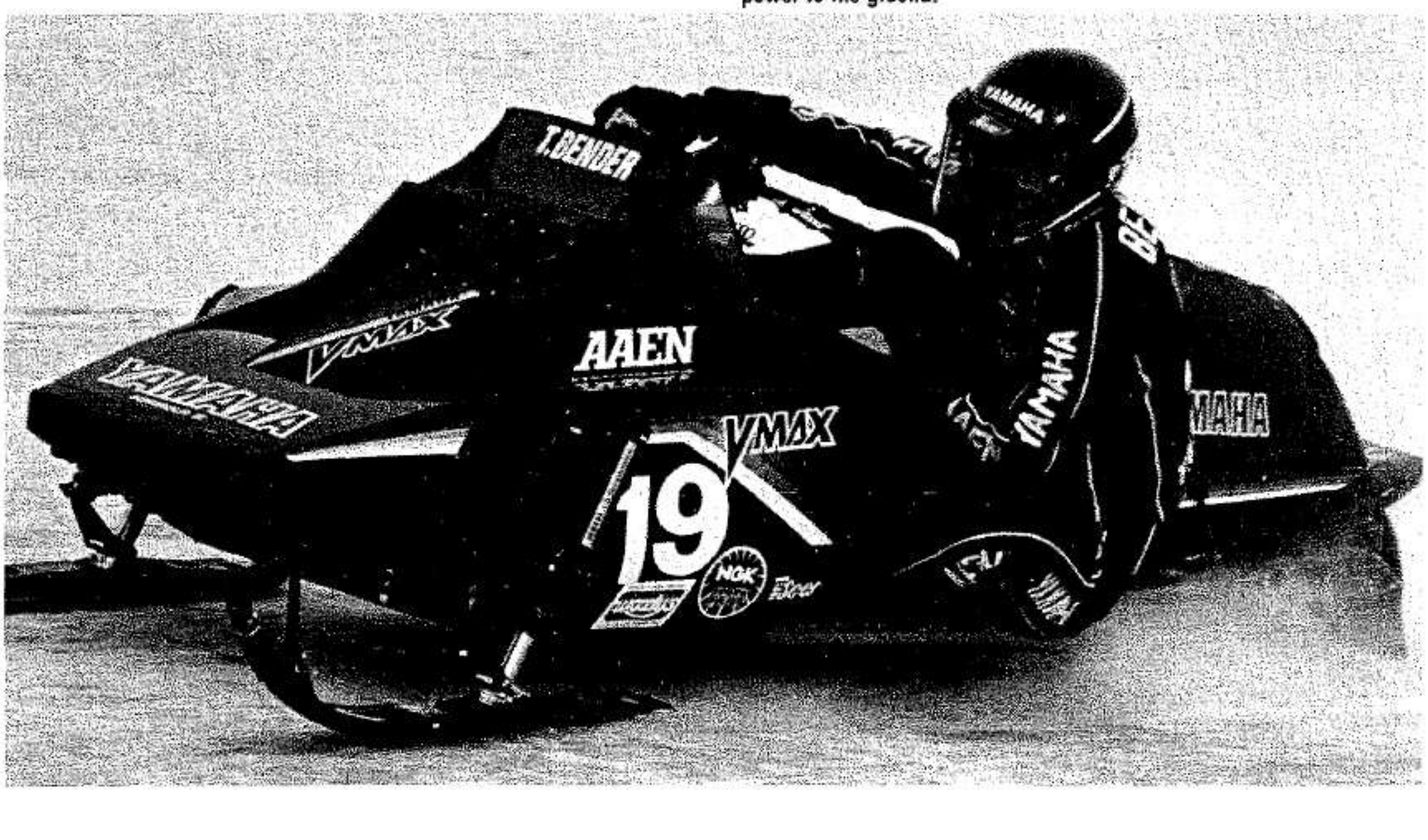
The drive ratio of the transmission is determined by the turning radiuses of the belt on the driver and driven clutch. This radius is measured at the pitch radius of the belt (Fig. 5). To obtain the ratio, the driven pitch radius is divided by the pitch radius of the driving clutch. The example below shows a clutch with an overall shift radio of 4:1.

	(A) (Secondary) Driven Radius	(B) (Primary) Driving Radius	(A/B) Ratio
Low Ratio	4.5″	1.5″	3:1
High Ratio	3″	4″	.75:1 overdrive

Overall shift ratio is arrived at by dividing the low ratio by the overdrive ratio. In this case the low ratio of 3 is divided by .75 to give us the 4:1 overall drive ratio.

Center distance and belt tension have an effect on the overall ratio. A belt that is too loose would ride too far out in the driver, reducing the overall ratio. A belt that is too tight would run further in on the driven, again reducing the overall ratio. The belt being too tight is not common since the machine would creep as a result. A third and worse combination is a worn belt. It sits further in on the secondary, and as a result also rides further out

Efficiency is the name of the game in racing. Knowing how to get maximum efficiency from your transmission puts more power to the ground.



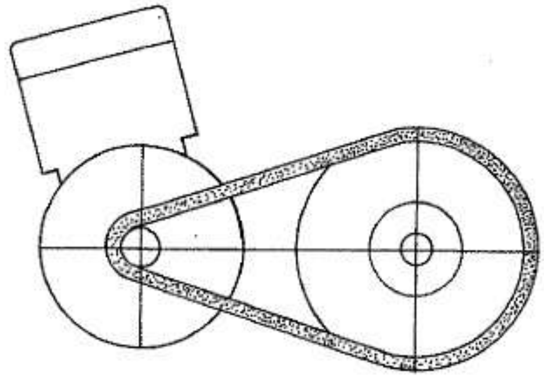


Fig. 1 - Low Ratio - The belt is at the smallest radius on the primary driving clutch, and at the largest radius on the secondary driven clutch. Total speed reduction ratio is 3:1 giving three times as much torque at the driven clutch.

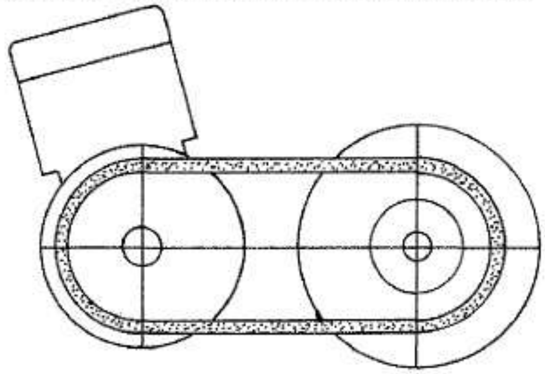


Fig. 2 - 1:1 - The belt is at equal radius on both clutches making the driven spin at the same speed as the engine.

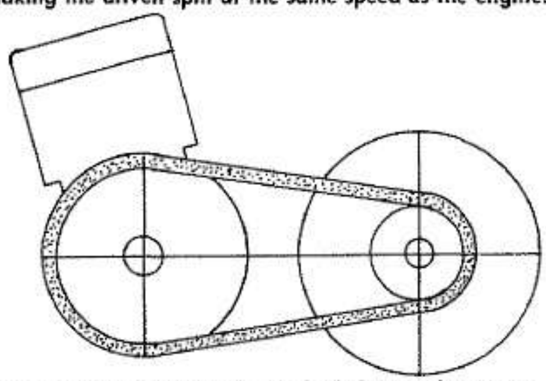


Fig. 3 - High Ratio Overdrive - The belt is at the largest radius on the primary driving clutch, and at the smallest radius on the secondary driven clutch. The speed ratio is .75:1 which means the secondary driven clutch is spinning faster than the primary driving clutch giving what is often referred to as an overdrive ratio.

on the primary. This usually creates a bog on take off and also changes the shift speed.

DRIVER VS. DRIVEN

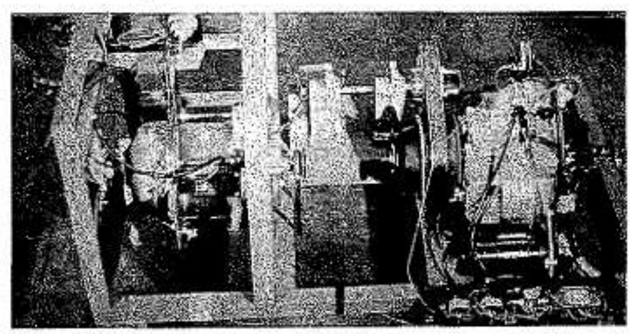
The driver and driven clutch have distinctly different jobs to do. To understand variable belt transmissions, and to be effective in tuning them, it is important to know what these job functions are.

THE DRIVER (PRIMARY)

The driving clutch has one main purpose: to control the engine speed in all shift ratios. A pressure spring and centrifugal mechanism work against each other to give the engagement speed, and the flyweights have to overcome the driven clutch pressure to maintain the shift speed. All tuning to obtain the desired engine speed should be done with the driving clutch, by modifying flyweights or changing springs. Keeping the engine on the power curve and pulling hard is the main purpose of the driving clutch.

THE DRIVEN (SECONDARY)

The driven clutch has one main purpose: to provide enough side pressure on the belt to allow the power to be transmitted. Belt pressure has a lot to do with power loss; the more pressure required the more power lost. Side pressure is therefore an important factor in determining overall efficiency of the drive system.



This portable dynamometer was built to measure efficiency of belt transmissions. The engine could run the dyno brake direct or through the transmission. A special load unit with low water pressure requirements made it possible to transport the dyno to mountain lodges for high altitude testing.

Losses are due to stretching and bending of the belt, and belt friction heating up the sheaves. When you are setting up the driven clutch you are interested in getting enough pressure to prevent the belt from slipping, and making the transmission backshift when it hits a hill or deep snow. The only reason to tighten up the driven clutch, if the belt is not spinning, is to improve back shifting. As we shall see in the next chapter, tuning on the rear by tightening up the spring also causes you to lose some efficiency.

THE MAIN RULE OF CLUTCH TUNING IS: IF YOU WANT TO CHANGE ENGINE SPEED, WORK ON THE DRIVING CLUTCH. IF YOU WANT TO IMPROVE EFFICIENCY OR BACKSHIFTING ADJUST THE DRIVEN CLUTCH.

DRIVE EFFICIENCY

High efficiency is free horsepower. Lots of power in the engine is of little use if you lose it on the way. Efficiency is defined as the power output from the driven clutch divided by the power input to the driving clutch.

$$\textbf{Efficiency} = \frac{\textbf{Driven HP (Secondary)}}{\textbf{Driver HP (Primary)}}$$

We should be careful to note that we are not talking about the maximum horsepower of the engine, but rather what is delivered to the driving clutch. Being off the power curve should not be confused with low efficiency. Tuning the RPM so that you are pulling on the power peak is one problem that we will deal with when we cover driving clutches. Once that power is delivered to the driving clutch, we are interested in minimizing the power loss by the belt as it shifts out in the transmission.

To understand how power loss occurs, it is important to know what the belt goes through on its way to transfer the power from the driver to the driven.

Belt loss is influenced by turning radius, belt speed, belt tension and friction loss. The tighter the belt has to turn, the higher the loss of power. The higher the belt speed the more often it has to be bent around the sheaves and the higher the belt tension becomes due to centrifugal force. Belt speed is determined by both the engine RPM and the radius. The higher the RPM and the further out on the sheave it runs, the higher the loss from centrifugal stretching. High driven sheave pressure results in high belt tension as the driving clutch tries to overcome the pressure to shift out the belt. The belt is in tension as the engine clutch pulls the belt to power the driven clutch. On the opposite side the belt is slack as it moves from the driving clutch back to the driven clutch. The belt is stretched and unloaded every revolution, and it is compressed by side forces twice per revolution. As the belt goes from unloaded to stretched and from stretched back to unloaded again it actually creeps along on the sheaves, and friction loss occurs. A number of other factors also contribute to losses. Each drive system and engine combination is a little different. A basic drive efficiency curve is pictured in Fig. 4.

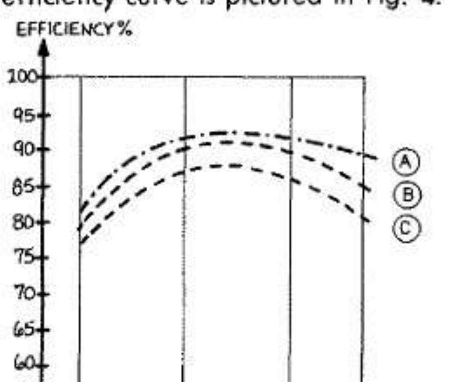


Fig. 4 - Efficiency varies with the shift ratio, the belt pressure and the clutch design. A basic efficiency curve starts out lowest in low ratio due to high belt loads. Between 2:1 and 1:1 the efficiency is best, and then it drops off due to high belt speeds in overdrive. Line B represents a transmission with correct side pressure on the belt. Line C shows the comparable lower efficiency of a transmission with too high of a belt pressure. B and C both have 13 degree angle sheaves. Line A shows the increase in efficiency with the 15 degree angle sheaves. Note the larger efficiency in overdrive because of the increased support of the belt.

In a low ratio the belt speed is low but the turning radius around the driving clutch is tight and the belt tension is high, giving lower efficiency. As the transmission moves toward a 1:1 ratio the turning radiuses become larger on the driven and the belt tension is reduced from the torque feed back, giving higher efficiency. When the transmission goes into overdrive the belt tension is down, but the belt speed is high and the turning radius on the driven clutch is getting tighter and contributing to a lower efficiency.

Efficiency is not the same throughout the shift range. This has an effect on your choice of gearing depending on your application. High side pressure, high engine speed and overdrive, all contribute to low efficiency.

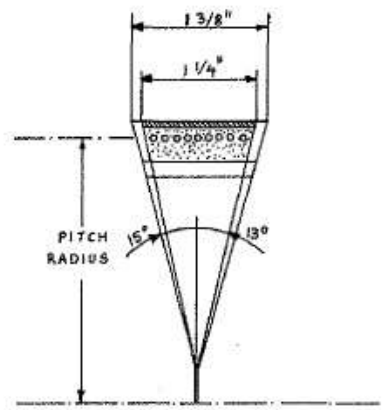


Fig. 5 - Changing sheave angles from 13° to 15° (26° total to 30° total) offers an improvement in overall efficiency as the belt is better supported, and the 30° total angle was used on OMC machines in the 1970's.

In the 70's OMC used a widebelt system with 15° sheaves (30° total) on its Johnson and Evinrude machines. This system offered improved efficiency over the 13° sheaves (26° total) because it offered the belt more support.

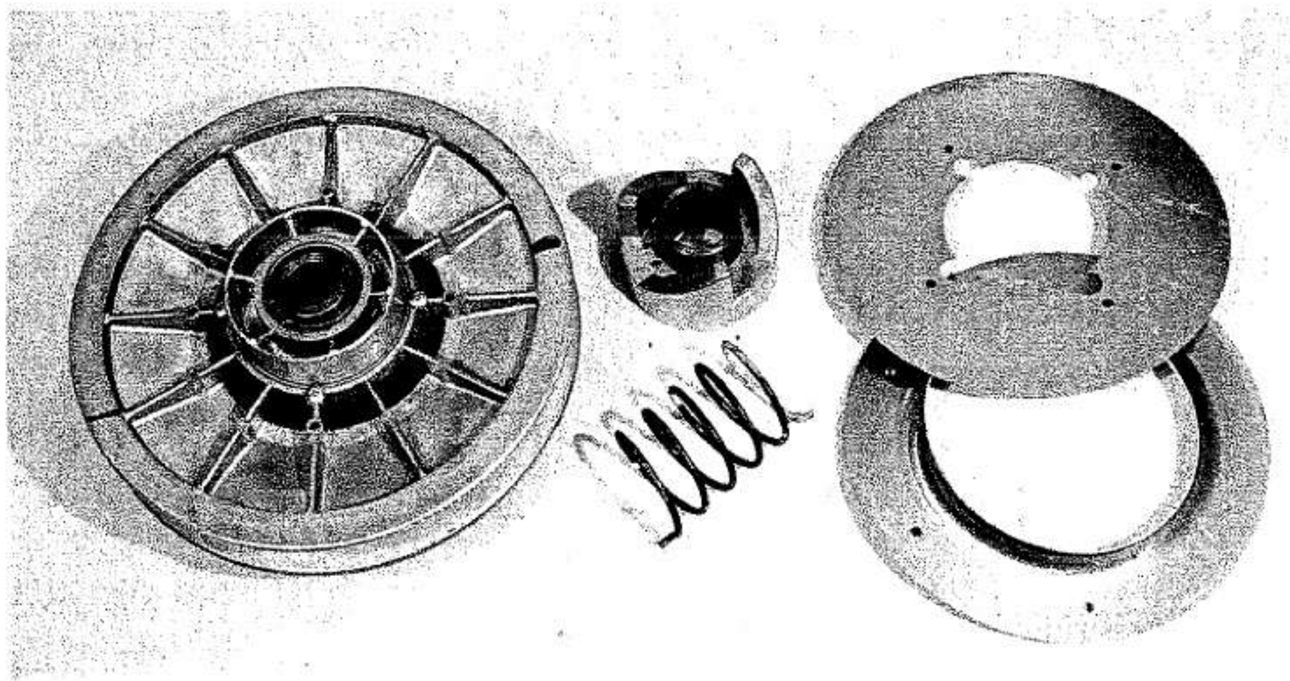
Other manufacturers at the time did not follow suit, and as horsepower output of larger engines increased, it was felt that the 26° systems were able to transmit more torque loads without belt slippage.

In modern systems primary angles are often a compromise with 27° on the primary and 29° on the secondary. This is because the belt has less "wrap" and operates at lower radiuses on the primary under hard take-offs. The belt is therefore compressed more on the primary than the secondary, and the smaller 27° angle better matches this compressed state of the belt.

This often also depends on belt design and compound, and some sheaves which are cut for use with a soft compound may experience belt slippage with harder compound belts.

CHAPTER 4

The Driven (Secondary) Clutch



The driven clutch is a simple design doing a complicated job. The spring and cam are the key to its performance. On the left is a complete assembly, then the cam and spring. Some models use windage plates mounted over the ribs to decrease "fan" losses.

Power is transferred from the driving primary clutch on the engine to the driven secondary clutch by the belt. In order to transfer the power, there has to be enough side pressure on the belt in any given shift ratio to prevent the belt from slipping. Horsepower is defined as torque times speed. When the clutch is in a 3:1 low ratio, the torque produced at the driven sheave is 3 times greater if efficiency losses are not considered. In a .75:1 overdrive position the torque at the driven sheave is only ¾ of the engine torque. Only in 1:1 should the torque output be the same.

This relationship is very important to understand for clutch tuning. To transfer 3 times the torque requires much more pressure on the belt than when the transmission is in overdrive.

What the driven unit is asked to deliver is this: high side force in low ratio which diminishes as the transmission shifts out into overdrive. Providing this characteristic is not easy. It cannot be done with a straight pressure spring as used in earlier transmissions. With a straight pressure spring you would need a high pretension to take care of the high torque in low ratio. As the transmission shifted out the spring would increase in tension and cause

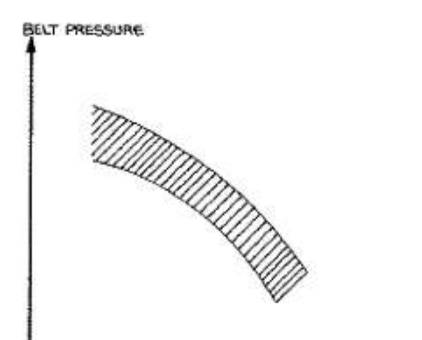


Fig. 1 - Side force is shown on the vertical axis, and shift ratio on the horizontal axis. The area between the two lines represents a design envelope with the side pressure necessary for good efficient power transfer. This relationship was first explored by a research test fixture that varied side pressure against power output in all ratios.

much higher belt loads than was necessary. This created low efficiency, high power losses, and very short belt life.

A transmission with an overall shift ratio of 4 will not require 4 times as much belt pressure in low ratio as in overdrive. In low ratio the belt travels at the outside diameter of the sheave and the torque has about twice the radius to work at compared to overdrive, when the belt is at the bottom of the sheave. This 2:1 radius ratio cuts the side pressure requirements down. Divide the overall shift ratio with the radius ratio and you come out with a side pressure ratio of 2:1.

This may be getting a little on the technical side, but the point is that approximately twice as much side pressure is needed in low ratio as in overdrive. If you stay outside this ratio you will either end up with belt slip or high belt wear and low efficiency. We are interested in getting as much power to the ground as possible, and we want to make sure we run the belt pressures that give us the highest efficiency.

Early transmissions with straight pressure springs were notorious for blowing belts on top end and having bad efficiency. They had more than twice the belt pressure needed when shifted out.

This put large limits on the amount of power that could be transferred. Belts blew regularly at 25 HP. If the situation could be remedied and the side pressure cut in half on top end, there should be no problem transferring 50 HP with the same belt.

The solution came with the torque sensitive ramp introduced on OMC's Evinrude and Johnson snowmobile models in 1964. With the torque sensitive ramp, torque was fed back through the angle of the ramp, and converted to side pressure against the belt. With high torque you got high side pressure and with less torque you got less side pressure; just what was needed.

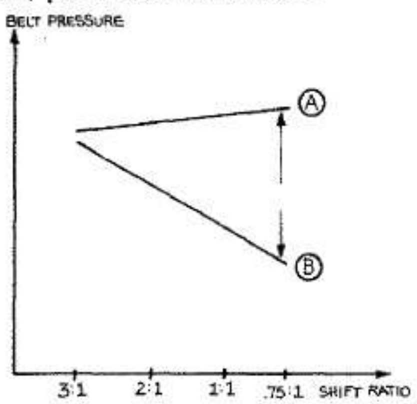
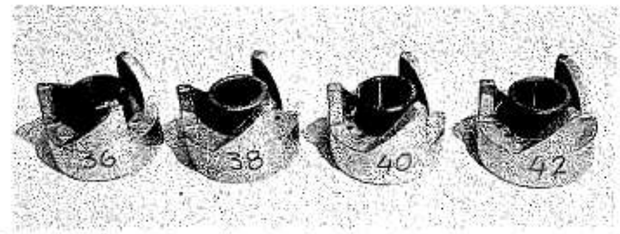


Fig. 2 - Side force vs. shift ratio. Line A represents the side force available with the old straight pressure spring design. When compared to the design envelope in Fig. 1, it becomes clear that too much side force is exerted in high ratio. This results in high power loss and excessive belt wear. Line B represents the improved ramp and torsion spring design which falls within the desirable design envelope in Fig. 1.

The new torsion spring and torque feed back ramp design soon became the accepted standard in variable speed transmissions. Great strides were also made in belt designs. Strengths more than doubled with use of new Kevlar fibers instead of the old glass fiber reinforcements. With these improvements in design, variable belt transmissions are now successfully transferring over 250 HP on some racing snowmobiles.

A driven clutch consists of a stationary sheave and a movable sheave working against the cam and spring. Both the cam and spring affect the tuning of the transmission so let's take a closer look at how they work.



Racers and experienced tuners keep a variety of ramps around to fine tune their machines. The smaller the angle, the more torque feed back and higher belt pressure you get.

THE TORQUE FEEDBACK RAMP

The torque ramp consists of three ramps spaced around a cylinder surface, and working against sliding buttons in the movable sheave. The angle of the ramps and the radius they work at both have an influence on how much side force is fed back into the sheaves from the torque. The smaller the ramp angle the more side force is produced. In Fig. 3, the relationship between the cam angle and the side force is shown. Side force is equal to the

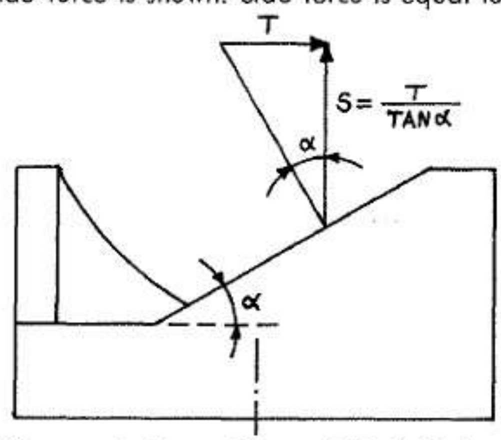


Fig. 3 - The amount of torque (T) converted into side force (S) on the belt is dependent on the ramp angle α. The smaller the angle the more side force is produced against the belt.

force produced at the ramp divided by the trigonometric function "Tangent of the ramp angle." Force produced at the ramp is dependent on the shift ratio and the belt radius compared to the ramp radius. The smaller the ramp radius the more side force is produced. Since the side force is defined by both the ramp angle and the ramp radius you can not directly compare ramp angles from one brand of transmission to another. For those of you who like to play with numbers, the theoretical side force produced by the cam has the following formula:

$$\text{Side force} = \frac{\text{Torque} \times \text{shift ratio} \times 12}{2 \times \text{Ramp radius} \times \tan\alpha} = \text{lbs.}$$

Torque is in ft. lbs.; ramp radius is in inches and tangent of the ramp angle.

The actual side force is also dependent on the side force from the spring, the pretension torque load of the spring, the efficiency, the sheave angle and any friction losses taking place.

There has to be enough side force on the belt to overcome the belt pull with the available friction between the belt and the sheave. Most drives are designed to have about 30% more side force than necessary to make up for variables in friction due to belt wear, sheave surface, belt speeds, temperature, etc. For the ultimate in low friction, check out the roller action secondaries on pages 19 - 20.

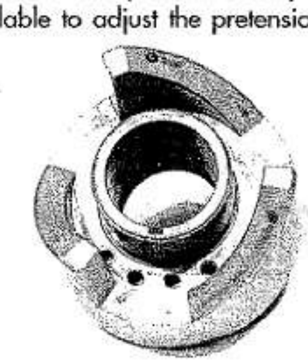
Combinations of ramp angles and pretension depends on the shape of the power curve and the intended use of the machine. Many cams have a combination of angles to give a harder shiftout without losing top end RPM. One such popular combination for the Polaris is the 42° - 38° - 34° cam. This cam starts in low with 42° for quick shiftout while the 34° maintains the correct RPM on top end. It has to be combined with a stiffer Blue Driven Spring with more pretension to work correctly.

Ramp rule: The smaller the ramp angle the higher the side force.

THE PRETENSION SPRING

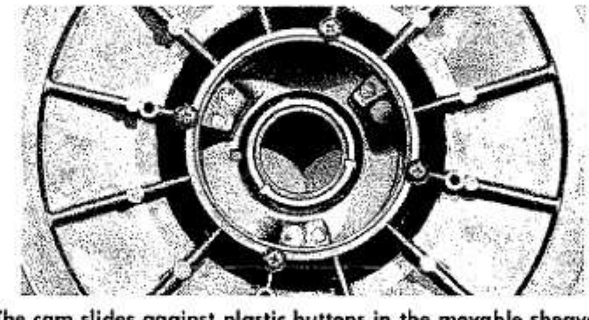
Some initial pressure is needed on the belt before the torque feed back can begin to increase the belt pressure. A driven clutch has a combination of ramp and spring. The spring works primarily in torsion although a small amount of side pressure is also present. Usually a number of holes are available to adjust the pretension

The spring and ramp assembly. Not only is the spring preloaded by twisting, but there is also side force from compressing in place.

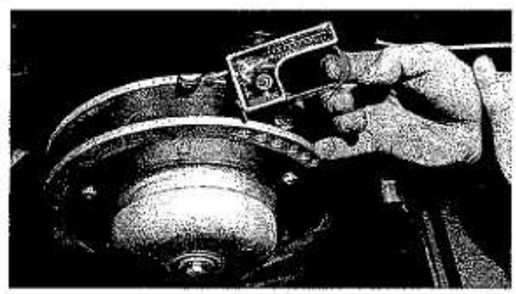


Spring preload can be changed by using different locating holes in the cam. Some cam designs have as many as six preload holes in the cam.

from 6 to 25 lbs. of pull on the outside diameter of the sheave. Pretension pull is easily measured by using a small fish scale and hooking the hook into one of the balancing holes. Spring tension is important for both initial belt pressure and back shifting. The higher the tension, the quicker the backshift. Springs with different rates are available, and springs with higher rates are used when more pretension is required for higher horsepower engines.



The cam slides against plastic buttons in the movable sheave. The other end of the spring is also mounted in the movable sheave. This clutch has one mounting hole, but some designs have 2 or 3 alternate locations in the sheave as well as the cam. On this Polaris design, the sheave slides on a small center bushing and a larger outside bushing that rides on the cam.

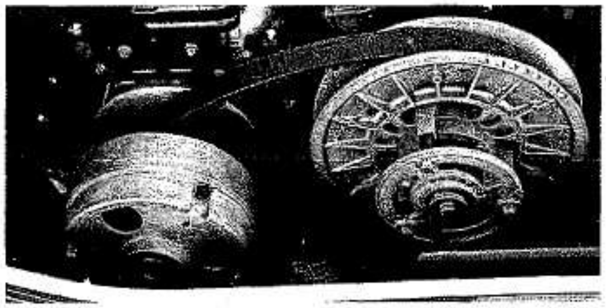


Keeping track of the pretension of your secondary driven clutch is important for clutch tuning. A simple fish scale can be used effectively if you make sure you measure the pull at right angles with the radius.

When the front clutch overcomes the side pressure of the driven clutch, the belt starts to shift out and change ratio. Tightening up the spring will require more force to open the sheaves, and more RPM is then required to overcome the extra tension. Many people use this effect to "tune" the RPM by changing the pretension. Although it works for a quick fix, it has some other effects such as changes in back shifting and loss of efficiency. The correct way to tune is to pick the setting on your driven clutch which gives you the best combination of back shifting and efficiency, and then obtain the correct engine speed by tuning on the front clutch.

When clutches get worn, friction loss increases and backshifting starts to suffer. Most riders try to fix the problem by increasing the spring tension. This may help for a short time, but it also speeds up the wear process and increases the friction loss. The correct procedure at that point is to go through the clutches and replace the worn parts.

Spring rule: The stiffer the spring and the higher the pretension, the higher the sideforce.

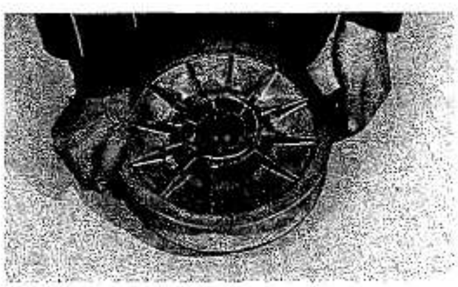


The Arctic secondary driven clutch is a "reverse cam" design. The cam and spring are mounted on the outside of the clutch for easier access and a closer mount to the chassis.

REVERSE CAM SECONDARY

Reverse cam designs have sometimes been advertised as a new and revolutionary breakthrough. This may baffle some people, but there is actually no great operational advantage in a reverse cam driven clutch. On a reverse cam clutch such as the Arctic, John Deere, and Kawasaki designs, the cam is simply placed on the outside of the clutch away from the chassis. The advantage is that the clutch can be placed close to the chassis, and this allows the engine to be placed closer to the center of the machine for better handling.

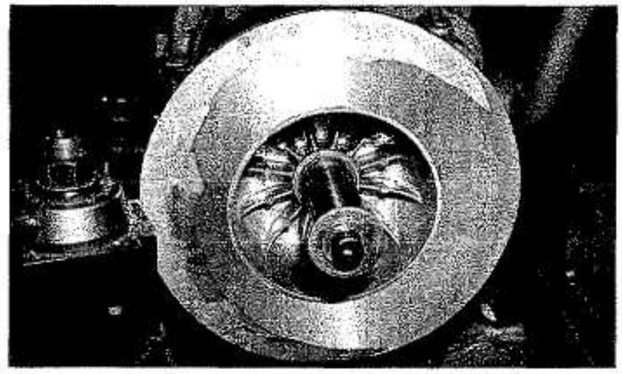
There are some extra complications with the design, and sometimes you are limited in what cams you can use. The Arctic reverse clutch used stiff springs and large ramp angles. This made the clutch less torque sensitive. Less torque sensing and more spring pressure can have advantages in the mountains where high altitude reduced torque. This concept has also been used on twin track Sno-Pro machines where the power curve is peaky and you don't want to lose RPM in the turns. Some efficiency is lost with this system, but sometimes this is of little consequence compared to falling totally off the power curve. You make your compromises based on the application.



As the sliding bushings wear, the movable sheave may start to jam in the movement. To check for this condition remove the spring and push down on one edge while holding light upward pressure on the other. If the sheave jams and refuses to slide down, it's time for new bushings.

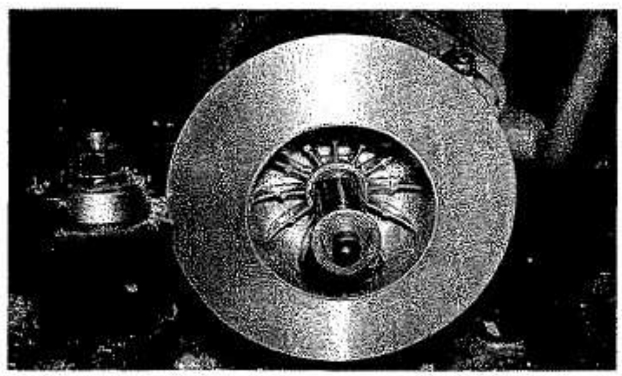
MAINTENANCE

Friction is the enemy of performance, and maintaining the driven clutch in good shape has great influence on power delivery and belt wear. The critical areas are the sliding bushings and the



Diecast sheave surfaces are not true and consistent. The part cools in the die at different rates making the surfaces uneven. A light cut has been taken on this sheave to show the uneveness.

cam buttons. All torque is transferred through these buttons, and they can get quickly worn down. When they are worn down they actually reduce the pretension. The increased friction deteriorates back shifting. Buttons should be inspected often for wear.

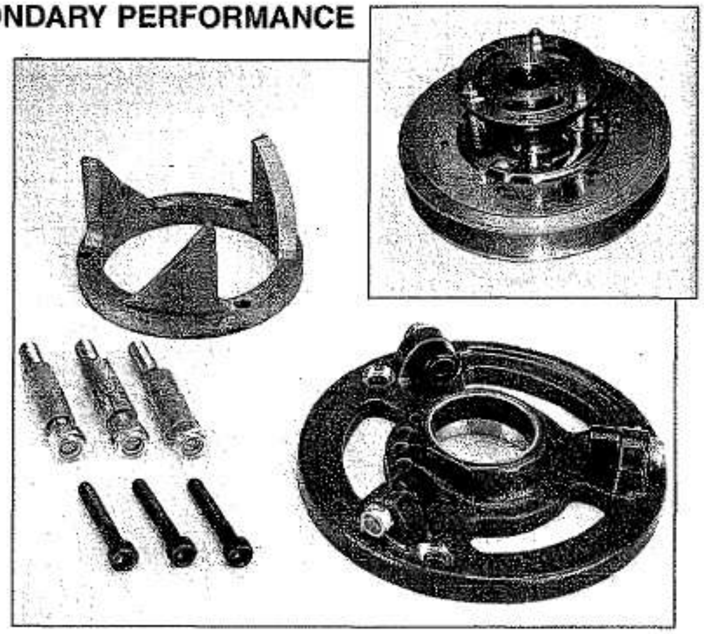


A fully machined sheave has true and consistent surfaces for better efficiency and belt wear.

The second critical part is the large bushing sliding on the outside cam surface. Because of the large diameter it is relatively narrow compared to its size, and this makes it easier for it to twist and jam against the cam when it gets worn. If the small sliding bushing is also worn the whole movable sheave may get stuck on the cam. A quick way to check for this is to remove the spring and then see if you can move the sheave up and down by pushing on the outside edge. If the sheave gets stuck on the cam, it's time for new bushings. Older clutches do not have machined belt surfaces. As the sheaves cool in the die during casting, they cool at different rates and do not come out perfectly straight. We prefer to machine the sheave surfaces so they are perfectly true for better performance. As little as a .020" cut is often enough to remove most of the uneven surface. All newer clutches now have machined sheave surfaces.

IMPROVING SECONDARY PERFORMANCE

This Arctic cam has received a surface hard coating and an outer layer of teflon.



This Roller Action kit converts the Yamaha YVX clutch to a full roller action setup to eliminate sliding friction. The steel needle bearing cam follower rides against a hardened steel cam. This steel against steel combination gives the quickest backshift reaction and the best wear characteristics.

Reducing Friction

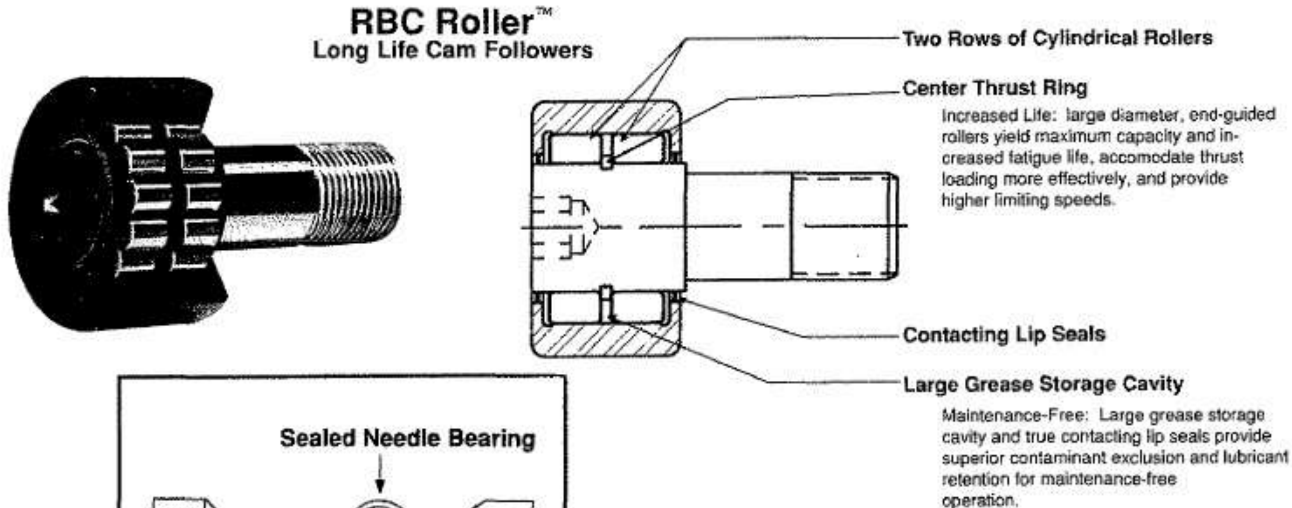
Friction is the enemy of quick response. This will show up in slow reaction to speed changes, such as slowing down for a corner or accelerating out of a corner. We often refer to this as the transmissions ability to "backshift" (go to a lower ratio) or to "upshift" during acceleration (go to a higher ratio). In a well maintained clutch system the largest source of friction is between the secondary cam surface and the plastic buttons they slide against. As the cam surfaces get smeared with belt dust, this friction will increase, and we may some times see a friction coefficient of over 10%. When the sliding motion changes direction, as it does when it goes from an upshift to a backshift, the friction force has to change from 10% in one direction to a 10% force in the opposite direction with a stop in between. This results in a lag in the response, and the engine speed will be pulled down until the drive system reaches its new ratiio.

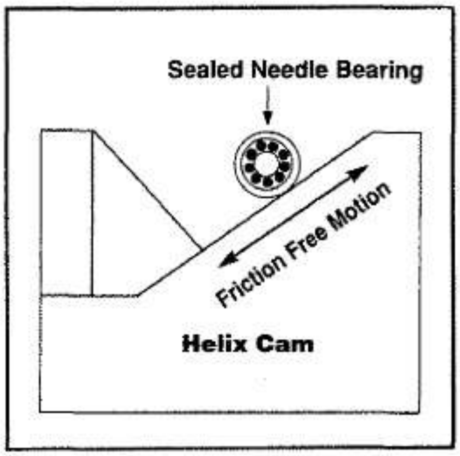
The solution to this problem is to reduce or eliminate the friction between the cam and its sliding button. As we shall see, it is possible to both reduce and eliminate friction, depending on the amount of money you want to invest. Reducing friction can be done by coating the cam surface with a teflon coating. This will reduce the friction by up to 50%. This means changing the friction from a possible 10% to a much lower 5%. This will improve response and enhance the performance of your transmission. These cams are usually aluminum cams that have first received surface hard coating, and then an outer layer of teflon coating.

This hard coating enhances the durability of the cams by preventing scratches in the surface from belt dust. The teflon coating not only reduces the friction, but it also prevents belt dust from sticking to the surface. As a result the cams run longer without requiring cleaning compared to stock uncoated aluminum cams.

Eliminating Friction: The Roller Action Clutch

While we worked on our CVT transmission in our race car, it became obvious that the teflon coated cams represented an improvement in performance. The next step was to try to eliminate the friction completely, and this led to the design and development of the first Roller Action secondary clutch used in a snowmobile drive system. We accomplished this by machining away the button towers in a Polaris clutch, and installing a 5/8 needle bearing cam follower to run against the cam. It was quickly obvious that the aluminum cams could not take the concentrated line contact of the roller without damaging the aluminum cam. We therefore switched to a steel cam surface, and also had to case harden the steel surface to get a satisfactory wear. The result has been very gratifying.





The heart of the roller action clutch is the steel needle bearing cam follower working against a hardened steel cam. The needle bearing design has a double row of rollers with a built in grease storage and lip seals on each end. This results in long life and great protection from outside contaminants like belt dust.

The concept not only proved to greatly improve both acceleration and backshifting, it also improved efficiency and reduced belt wear. The key to the enhanced performance is the elimination of sliding friction by using needle bearing rollers against the hardened steel cam. The hard steel roller against the hardened steel cam also eliminates any rolling resistance that will occur if a more flexible material is used in the roller. Hardened steel rollers against a hardened steel cam is the best for elimination of friction and rolling resistance, and gives the best wear resistance. The roller action secondary clutches were tested thoroughly in both ~~our race car and in snowmobiles~~ in oval and snow cross racing and in trail riding before they were released in the Polaris version. The Polaris roller action clutch was well received and the following year we released an Arctic version, and now also offer one for the Yamaha YVX clutch. Roller action secondary clutches are now available from both Arctic Cat and Polaris.

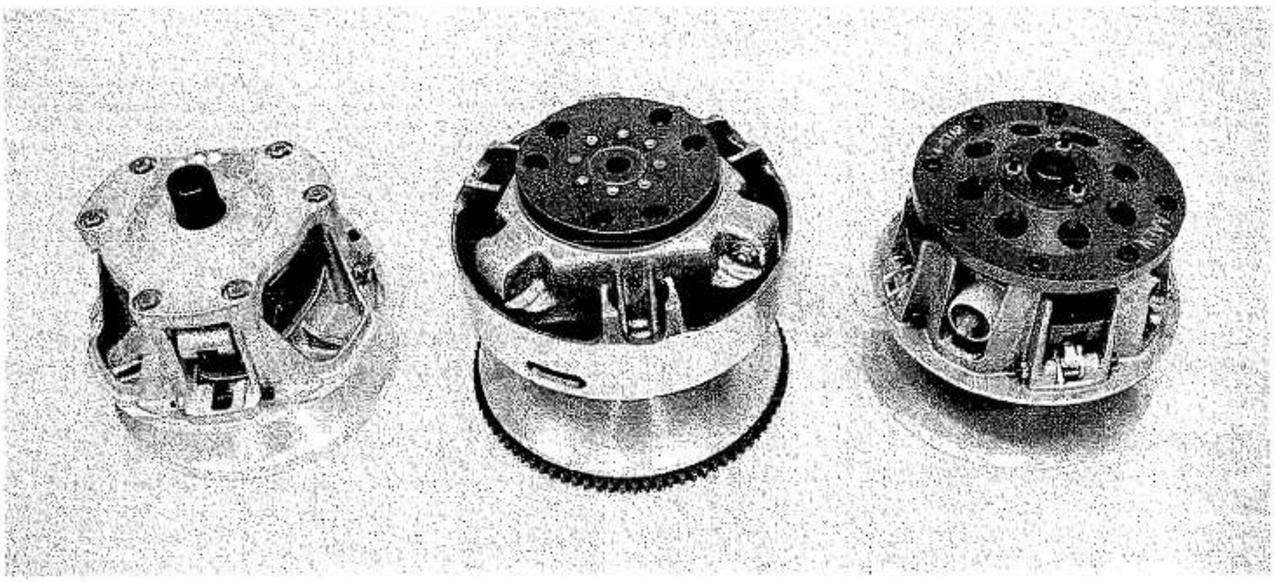
For best performance, you need the needle bearing rollers working against a steel cam, a costly setup. The factories decided to cheapen up the design with fiber rollers against aluminum cams. This is an improvement over the plastic button, but will not perform as consistently as a steel against steel roller design on higher horsepower engines.

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.

CHAPTER 5

The Driving (Primary) Clutch



From left to right: Polaris 3 flyweight, Ski-Doo TRA and Aaen 4-Star 4 flyweight roller clutch.

While the correct design and adjustments on the driven secondary sheaves determines the efficiency of the transmission system, the driving primary clutch must control the engine speed and keep it running on the power curve through the entire shift range. When both systems function correctly and give maximum horsepower coupled with best efficiency, you have a correctly tuned clutch.

The movement of the sheaves and the belt is controlled by flyweight and cam-mechanisms in different arrangements from one design to another. Much of the "tunability" of the clutch system depends on the design of this mechanism and there are advantages and drawbacks to all of them. Basically the systems have to overcome the forces of the pressure spring, and then match the side pressure requirements of the driven.

The driving clutch is required to take care of all the speed variations from stand still to top end. These distinct phases include free-running, engagement, clutching, low ratio acceleration, over rev shiftout, straight shift out and high ratio overun. Fig. 1 shows the different phases in relation to vehicle speed on the speed-diagram.

Centrifugal force is created by the flyweights and transfered through the roller and flyweight cam surface into the movable sheave. This flyweight force first works against the pretension of the pressure spring. As the driver shifts out, the flyweight force works against both the pressure spring and the load of the driven sheave. To accomplish these speed changes, the centrifugal force and the cam profile have to be matched to overcome the pressure spring and driven clutch forces at the right point on the shift curve.

Free running (disengaged). The driving clutch is disengaged below the engagement speed, to permit the engine to be started and to idle when the vehicle is at a stand still. **At engine speeds below engagement, the pretension load of the pressure spring is larger than the flyweight force.**

Engagement. Engine speed has to be high enough to produce useful torque that can accelerate the machine and driver from a standstill. Trail machines usually are tuned to have good low end torque for engagement in the 3000-4000 RPM area. Low engagement gives smooth take off and good flexibility on tight trails if coupled with good low end torque. Racing machines are tuned for maximum top end performance with sacrifices in low end torque. To come off the line hard with a racing engine, it is often necessary to engage in the 5500-7000 RPM range. **When the sheaves engage the**

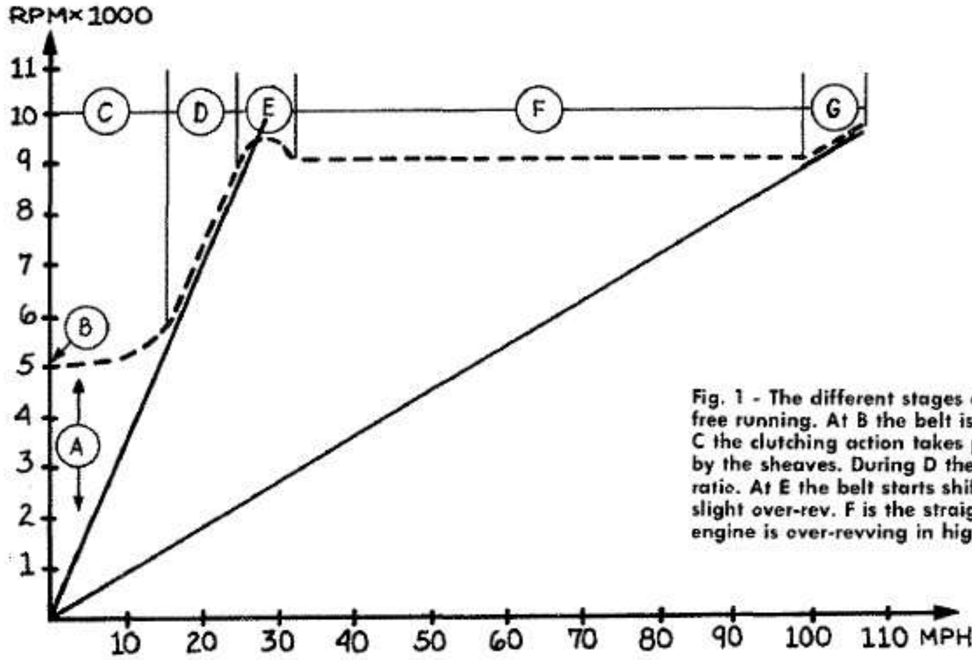


Fig. 1 - The different stages of transmission operation. A is free running. At B the belt is engaged. In the area marked by C the clutching action takes place where the belt is gripped by the sheaves. During D the machine accelerates in low ratio. At E the belt starts shifting out into higher ratio with a slight over-rev. F is the straight shift portion and at G the engine is over-revving in high ratio.

belt, the flyweight force is just overcoming the pretension of the pressure spring.

CLUTCHING

Belt engagement starts with a lot of slip when the vehicle is stationary, increasing to 100% engagement when the vehicle is accelerating along the low ratio line. In between these two points there is slippage between the belt and the sheave. After the flyweight force overcomes the spring tension, the RPM has to increase and give more sheave pressure on the belt until there is no slippage and full engagement. This takes place as the vehicle is starting to move and as the engine speed and vehicle speed intersect at the low ratio line. This area is often critical for good and smooth take-off. Incorrect matching of springs and weights may lead to excessive slippage and belt wear.

The clutching phase begins when the flyweight force overcomes the pretension, and lasts until the flyweights have generated enough side force to transfer the engine torque without slipping the belt.

LOW RATIO ACCELERATION

After the sheaves have gripped the belt and they are transferring the torque without slipping, the machine will accelerate along the low ratio line. The engine speed will continue to increase as if you were in low gear in your car. While the vehicle speed is increasing the belt remains in low ratio at the bottom of the driving sheaves. The belt will not start moving out on the sheave until the flyweight force has become large enough to overcome both the pressure spring and the side pressure of the belt on the driven sheave. When the engine speed has built up to create enough centrifugal force to overcome the pressure spring and the driven sheave pressures, the belt will start shifting out and the ratio will change. This is called the "shift point" and should coincide with the power peak of the engine.

Between full engagement and the shift point, centrifugal forces are larger than the pressure spring but less than the belt pressure from the driven clutch.

SHIFT-OUT-POINT

As the RPM of the driving clutch increases in low ratio, the centrifugal force from the flyweights will eventually be large enough to overcome the tension on the belt from the driven clutch. When this occurs, the transmission has reached the shift out point and the belt will start to move outward on the driving clutch and inward on the driven clutch. As the belt starts to move, the drive ratio is changed.

Ideally, the shift out point should be at the power peak of the engine and the shift curve should be straight from there through high ratio. In practice it may sometimes end up a little different. If you have a heavy machine and a small high revving engine, you may want the shift out point to be several hundred RPMs higher than the power peak. As the machine increases acceleration the engine may be pulled down in RPM by the increasing load as the transmission shifts out. By over-revving slightly in the beginning the engine will be pulled down on to the power peak, and smooth acceleration continues. If the shift out point was right on the power peak, the reduction in RPM would have pulled the transmission down off the peak and a more sluggish acceleration would have occured as the transmission worked its RPM up again.

This shift out over-rev is of short duration, and highly individual depending on the combination of machine weight and engine power. On a super light drag machine with a big engine, you may get away with starting the shift-out slightly before the power peak as the power will pull you right through without an RPM drop. The best way to achieve the initial over-rev is to modify the shift curve of the flyweight right off the engagement point. This is usually done by extending the engagement flat into the shift curvature. The length and shape of the transition from engagement to shift curve will determine the amount and duration of the shift out over-rev.

The shift out point occurs when the flyweight force overcomes the belt pressure from the driven clutch.

STRAIGHT SHIFT

The ideal shift-curve is "straight" between the low and high ratio. This means that the engine speed is held constant at the power peak while the transmission is shifting out and the vehicle speed is increasing. With the exception of a slight shift-out over-rev as discussed in the previous section, a straight shift at the power peak will give the maximum performance. To maintain a straight shift is not an easy task. As we saw under the discussion of the belt pressure requirements, the pressure on the belt is actually decreasing as the transmission shifts out. This decrease in belt tension may be as much as 50% between low and high ratio. If the decrease is not matched by the flyweight force, the driving clutch will compensate by lowering the engine speed. The driving clutch will always match the required belt tension by changing the engine speed if the flyweight system is not correctly calibrated. Our task is to make sure the flyweight system matches the belt forces at the peak power engine speed. Straight shifting is accomplished by calibrating the relationship of the flyweight curvature and roller location as well as the arc traveled by the flyweight. Engine speed is determined by the weight of the flyweight system.

Since the flyweight system works against the pressure spring in the driving clutch, the loads and rates of this spring also influence straight shift and engine RPM. Flyweight curvatures have been experimented with and fine tuned by the manufacturers over the last 15 years and they provide a good selection of tuning components. Polaris is a good example; they provide two basic flyweight curvatures, trail and racing. The trail curvature has lower engagement and softer shift-out than the racing curvature. Each flyweight group is available with weights in 2-4 gram increments for each curvature. Each manufacturer also has a large selection of springs with different pretensions and rates for calibration purposes. Achieving the correct combination of flyweight and spring to obtain a straight shift at the correct engine speed is the task of the tuner. This will be covered in greater detail under the Selection of Components and Testing chapters.

Straight shift is obtained by matching curvature and spring rates; correct engine speed is dependent on the weight of the flyweight.

OVER-RUN

When the transmission is shifted all the way out into high ratio, engine speed must increase along the high ratio line as vehicle speed increases.

As the engine speed increases, the power curve goes off the peak on the back side and less power is available to propel the machine. Maximum speed cannot be obtained if you are off the peak; this means the machine is geared too low. There may be conditions where this is done on purpose because acceleration is more important than top speed. In most cases the machines are geared to give maximum top speed, which means they will seldom reach high ratio and over-run is not usually experienced.

In over-run, the clutches are shifted all the way out and the flyweight forces have no more influence on the engine speed. Engine speed will increase along the fixed high ratio line.

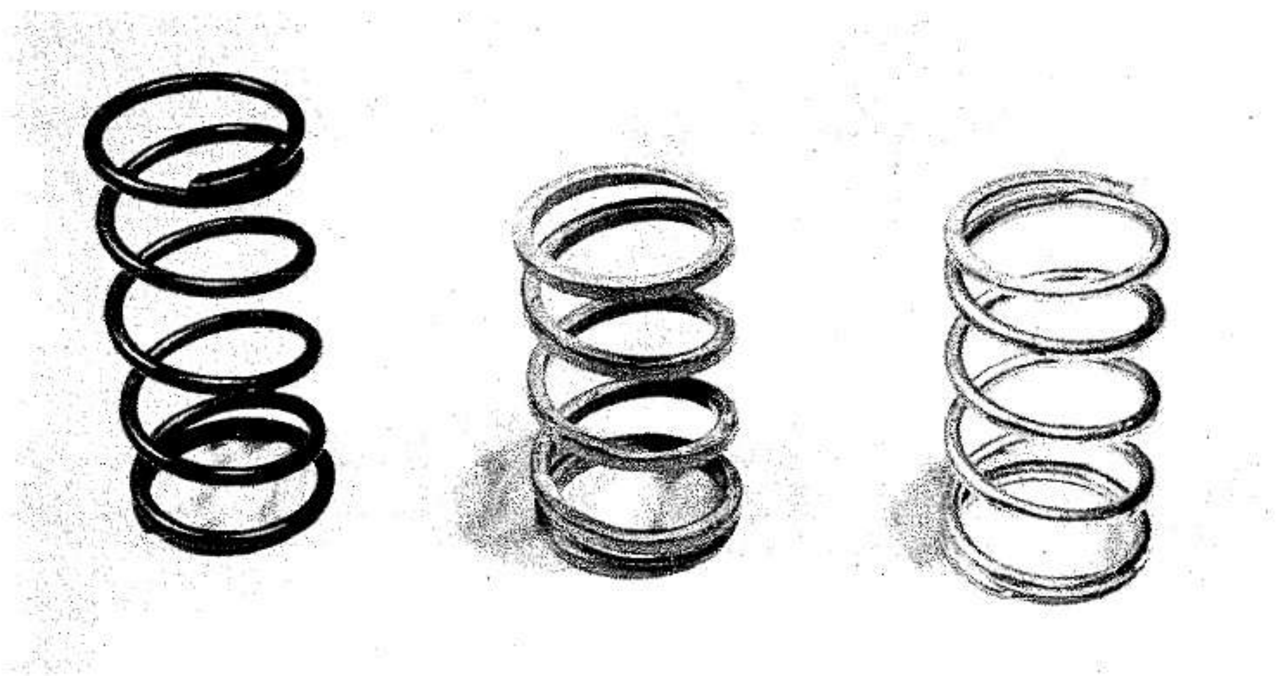
BALANCE

The driving clutch may reach speeds of 12,000 RPM, and your driven clutch even higher, and small amounts of unbalance may affect the performance and cause severe loads and loss of efficiency. It is therefore important that all parts are balanced as closely as possible.

Most driving assemblies have marks on their components so that they can be assembled in the same relationship each time to keep the balance correct. Sometimes the balance may be changed when the sheaves are machined true. This will change the original balance, and the parts should be rebalanced. This can if needed be done statically with a balance stand such as those used for crankshafts because the assembly is relatively narrow. Unbalanced clutches often result in poor shifting, premature wear, and sometimes broken or bent crankshafts.

CORRECT INERTIA

This is a phenomena many tuners are not aware of. Each engine has a natural torsional frequency in the crankshaft assembly, depending upon the flywheel effect (inertia) of the shaft itself and all the parts mounted on it. A natural frequency is what a tuning fork vibrates at when struck against an object. If the torsional frequency of the crankshaft happens to occur close to the power peak of the engine where the clutch has to operate, the result can often be disastrous. Clutch parts wear at astronomical rates and even crankshafts break. This often occurs when other than an original clutch is used. The cure is to add or reduce the inertia of other components, usually the flywheel. Yamaha did, for several years, have a wear problem with their SRX clutches until it was traced to be too little total inertia. The problem was cured by adding an inertia wheel to the flywheel. If you feel you have premature clutch wear, wrong inertia may well be your problem.



Three pressure springs with different characteristics. The spring on the left has thin wire and many coils, giving it a soft rate. To get good pretension it needs a tall free length. The center spring has a thick square wire and few turns, giving it a stiff rate. The spring on the right has a thicker wire than the one on the left but less rate than the center spring.

CHAPTER 6

The Pressure Spring

The pressure spring and flyweight mechanism work against each other in opposite directions. To achieve a free running condition and an engagement speed, the flyweight force must be opposed by the pressure spring. Engagement speed is determined by the amount of pretension the spring has been compressed to when installed in the clutch. Engine speed has to increase until the flyweight forces can overcome the pretension pressure and the sheave will then start to move and engage the belt.

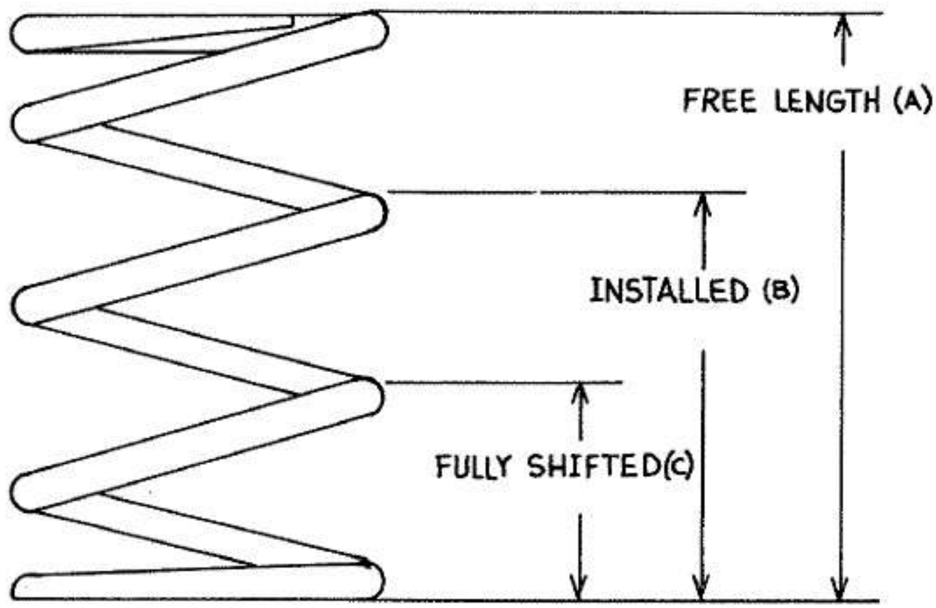
While the pretension determines the engagement speed, the spring rate has an influence on the shift characteristics. The side force on the belt is the result of subtracting the spring force from the flyweight force as they work against each other.

BELT PRESSURE=FLYWEIGHT FORCE– SPRING FORCE

RATE

A spring with 100 lbs. of pretension and a 50 lbs./in. rate will have a total load of 150 lbs. after 1" of sheave travel. Total sheave travel depends on the design. On a Polaris it is usually 1.340" with standard belts. The higher the load, the more RPM or heavier weights are needed to overcome it. Spring rate can be used to influence the shift curve and obtain a desirable "straight shift".

If you have an engine with a power peak at 9000 RPM, you would like the clutch to hold this RPM from low ratio to over-drive. If the clutch starts out at 9000 RPM but the engine speed drops slowly to 8500 in over-drive, you can compensate with a spring that has the same pretension but a higher rate. The higher rate means more spring load in over-drive since both springs start out with the same pretension. It will require more RPM from the flyweights to overcome the higher spring load in over-drive. The shift curve will straighten out and



Three data points are of interest to the tuner; the free length A, the installed length B, and the fully shifted out length C. Keep track of the free length to check if the spring took a set.

B and C are used to find the pretension and fully shifted out load.

stay at 9000 from low ratio to over-drive.If you had chosen a lighter flyweight with the same curvature instead of changing the spring, the problem would only have shifted to the low ratio. With the first spring and lighter weights you would start out revving at 9500 and fall to 9000 in over-drive, and your engagement speed would also have been higher.

The opposite would be the case if the clutch shifts up from 9000 in low ratio to 9500 in over-drive. In this case you want to find a spring with the same pretension but less rate to bring the RPMs down in over-drive.

Obtaining a straight shift at the power peak is your first objective when tuning your clutch. Correct engagement speed is a secondary objective. I say that because there are a number of ways to change engagement speed other than using a spring with high pretensions. Spring selection should therefore be focused primarily on obtaining a straight shift. You only need engagement speed when you start out. The rest of the time the transmission is shifting somewhere between low ratio and over-drive, and you want to make sure you are at the power peak. The correct procedure is to pick a flyweight and spring combination that gives you a straight shift. To fine tune the RPM, change the weight of the flyweight. To get correct engagement speed, grind flats or notches on the flyweight or tuck it under more. All of these methods will be discussed later.

To know what you are doing, you need to know the exact pre-loads and rates of your springs. This information is given in the appendix for each brand of clutch. Springs are usually color coded but sometimes the manufacturers will supercede the part number and use the color code for a new spring.

It's a good idea to understand how springs are designed so that you can make your own judgement in case there is some doubt and you are unable to get the correct information. A spring is like a long beam or rod except it's coiled up in a cylindrical shape. A long thin rod is easier to bend than a short thick one. The length of the coil wire is determined by the number of coils in the spring. A spring with five coils is softer than a spring with four coils of the same wire diameter. A spring with smaller diameter spring wire is softer than a spring with thicker diameter wire if they have the same amount of coils. High rate springs have fewer coils and thicker wire than soft rate springs.

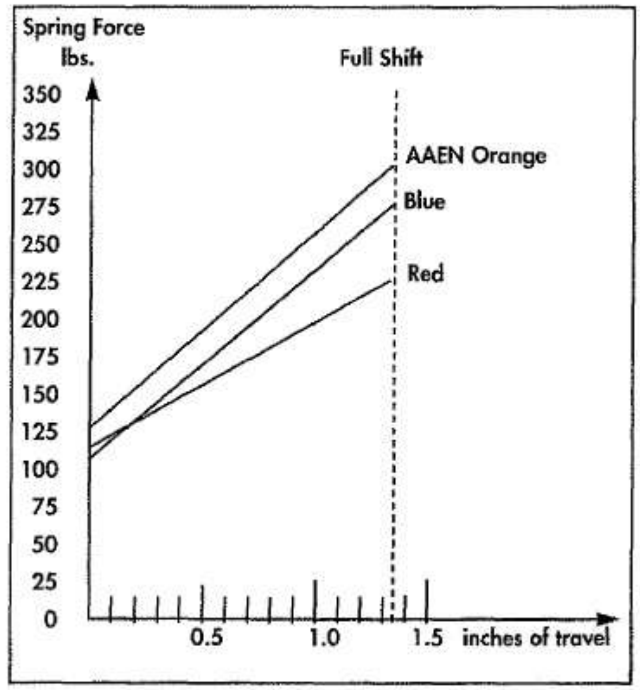
PRETENSION

Pretension is controlled by the amount the spring is compressed when installed in the clutch. A spring with a 50 lbs./in. rate needs to be compressed 2 inches to give 100 lbs. of pretension. If two springs have the same number of coils and wire diameter, the one with the larger free length will have a higher pretension.

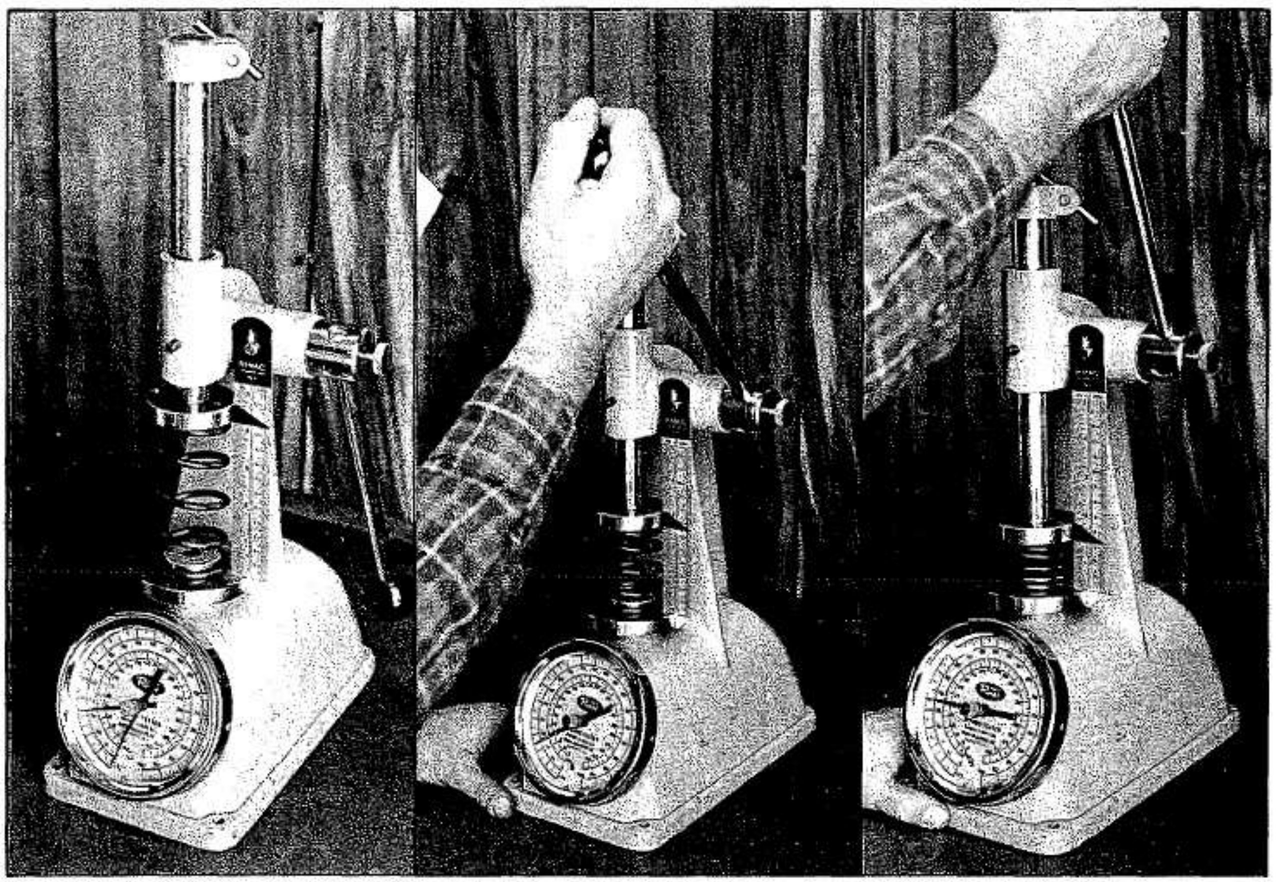
Some brands like Comet offer 1/16 spring shims to place between the spring and cover to increase tension. Caution should be used when stacking shims, more than 3 usually means that the spring will coil-bind before it reaches full travel. Some springs cannot be used with spring shims at all, because they are already close to coil binding.

Springs are not always the same. The manufacturer may change suppliers, the material quality may be different from one year to another, and the free length may be off 1/8" from specifications. Springs that are close to the stress limit may take a set after some use. This means that the free length of the spring will get shorter by sometimes as much as half an inch. On a spring with a 50 lb. rate, you would lose 25 lbs. of tension throughout the range. The effect will be a clutch that loses engagement speed and shift speed after some use. For a race driver, this could be a disaster. To make sure that the clutch doesn't change they start out calibrating with springs that have already seen some use and have taken a set. This only applies to some of the highly stressed springs and is not an area of general concern, but it may be worth while to keep your eyes open.

To be completely accurate, you can use a spring tester to check out the actual part. We use a Rimac automotive spring tester designed to check valve springs (see photo). Our spring tester is marked for the installed length and the fully shifted out length. In case of a Polaris clutch this is 2.530" for installed length and 1.190" for full travel. Sheave travel is then 1.340". We measure the load at the installed length of 2.530" which gives you the pretension load for engagement. The fully shifted out load is then measured at 1.190" to get maximum load. By subtracting the pretension load from the maximum load and dividing by the 1.340" travel we obtain the rate in lbs. per inch.



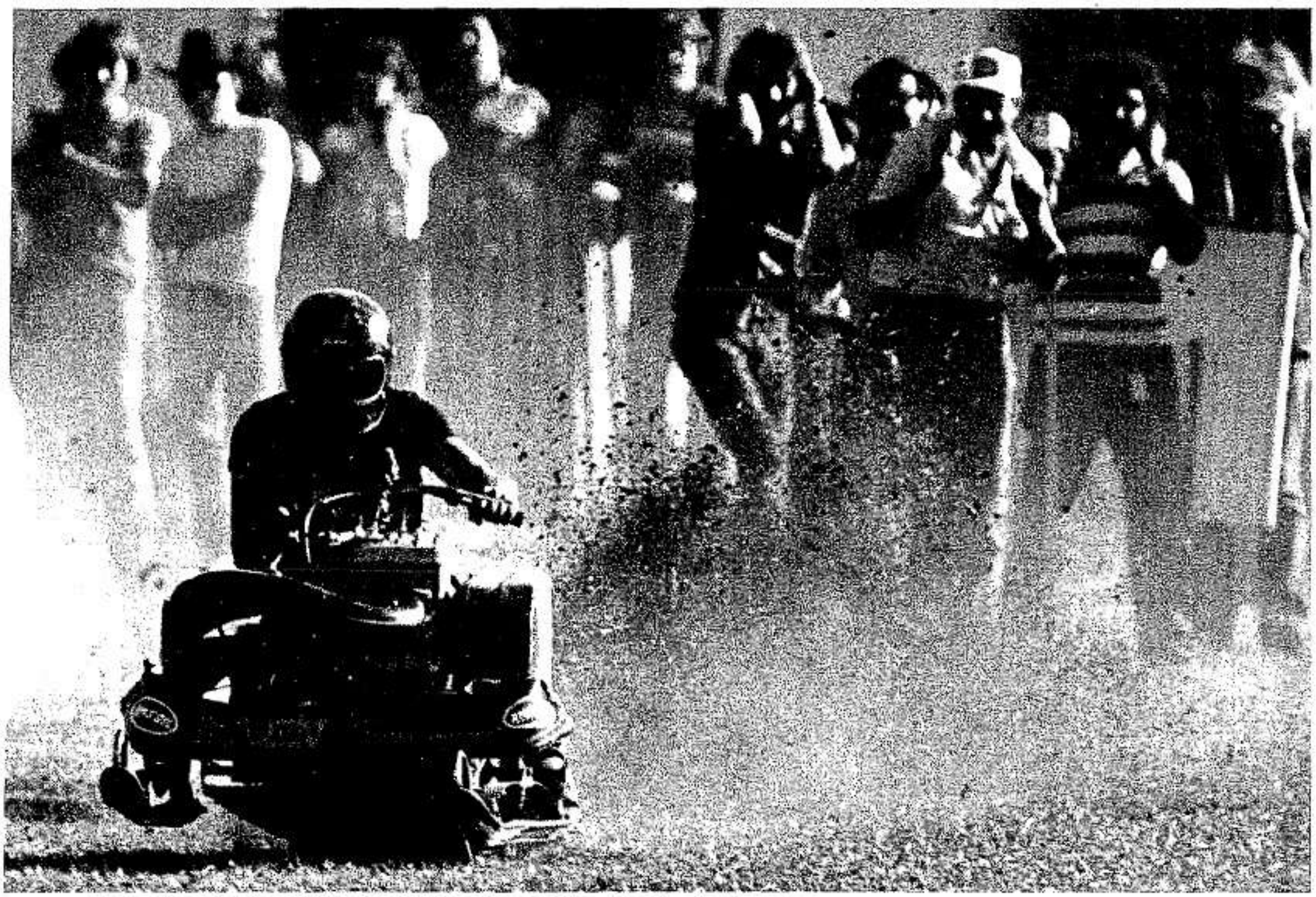
Three springs with different characteristics. The Blue and Red engage at the same speed, but the Blue has a higher rate than the Red, and will pull more RPM on top end. The Orange spring has almost the same rate as the Blue spring but engages higher and will also pull more RPM through the whole shift.



An automotive valve spring tester like this Rimac unit is used to test springs. The spring installed in this tester is in its unloaded free length position.

The spring has been compressed to its installed length, and the scale shows a pretension of 125 lbs.

The spring has been compressed to its fully shifted out length and the scale shows 230 lbs.


Drag racers grind flyweights with high engagement for a hard launch off the line.

CHAPTER 7

The Flyweight System

All shift force is generated by the flyweight system. The flyweight force first works against the pressure spring, and then works to overcome the belt forces from the driven clutch. To keep the engine on the power curve, the flyweight force has to match the driven belt pressure curve all the way through the shift range from low ratio to overdrive. If it does not match the driven belt pressure the engine speed will change until it does, but by then you may be off the power curve.

The tuner's job is first to match the shift force from the flyweights to get a straight shift, and then to get a good engagement speed. This may sound backwards, but we are talking about priorities. It is more important to get a straight shift first, because engagement speeds can be accomplished in several ways.

The early transmissions had only a straight spring for belt load in the driven unit, and a simple system in the front clutch was therefore sufficient. Many early clutches had straight weights, balls or kidney weights to overcome the pressure spring. None of these systems had a cam surface that could be used for tuning. You could only use the combination of weight working against the pressure spring.

With the introduction of the torque sensing system came the need for better tuning. The new flyweight systems involving weights, rollers and cam surfaces evolved.

There are a number of systems in use and they all have their advantages and drawbacks. Polaris, Comet and Yamaha use a flyweight with a cam surface working against a roller. This system starts with the weight in 6 o'clock position and it swings through to 9 o'clock for shift out. Kawasaki had a system where the flyweight starts out in 8 o'clock position and swings then to 11 o'clock. This system was unstable and hard to work with. Arctic's hex clutch, John Deere's TR-800 and Ski-Doo's new clutch have the cam stationary and the roller mounted on the arm. Because the arm was separate from the cam, shift speed could easily be tuned by changing weight, once you had the correct cam. This made it popular for certain racing applications, but bad back shifting was a draw back. This was later corrected in the John Deere TR-800. John Deere only made a limited quantity before they withdrew from the snowmobile market. Arctic long used a Comet primary clutch, but now has their own strengthened and improved version.

The largest share of the market is held by the flyweight design used by Polaris, Yamaha, Arctic, and Comet. Although this design is harder to tune than the hex Arctic, it has many advantages when correctly calibrated. Better transition from engagement to shift, dynamic stability in overdrive position and good back shifting are some of the features that have made it popular.

We are going to use this flyweight system as an example, and we will cover the other systems in the chapters on the particular clutches.

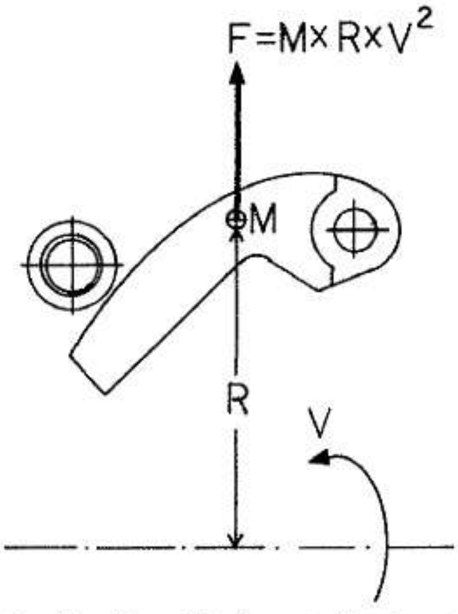


Fig. 1 - Centrifugal force (F) is determined by the weight of the flyweight (M), the radius of the center of gravity from the center line of rotation (R), and the square of the speed of rotation (V).

CENTRIFUGAL FORCE

To understand the flyweight system, it is important to know the fundamentals of centrifugal force. Knowing the basic physical rules governing centrifugal force makes it easier to make an intelligent choice when choosing flyweight components. To make it easier to understand, the basic mathematical formula for centrifugal force is:

$$F=MRV^2$$

Force = mass x radius x speed^2

We will not be going into the finer details. For those who aspire to make calculations with the correct terms, please refer to your physics book. One very good reason for this is that once you have figured out what the centrifugal force of your flyweight is, you will not know what to do with it. You need to know how to draw free body diagrams and have a sophisticated computer program to plug the results into. Even then, you will not be anywhere close to what you want to accomplish. You would have a hard time predicting all the variables in the computer program and it would be too inaccurate.

There is only one good way of calibrating flyweight systems and that is in the field. Knowing the relationship between the forces will help your testing when you have to make choices between flyweights. Decisions on where to grind flyweights, and how much to grind off, are also based on an understanding of the formula.

If you are of the real experimental type, there is a simple experiment you can do to gain a good feel for the importance of each variable. The author takes no responsibility for possible injuries due to inadequate safety measures when conducting this experiment. Most of you have already done this as part of growing up. Tie a rock to a rope, start swinging it in a circle, and observe the pull on the rope. To demonstrate the first part of the formula pay attention to the mass (or weight) of the rock. A heavier rock creates more force than a light one. The same is true with the flyweight system. The heavier the weight, the more force it produces.

Distance is the second variable. The longer the rope is, the harder it pulls as you swing it at the same speed. In a flyweight system the centrifugal force increases proportionally with the radius of the weight's center of gravity from the center of the shaft. When you grind weight off a flyweight you have to be careful where you remove it. By changing the flyweight you are moving the weight's center of gravity to a different radius, and this also has an effect on the shift force.

When the flyweight swings out, the center of gravity will move further away from the center of the shaft, and the force will increase. As we have seen from the analysis of driven clutch belt pressures, we want the force to decrease as the weight swings out, so the increased radius has to be compensated for by a cam shape which transfers less of the force.

Speed is the third variable. Force increases with the square function of the speed. This means that if you whirl your rock around twice as fast at the same radius as before, the force will be four times as great. In practical terms, this means that increasing the RPM of your clutch takes more than a straight relationship in weight reduction to increase the speed.

As an example, let's look at a clutch which runs at 8500 RPM with a 50 gram weight. What would be the necessary weight reduction to reach 9000 RPM if everything else remained the same? The difference in weight would be proportional to the difference in the square of the two speeds.

$$\textbf{Force difference} = \frac{9^2}{8.5^2} = \textbf{1.12 or 12\%}$$

$$\textbf{The new weight would be } \frac{50}{1.12} = \textbf{44.6 grams.}$$

Due to a number of variables affecting the tuning of the transmission, weight differences do not come out exactly as calculated. This confirms the rule that the best way to calibrate a clutch is under actual field conditions, but it helps to be aware of the relationships between weight, radius and speed. A quick repeat of the formula and how it affects centrifugal force:

$$F = MRV^2$$

Centrifugal force increases proportionally with the weight.

Centrifugal force increases proportionally with the radius.

Centrifugal force increases with the square of the speed.

CURVATURE

Introducing a flyweight system where a curved cam surface worked against a roller increased the tunability of the driving clutch. The force from a flyweight system cannot be made to match all the requirements from engagement to overdrive without compromise in some areas, unless there is a cam curvature to calibrate with.

Cam curvatures can be modified to give different engagement speeds, more aggressive shift-out, shift-out over revving, and smooth or aggressive shift-out patterns.

Before you go wild grinding on your cam surface, a little warning is in order. The manufacturers of the clutch systems have spent thousands of hours coming up with curvatures that work. Grinding as little as twenty five thousandths off a cam surface may change the engine speed by as much as 500 RPM. Grinding on the cam surface should be limited to the engagement and shift point area until the very last resort. You have to be very good, very accurate and have a good understanding of your purpose before you undertake modifications to the shift surface. The good news is that the manufacturers have a good selection of cams available to take care of 95% of all situations that occur. You just have to have the basic knowledge of how to mix flyweights and springs to obtain your desired results.

Roller position in relation to the flyweight pivot is also important and we shall take a closer look at this a little later on.

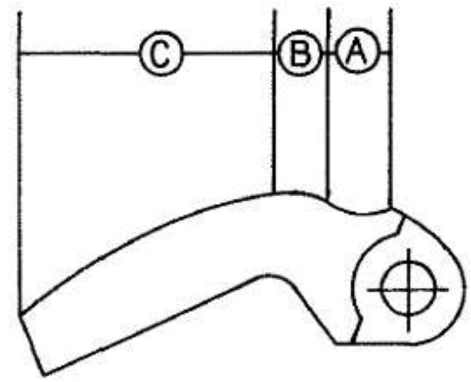


Fig. 2 - A flyweight has three distinctive phases: the engagement area A, the transition area B, and the shift curve C.

A typical flyweight has three areas we are concerned with: the engagement, the shift-out transition, and the shift curve.

The engagement position may be just an extension of the shift curve for low engagement from 3000 to 4000 RPM. For engagements between 4000 and 5500 RPM a flat is usually ground. For engagements over 5500 you will need a distinctive notch ground into the weight.

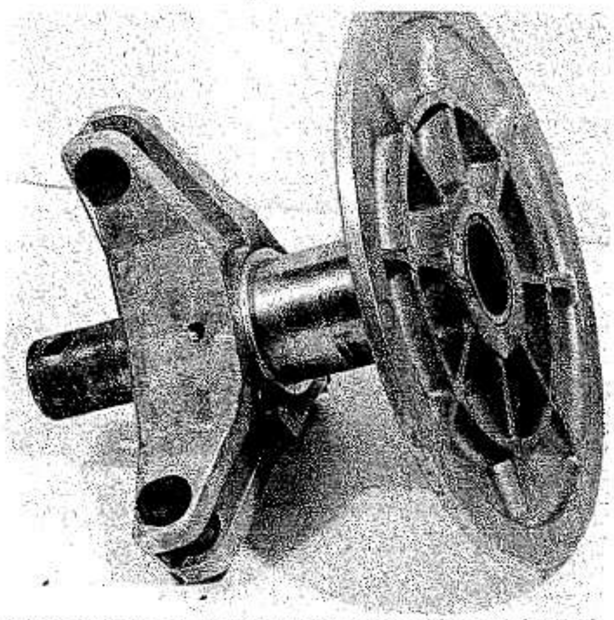
The transition portion is represented by approximately the first quarter of an inch of the shift curve beyond the engagement notch. Modifications in this area include extending flats for over rev or different curvatures to blend between the shift curve and the engagement notch.

The shift curve itself has a basic radiused shape carefully developed to match the side force requirements of the driven clutch belt pressure. Both Polaris and Comet have different sets of flyweight curvatures, from trail to race. Polaris's aggressive weights were developed for racing with higher engagement and harder acceleration in the early shift-out phase. Intended for light racers, they are now also used in larger high performance models. Comet's A-1, A-2, and A-3 weights are very close to the Polaris curvature, with slight modifications in engagement and transition. These weights are used on larger Arctic and Yamaha models. Polaris's and Comet's mild curvatures are fairly close, and both flyweights can be used in each clutch. Comet also has a set of aggressive "mild" weights intended for low RPM high torque motors such as big fan cooled engines.

ROLLER AND SPIDER POSITION

Polaris, Comet and Yamaha all have the same basic design in common. Flyweights are mounted to the movable sheave and work against rollers mounted on towers rotating with the shaft. This mounting tower is usually referred to as the "spider" due to its appearance. The spider tower also does double duty by transferring the torque to the movable sheave through "buttons" usually mounted outside on the roller pins.

Location of the roller in relation to the flyweight is critical for the shift curve. Mounting distance from the shaft in relation to the location of the flyweight pin is carefully determined to get a desirable offset between the two pivot points. This offset dimension cannot be changed, but the starting distance between the two pivot points along the shift axis can be changed. The relative distance between the two points during the shifting is dependent on the distance at the start of the shift. Most spiders are screwed on to the shaft, and the number of spider shims determines the distance. Shift characteristics can be influenced by stacking of the shims.



Spider washers are located between the spider and the shaft. Spider washers are used to control belt clearance, but they also have an effect on engagement and shift speed.

This is not a normal tuning procedure but we mention it because some racers remove the shims to get the roller further into the notch on a high engagement flyweight. When the roller is closer to the flyweight pivot, the shift speed will drop. When the roller is further away the shift speed will increase as a rule. The speed variation is in the order of 100 RPM per .062" shim, but is also influenced by the location of the center of gravity of the weight, and the change of radius it goes through. All this is not of great concern, but we mention it in case you wondered why your clutch shifted a little differently when you removed a spider shim.

Rollers are a critical part of transmission performance. Correctly working rollers give accurate and repeatable up and down shift, with quick response to changing load conditions. Bad rollers mean an inefficient clutch and poor performance. Roller bearings have gone through a lot of development in the last 10 years. Early designs had metal to metal bearings because this was the only thing that held up. Plastic bearings got crushed and fiber bushings wore out. Comet and Kawasaki arrived at better bearing life by increasing pin size from 1/4 to 5/16 and got more favorable loads. Polaris had a needle bearing set-up for racing, but it was sensitive to dirt and had to be cleaned often. The accepted bearing standard is now either a fiber bushing or a copper backed teflon bushing on a large diameter pin. Both the fiber and the teflon have the ability to function with contamination without seizing on the pin. As soon as the roller starts turning hard, the flyweight will start sliding on the surface and create flat spots which totally destroys the shift pattern. It is important that there is traction between the roller and flyweight, so keep lubrication far away from these surfaces.

While the rollers actually rotate several revolutions for each up and down shift, the flyweight only pivots through a quarter revolution at most. Bearings in the flyweight pivot are therefore not of as much concern as in the rollers. Some designs use a fiber bushing, but most rely on metal to metal contact.

CONTACT ANGLE

The contact angle between the roller and the flyweight determines the portion of the centrifugal force that will be converted to shift force. If the flyweight is pointed straight out and the flyweight surface is parallel to the shift, there will be no force left over for shifting. If the cam surface is at a right angle, most of the force is used to shift the clutch. Flyweights have curved cam surfaces to produce contact angles in between these two extreme positions to get the correct load from the centrifugal force. The contact angle is defined as in relationship to the shaft center line. Small angles convert only a small part of the centrifugal force to shift force, while large angles convert most of the centrifugal force to shift force. When we are tuning, we usually look at it in a reverse manner: **Small angles require more RPM to produce the same force, while larger angles require less RPM to produce the same force.** This last rule becomes important when we look at grinding flats and notches to increase engagement speed.

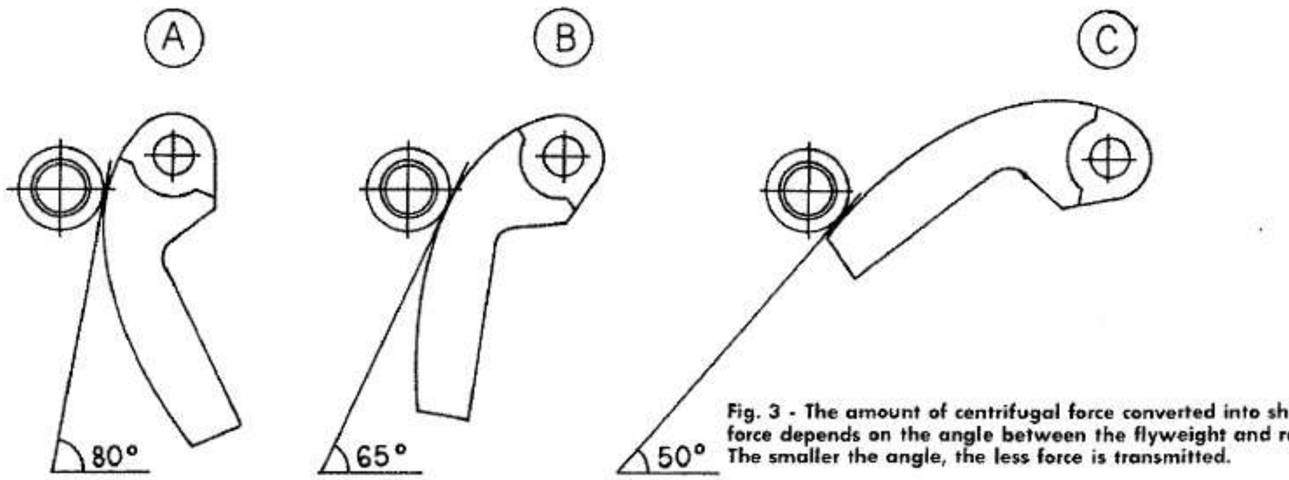


Fig. 3 - The amount of centrifugal force converted into shift force depends on the angle between the flyweight and roller. The smaller the angle, the less force is transmitted.

ENGAGEMENT

There are two ways to obtain the desirable engagement speed. One is to use the pretension of the pressure spring which opposes the flyweight. The other is to tune the flyweight system to overcome a given pretension at the desired RPM. **The rule should be to come as close as possible with spring selection without affecting the "straight shift" characteristic of the transmission.** With this approach you would not have to be quite as radical in the weight modification. There are two modifications that can be made to the flyweight to increase engagement speed. One is to modify the flyweight with a notch, and the other is referred to as tucking the center of gravity under the pivot point. Grinding notches is the preferred first step. "Tucking under" should only be used as a last step.

GRINDING NOTCHES

What we attempt to accomplish when grinding engagement notches or flats is to modify the angle with the roller. To increase the engagement we want less of an angle, and the only way we can accomplish this is by grinding into the body of the flyweight and creating a notch. Fig. 4 shows three different flyweights, one with a normal shift curve, one with a flat ground into it, and one with a notch. The difference can be seen in the contact angle with the roller. Creating a force to overcome the pretension of the spring requires much more RPM with the notched flyweight than with the standard one.

Grinding the notch becomes trial and error with repeated tests to judge the effect. The notch must always have a radius larger than the roller and be deep enough to prevent the roller from touching on the backside. With the roller touching the backside first (see Fig. 5) you would get a low initial engagement followed by a second high RPM condition, or what is referred to as a "double engagement". The only place this configuration is used is to get around a stock racing rule which requires a maximum 5000 RPM engagement.

When you grind the notch, you remove material and make the whole flyweight lighter. This could result in a higher shift speed. You may want to start out with a weight a couple of grams heavier to compensate for the material you plan to remove. You can only do so much with a notch and the next step becomes "the tuck".

"TUCKING UNDER"

"Tucking under" refers to the relative position of the flyweight's center of gravity to the pivot point. In the extreme case when the center of gravity of the

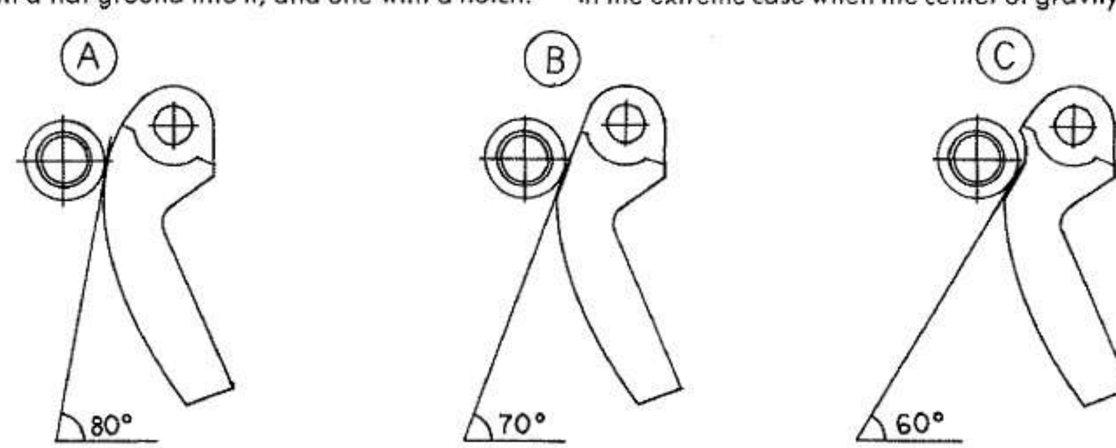


Fig. 4 - Engagement speed is dependent on the angle between the flyweight and roller. Weight A has a regular curvature with a standard engagement. Weight B has a flat ground to give higher engagement. Weight C has a notch ground to give high enough engagement for competition.

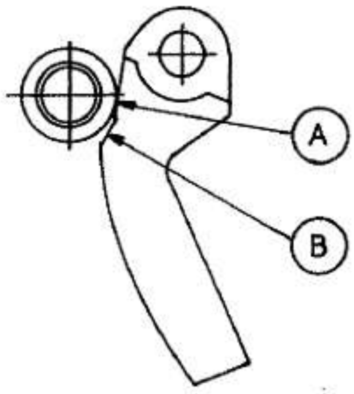


Fig. 5 - This weight has a "double engagement" or "cheater notch" as it is often called. The angle at A is large giving a normal engagement. As soon as the weight starts swinging out it hits the second angle at B and the RPM increases.

weight is directly under the pivot point, your engagement speed is unlimited. You can rev the motor as much as you want; there is no momentum arm to swing the flyweight out. The biggest shortcoming with"tucking under" is that the closer you get to being under the pivot point, the faster your engagement speed increases. With very small momentum arms the system becomes unstable and may not shift out the same way every time. Polaris used this system to adjust the engagement speed on their RXL racers. A set screw was installed under the heel of the flyweight to give different amounts of "tuck under." Since tucking the weight under also moves the sheave away from the belt, belt clearance was adjusted by using a number of different spider spacing rings.

A popular method of adjusting engagement speed on race sleds is an adjuster ring mounted on the center bolt and pushing on the outside cover. In the full out position the flyweight is at maximum tuck and highest engagement. As the cover is moved in by the threaded ring, engagement speed goes down as the flyweight swings further out to contact the roller.

On our 4-Star billet cover engagement speed can be adjusted up by tightening three allen screws and increasing pretension on the pressure spring.

BELT CLEARANCE

Some racers prefer to space the sheaves as close as .020" to the belt for a "smooth engagement." This can be done by removing spider shims, or, as with the RXL clutch, by changing spacers. The correct procedure is to set up the belt clearance first, and then tune the flyweight.

We have some reservations about tight belt clearances. This procedure was developed for the RXL racer which had a high revving engine and narrow power band. You did not want to break the track loose. Using the same procedure on an 800 cc drag racer with 1" studs only results in the belt burning, the engine screaming and you going nowhere. The reason is that the close belt clearance doesn't allow the flyweight to move out fast enough in the acceleration phase because it is still "tucked under." We prefer to work with a belt clearance of .050" for best all around results. We have used as much as .100" on some grass drag machines when we wanted the clutch to slam in hard and the track to throw a "roost."

TRANSITION

The transition between engagement flat and shift curve is often critical to a good shift-out at the correct shift RPM. This transition curve may extend as much as a quarter inch into the shift surface depending on your objective. We mentioned an initial over-rev use on some machines. This over-rev is intentional to compensate for a drop in RPM that would otherwise occur when a heavy machine suddenly starts to accelerate hard on shift-out. A normal over-rev would be in the area of 250 RPM and only last for a very short period until the acceleration had been established. We are referring to a time period as small as one or two seconds before the transmission settles in on the power peak.

This over-rev can be obtained by extending the engagement flat further down into the transition area. The angle of the curvature and the length into the transition determines the RPM and duration of the over-rev. Getting the flyweight right in this area is critical, as you don't want to be stuck in this condition longer than necessary before the machine accelerates hard. As little as a .060" can make a difference in performance.

In a high torque, low engagement machine there is usually no over-rev transition. Over-rev becomes more desirable in a racing machine with a narrow power band where a drop in RPM would mean falling completely off the power band.

I have seen cases where drag racers have made a curved notch with a sharp edge into the shift

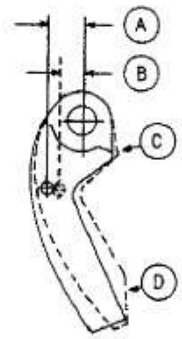


Fig. 6 - Tucking under is another popular method of increasing engagement speed. By making the flyweight swing further in, the center of gravity "tucks under" the pivot point more. The smaller leverage distance B will give a higher engagement than A. To accomplish more tuck, material can be removed at C on Polaris weights, and at D on Comets.

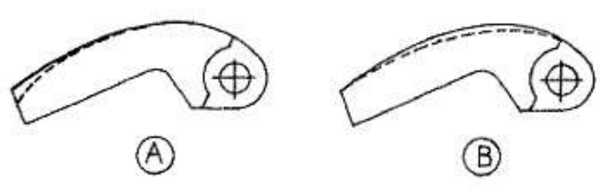


Fig. 7 - Grinding the weight by the dotted line in A will make it more aggressive and lower the RPM in high ratio. Grinding the weight by the dotted line in B will make it less aggressive and increase the RPM in mid-range.

curve that fit the roller. Once the roller got out of the notch the clutch would slam in hard as the roller went over the sharp edge onto the shift curve. This is only useful on some high power super light grass drag machines where you want to throw a roost. The sharp edge is too hard on rollers to be used for anything of longer duration.

Be careful when you experiment in the transition area, small changes can make a big difference on your shift out.

SHIFT CURVE

A straight shift is the ideal shift curve. The flyweight curvature is designed to obtain a straight shift from low ratio to overdrive. Each point on the shift surface has been developed to give the correct angle with the roller to convert a portion of the centrifugal force. This force is used to overcome the pressure spring, and to overcome the belt force from the driven clutch.

The resultant side force must match the side force curve of the driven clutch through the ratios. We have seen in the pressure spring section, how spring rate affects "straight shift". A shift ratio can be straightened out by selecting spring rates. When everything else has failed, you may try to grind the cam surfaces. Small changes in curvature can make big changes in RPM, so this is an area to progress in with caution.

Engine speed will depend on the angle of the curvature with the roller. The smaller the angle becomes with the center of the shaft, the more RPM it takes to generate shift force. Flyweights with flatter curvatures need to be heavier than flyweights with more curvature in order to hold the same engine speed. To make a flyweight flatter, you have to grind off in the middle and taper it towards each end. (Fig. 7B). To make a flyweight more aggressive you start at the transition and decrease the curvature radius toward the tip (Fig. 7A). In the first case you would increase the RPM toward the end of the shift, and in the second case you would decrease the RPM.

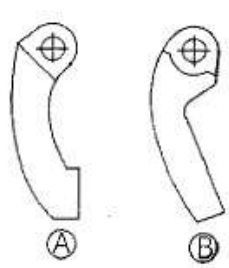
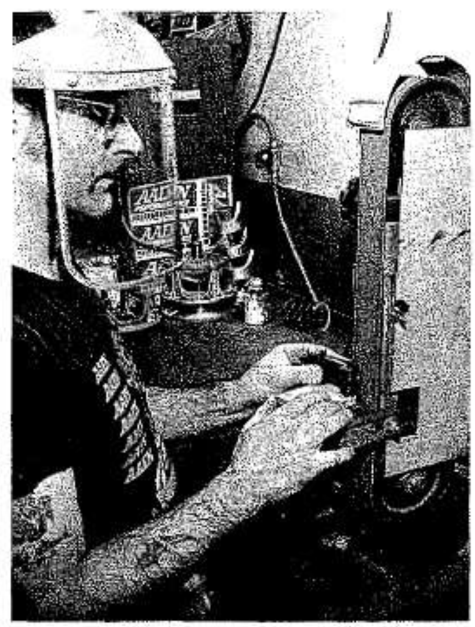


Fig. 8 - Weight A is heavier on the end and more aggressive in high ratio. Weight B is lighter on the end and is more aggressive in mid-range.

Weight distribution also has an effect on the shift characteristic. Flyweights that are heavy on the end (Fig. 8) tend to hold the RPM down, be more aggressive on top end and softer on the shift out. Flyweights that are heavy in the middle, and light on the tip tend to be aggressive on the shift out and rev more on top end. This has to do with the location of the center of gravity on the flyweight, and becomes important when you start grinding off weight. You must be careful where you remove the weight to get the result you desire. The preferred

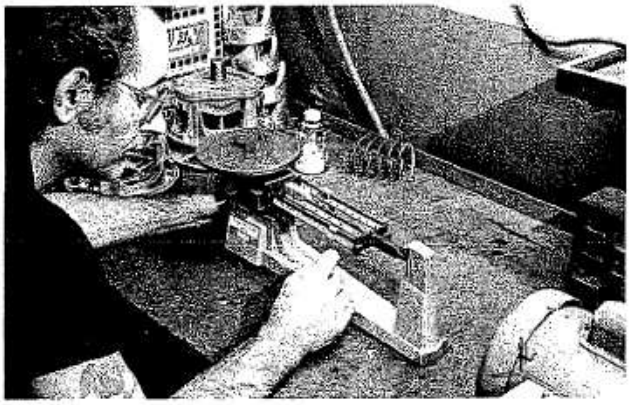


The flyweights are ground using a belt sander and a grinding block.

way to remove weight is on the sides of the weight to keep the center of gravity in the same place. Most manufacturers do exactly that, and provide flyweights with the same shape in a variety of thicknesses in 2 - 4 gram increments. For more information on the availablility of flyweights, check in the appendix for each brand.

GRINDING WEIGHTS

Before you start grinding on your weights, it's important to check the system for faults. There is little use in spending much time testing and grinding if there is a point in the transmission that creates high friction. The critical areas for the flyweight system are the roller bearings and the flyweight pivot. These bearing surfaces must be in good condition for a satisfactory result. Sloppy pivot points due to worn out bearings or a worn down

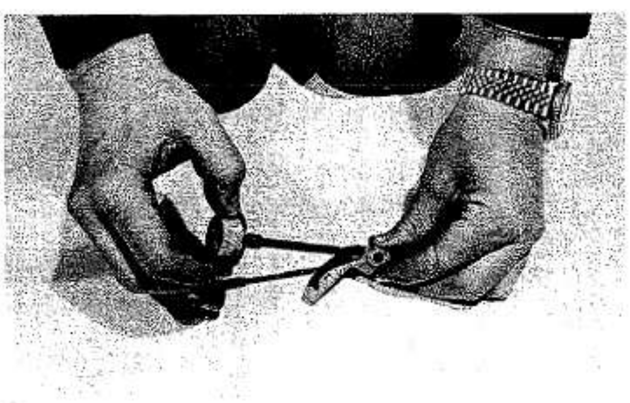


A Triple Beam Gramscale is a good tool to check the weights. This O'Haus scale is accurate to a tenth of a gram. Individual weights may vary 2 grams, and should be checked before they are matched in sets and ground.

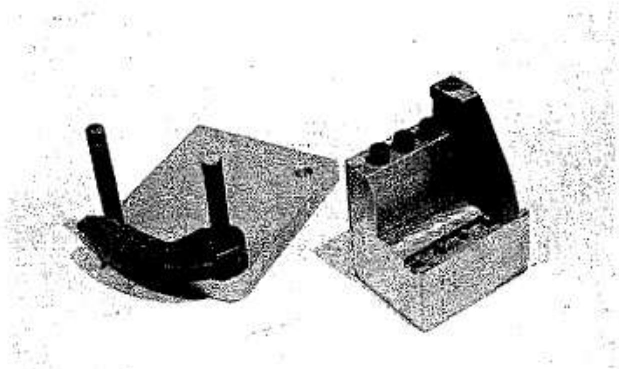
A pattern weight is placed on top of the weight to be ground, and the shape of the notch is transferred with a scribe.

pivot shaft can cause a number of problems. The flyweight will act erratically if its pivot and bushing have too much slop. Imbalance is one problem; flipping back and forth and hitting the spider tower is another. Sloppy flyweights also quickly wear down the roller bearings and put extra side thrust on both the roller and flyweight thrust washers. Once these washers get overheated, warped, and worn out, the roller on the flyweight starts digging directly into the aluminum around them. Loose aluminum shavings get in the bearings, rollers jam against sides, and friction abounds everywhere.

A small but important point has to do with the pivot bolts for the flyweights. A first time mistake is usually to tighten the bolts down. This is wrong for two reasons. First, if the bolts are tightened down, they cannot pivot in the aluminum housing, and that limits the bearing surface to the flyweight bushing itself. By tightening down the bolt you also risk squeezing the mounting surfaces together and pinching the weights so they don't move. In addition to these obvious disadvantages there is the danger that tightening down puts extra stress on the sheave which causes it to crack and the sheave may end up exploding. A good procedure is to use plenty of red Locktite, and get a new locknut every time. Snug the nut down until it touches the housing, then



We apply Dykem to the flyweight to mark the new curvature up before grinding.



To get accuracy when sanding ramps, use a ramp grinding block. The block on the left takes Polaris or Comet weights. The block on the right is designed for Arctic ramps.

back it off an eighth of a turn so the bolt rotates in the mounting boss. Make sure the thrust washers are in place and in good shape.

It pays to have the right tools when grinding on the weights. The flyweight surfaces have to be parallel with the pivot bolt, otherwise only one edge will contact the roller. This creates a side load and quickly ruins the roller and the thrust washers. To make sure all three weights are ground the same, we use a belt sander and a grinding block. The belt sander is squared up and the three weights held square in the grinding block.

Before grinding the weights we check them on a 3-Beam O'Haus Gramscale. Most flyweights are pressed from powdered metal and the density can vary by 2 grams per weight. Weights are sorted to within a half gram, and then we take a little off the sides to get them within a tenth of a gram of each other. Anybody that is serious about clutch tuning should have a good gramscale, a belt sander and a grinding block. The total price for all three is about the same as a new clutch but, the equipment will quickly pay for itself with the time you save.

Before we start grinding we use some blue Dykem tool dye on the top flyweight. We then use a scribe to draw up the new surface we want. We keep good weights which we know work as templates, but we never leave this weight stacked with the others when we grind. You may think you only grind down to it, but if you only removed ten thousandths off the master every time, it would be worthless in two to three uses.

We keep a small container with water next to the grinder to cool off the weights as we grind. You do not want the weights to get so hot that you cannot touch them with your hand, or the hardness will go out of them. Also be careful never to grind an area too thin or leave sharp notches that may weaken the weight and lead to a failure.

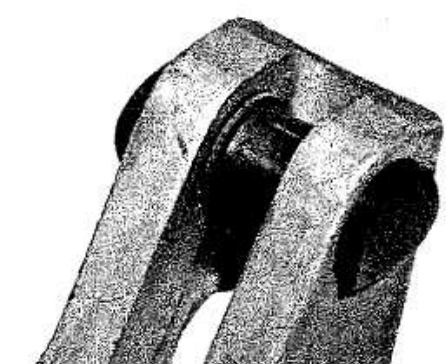
After grinding check the weights again and make sure they are within a tenth of a gram. If they are not, trim off the sides until they are. Never trim off the heel of the weight. They may end up the same weight, but the center of gravity will be different. When grinding on the shift surface you only want to go in small increments of ten thousandths between each test, it's easy to overdo it.

Keep careful notes of what you are doing. Only 10% ever keep good notes, and they win 90% of the races. We shall look closer at record keeping in the test section.

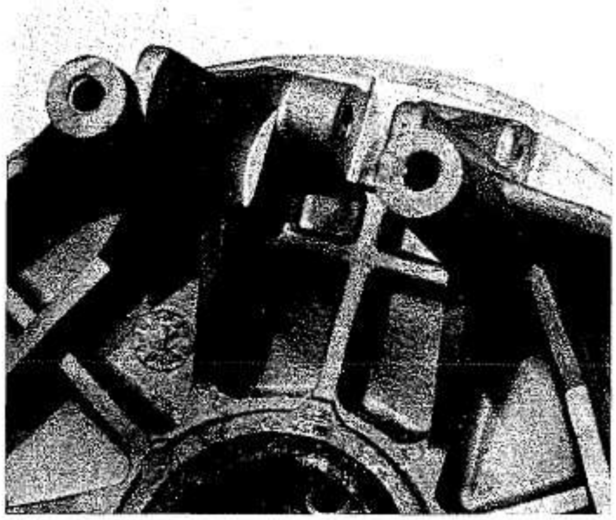
FRICTION POINTS AND BACKSHIFTING

The true test of a good clutch comes when it has to backshift in response to increases in load. Coming out of a corner, climbing a hill or hitting loose deep powder snow puts demands on the transmission to quickly shift down to a lower ratio. The transmission should preferably remain at the same shift speed on down-shift as on up-shift.

The main obstacle to a perfect clutch, that keeps the engine at the same speed on up and down shift, is friction. Friction works against the direction of travel so it would require more force to shift up, and the force would have to drop more to shift down. Once the flyweight system is calibrated, the only variable left is the speed. To overcome friction, the centrifugal force has to change by changing the speed. Large friction losses therefore show up in large speed differences between up and down shift. When clutches get real worn they simply refuse to back shift and you start to get stuck. A good clutch will back shift within one to two hundred RPM of the up shift, but it took a lot of development and design changes to get to where we are today. The biggest problem was to transfer torque from the shaft to the movable housing on the driving clutch. As the clutch shifts up and down, the housing has to move freely while it also delivers half of the torque to the belt. The other half of the torque is delivered by the stationary sheave.



Torque transfer buttons are placed on the spider on the outside of the roller pin, and slide against surfaces on the movable sheave.



Sliding surfaces for the torque transfer buttons are machined into the towers on the movable sheave.

On early transmissions the housing would ride on splines on the shaft. In some designs, fiber, hex and square bushings were used. The biggest problem with transferring the torque through splines and bushings was the small radius they worked at. Torque is the twisting momentum in ft.-lbs., and the smaller the radius, the higher the load is for a given torque.

Let's look at 50 ft.-lbs. of torque as an example. Splines on a one inch shaft would work on a half inch radius. The force needed at half an inch would be 25 multiplied by 12 to get inch-lbs., and then divided by .5 to get the force at half an inch. We use 25 lbs. because half of the torque is transferred through the movable sheave. The total load on the spline would be 600 lbs. Friction is load times the co-efficient of friction, which in the case of steel against steel is around .15. Friction load against the movement would be 600 x .15 = 90 lbs. with a spline design.

These early spline clutches were notorious for bad back shifting, and sometimes the up and down shift could vary with as much as 500 RPM.

Polaris was the first to come up with a good solution. Polaris engineers reasoned that the further away from the center the torque transfer point was located, the less force and less friction there would be. The new clutch which today is the model for many other brands had the torque transfer point moved out to the top of the spider tower, and used plastic buttons against aluminum for the sliding surface. With the torque transfer point now being 2.75 inches from center, the force was reduced by over five times from the spline design. The new load with over 50 ft.-lbs. of torque would be:

$$\frac{25 \times 12}{2.75} = 109 \text{ lbs. at the 2.75 radius}$$

Plastic against aluminum also has a lower co-efficient of friction than steel against steel. With a 109 lbs. load and a new coefficient of friction of around .10 the total friction load would be 109 x .10 = 11 lbs.

By changing the location and using different material, the friction against movement of the sheave was reduced from 90 lbs. to 11 lbs. The total friction load was reduced by 88%. This development greatly improved backshifting, and it also freed up the shaft bushings to only deal with sliding rather than also having to transfer torque. Another advantage was that five thousandths of wear on the large radius was not nearly as critical as on the spline. The buttons were also easy to replace, where worn out splines usually required a new clutch.

Other friction areas are the sliding bushings on both the driver and driven clutch. These are now either made of impregnated bronze, fiber or teflon-coated bronze. They need cleaning at regular intervals as dirt and belt dust get stuck on the surfaces and may cause them to jam up. As these bushings wear, they reach a point where the sliding sheaves can cock and get stuck. An easy way to check for this is to push down on the outer edge of the movable sheave. If the sheave still moves down, bearings are good. If the sheave twists and binds, it's time for new bearings. The latest development in our quest to reduce friction is our Four Star primary roller clutch (see page 73). In this unit the torque transfer buttons are removed and replaced with ball bearing rollers running in machined channels. This removes the button friction and the result is quicker reaction to load changes and a very smooth engagement. With friction eliminated, considerably lighter weights had to be used for correct calibration.

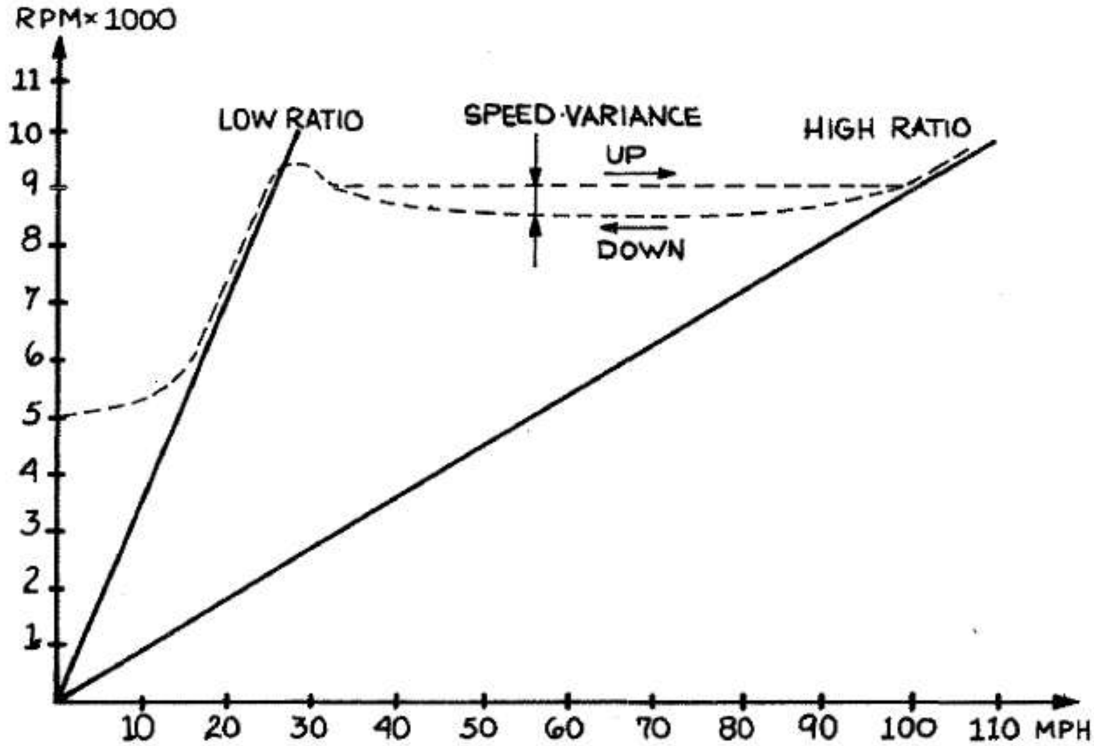
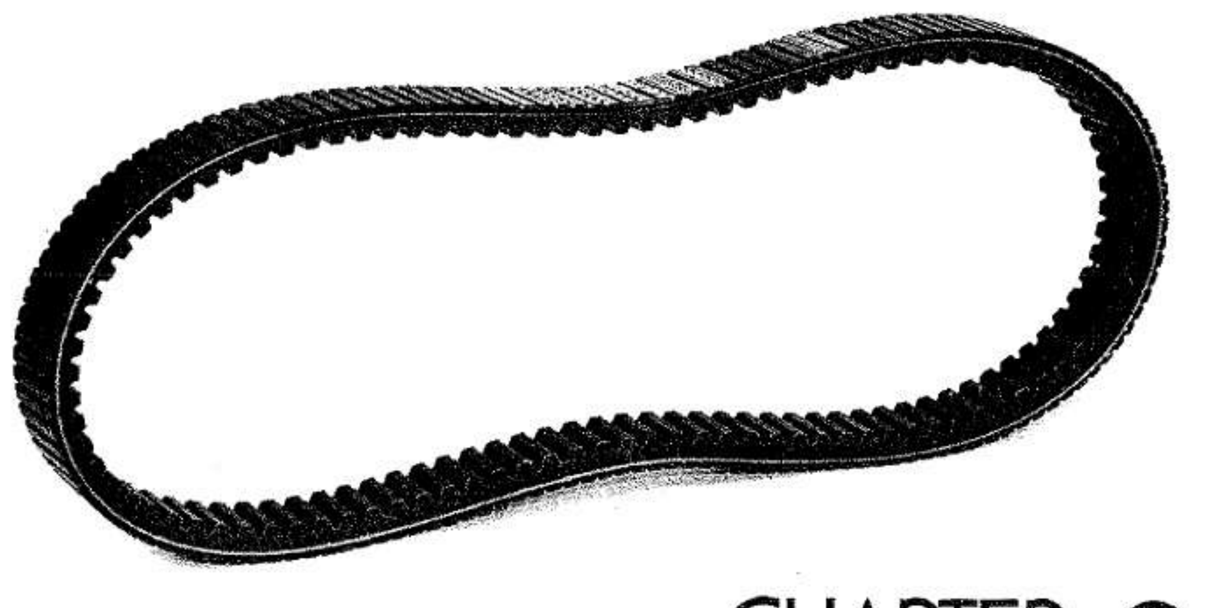


Fig. 9 - Friction has a large effect on back shifting. With worn parts and higher friction the transmission will shift back at a lower RPM giving a speed variable between up and down shift.

All the friction points are important to the back shifting of the transmission. They all change gradually, and you may not notice that things are getting bad until backshifting deteriorates and you get stuck. Tightening up the driven spring only moves up the RPM, it does not change the difference in RPM between up and down shift caused by friction. A tighter rear clutch actually increases friction and makes the clutch deteriorate faster. The only solution is a complete overhaul of the transmission to correct the problem.

Tuning clutches with bad friction points is a waste of time. Make sure the clutches are in top shape. Much of the success of race drivers has to do with maintenance. They spend a lot of time dialing in flyweights, and don't want to lose because some parts are worn and cause excessive friction.



CHAPTER 8

Today's Kevlar reinforced "Top Cog"™ belts transfer over 300 HP with reliability

The Belt

Today's racing belts transmit over 300 HP with efficiency and consistency. The development of drive belts has been amazing in the last 20 years. It used to be that broken belts were commonplace, and horsepower over 40 almost impossible to transmit.

The earlier belts were made with fiberglass cords which were stiff and had poor adhesion with rubber. Then a new material called Kevlar or Fiber "B" started to make inroads. Engineers found that 45 gauge Kevlar cord was just as strong as a 90 gauge fiberglass cord and gave the belts much more flexibility.

The use of Kevlar increased both efficiency and strength and is today the only material used in racing belts. Other such factors as special bonding procedure, rubber material and cog design also influence the efficiency and strength of the belt.

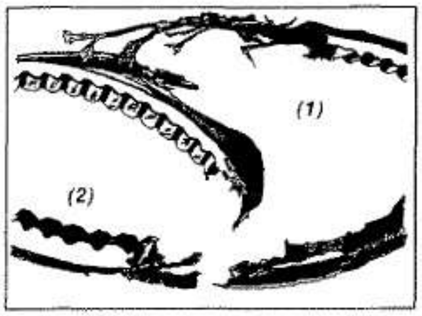
The latest developments are the new Dayco "Top Cog"™ belt designs. Instead of a fabric cover, the top of the belt has small cogs or ribs running across them. This improves flexibility and therefore efficiency, and also promotes better cooling of the belt by providing more surface area. New compounds, like the harder "CVT Compound" are also proving to hold up better on higher horsepower machines where smearing of rubber on the sheaves and burning the belt on takeoff was a problem.

To insure efficiency and good power transfer, it is important that the sheave surfaces are free from grease, oil or rubber deposits and are smooth. Sheaves should be sanded down at regular intervals with a fine grit emery paper to insure the best working conditions for the belt.

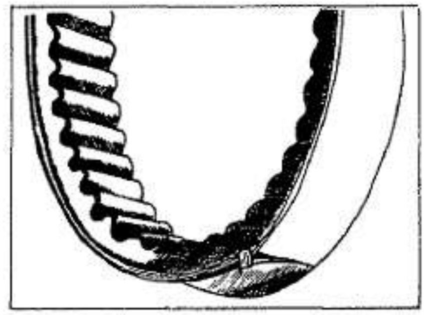
Excessive heat buildup in the sheaves or the air around them adversly affects the strength and the efficiency. and it pays to have the clutch area well ventilated for this reason.

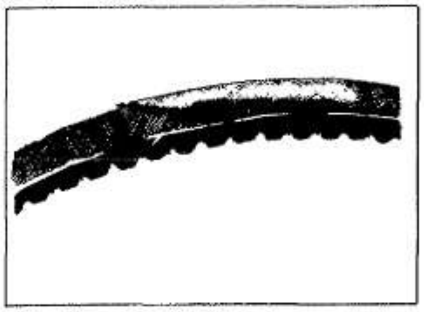
A new belt should be run-in for a short period before it is used for full power. This can be done on a warmup stand or during a short easy run. An experienced race driver will buy at least half a dozen belts at a time and after run-in measure the length of the belt. Belts made by the same manufacturer may vary as much as 1/2" in circumference. Belts of close length are then paired together and the rotating direction and length marked on the belt with a felt tip pen. This insures

Broken Belt/ Flex Fatigue
Usually caused by torque loads which exceed the tensile strength of the belt. Note fiberglass breaks with cords exposed (1), which can badly bind in the clutch. Kevlar Fiber B breaks clean and is easier to remove (2).

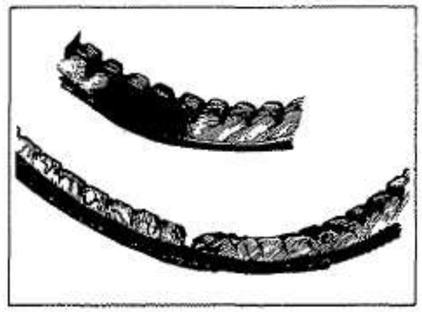


Overcord Fabric Cracking
Can be caused by excessive heat, mechanical damage or manufacturing flaw. Overcord will continue to peel and separate if not replaced.





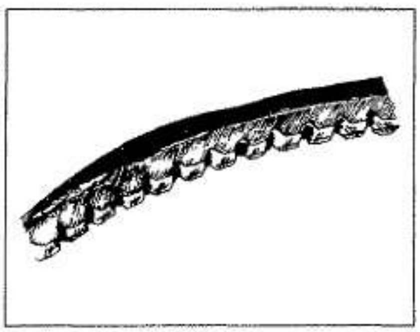
Overcord Fabric Loss
Often associated with edge cord separation, or belt being too long for the drive system, or belt coming into contact with the superstructure surrounding the drive units.



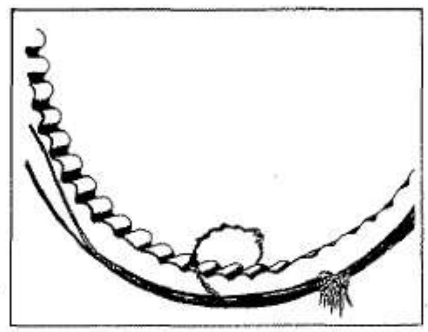
Spin Burn
Caused by excessive slipping of the belt.

Undercord Cracking/ Cog Loss
Normal worn-out condition, possibly induced by excessively high operating temperatures.

Severe Sidewall Wear
Caused by misaligned clutch sheaves, loose engine mounts, sheave face damage or use of the incorrect belt size.



Edge Cord Separation
Many causes, including misalignment of clutches, rough sheave faces, incorrect belt-pulley angle which doesn't allow the belt to ride fully on its sidewalls.



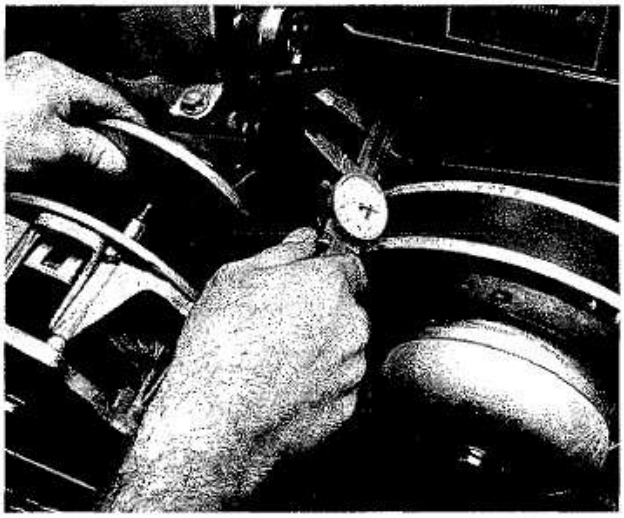
that once the center distance is adjusted for these belts, you will have more consistency in your selection.

As the sides of the belts wear, it will pull further into the driven sheaves in low gear and loop further out around the shaft on the driven sheave. When this becomes excessive, the clearance between the sheave and the belt becomes larger before engagement and the position of the belt puts it in a higher ratio on engagement. The result is that the sheaves slam into the belt with a jerking motion, but since the system is now in "2nd gear", it tends to pull the speed down and bog the machine off the line before it starts to shift out. This can all be corrected by installing a new belt with the correct length and width.

We usually consider a sixteenth of an inch too much wear for racing and change the belt. When all the belts are worn the same amount, we reset the driven clearance and use the belts over. After the belts are worn an eighth of an inch they are usually too narrow to be used further. Many driven clutches now have shims between the sheaves that you can remove to accomodate a narrower belt. This shimming can also be used to set the right belt tension on a new belt. As we mentioned before, belts can vary in length and width, and you can correct by changing the center distance or shimming the driven sheaves. It's easier to shim the driven sheaves, and this is why it is becoming more popular. The correct center distance should give a belt deflection of approximately an inch. Keeping the center distance too long makes the belt grind around the driving hub and the machine tends to creep. Too short of a center distance means the belt is too far out on the driven and you almost start in "second gear" with a bog and bad low end performance as a result.

In my years of fixing transmissions that don't work I have seen more than enough cases where a new belt fixed the problem. In the majority of other cases a good cleaning did wonders.

It is important to pay attention to these details as they may make the difference between winning or losing. Always keep a close eye on the belt's condition to spot cuts, cracks, frayed cords or other



To check the alignment, use a straight edge and a caliper. Measuring in front and back of the driven clutch gives you both the offset and any misalignment of the engine.

damage. The pictures on the following pages and the drive belt maintenance chart should give you a guideline to solve any of your belt problems.

ALIGNMENT

The correct alignment of the two clutches is critical for performance and belt life. A misaligned clutch puts an extra load on the belt and can drastically shorten belt life. There are three ways a belt can be misaligned. The first is to have an incorrect offset (Fig. 1). In this case the engine and secondary shafts are parallel but the driven sheave is offset too far out. This can be corrected by removing shims and spacers to get it back to the correct measurement. All manufacturers have a recommended measurement considered correct for their transmission. Some have special tools to set up the offset, others recommend placing a straight edge behind the driving clutch and measuring the offset to the driven clutch.

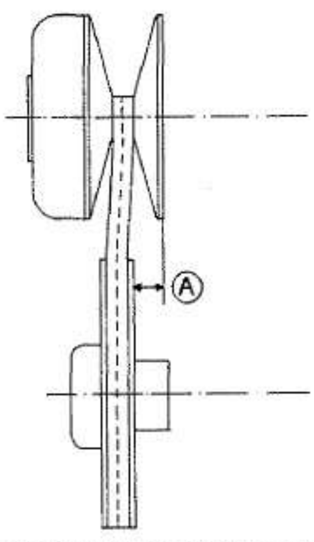


Fig. 1 - This transmission has been set up with too much of an offset (A), resulting in a twist in the belt.

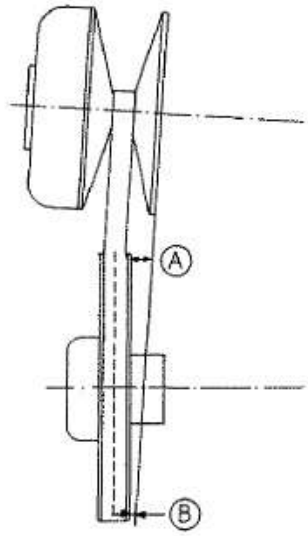


Fig. 2 - This transmission has the engine misaligned horizontally with the driven shaft. Measuring at A and B gives an indication of the misalignment.

This method also has the advantage of correcting misalignments which are the result of the engine and secondary shafts not being parallel. (Fig. 2) If the measurements are not the same at the front and rear of the driven sheaves, the engine and secondary shaft are not parallel. To make them parallel the engine has to be twisted on the mounts. Sometimes the engine is misaligned on purpose to compensate for twisting on the rubber mounts. The engine is then usually twisted a couple of degrees forward on the drive side. As the torque load increases the engine will be pulled back into parallel. The biggest draw back with this is that when the transmission shifts into high ratio, the engine twists back into misalignment again as the belt pull diminishes.

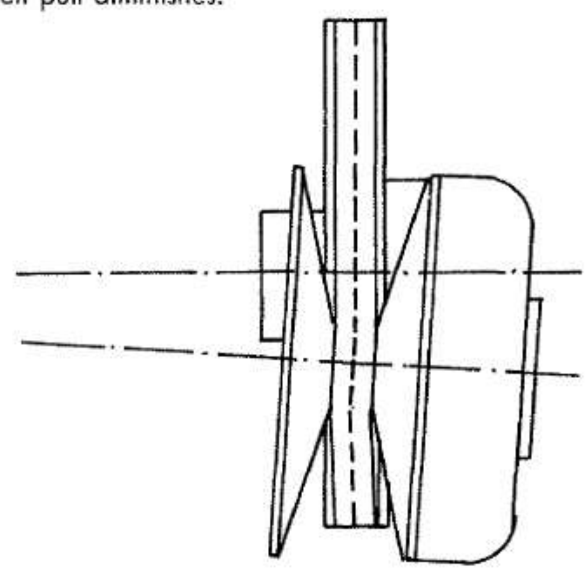
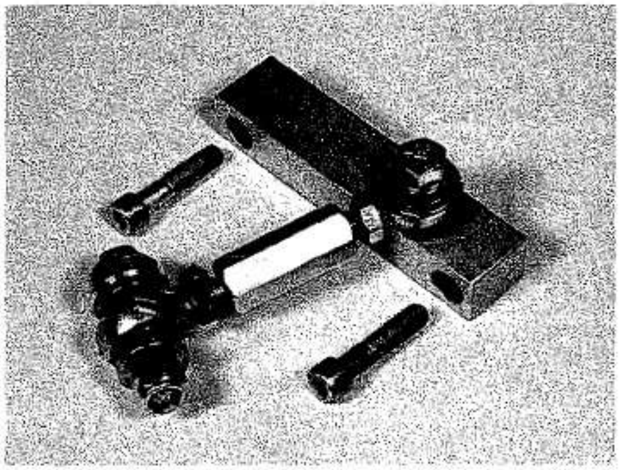


Fig. 3 - This transmission has the engine too high on the starter side, causing a twist in the belt. By measuring on top and bottom the misalignment can be checked.

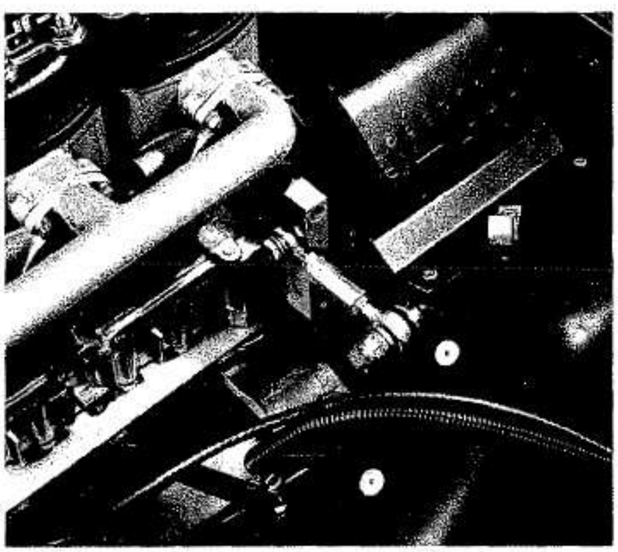


A torque strap with two rod ends ties the engine to the chassis and prevents misalignment under load.

Racers don't have this problem when they mount the engine solid, but this is not practical on trail machines because too much of the vibration is transferred into the chassis.

The solution to this problem is called a torque arm. This arm connects the engine and the chassis, preferably close to the driving clutch. A heim joint on each end makes it possible for the engine to move up and down on the rubber mounts, but the arm prevents the engine from twisting. A torque arm is a good investment. They usually cost no more than a belt, and they pay for themselves in a short time through less belt wear and better performance.

The last type of misalignment occurs when the engine and drive shaft are not parallel horizontally, as viewed from the front. (Fig. 3) You can measure this by using the straight edge again and compare measurements at the top and bottom of the two clutches. In this case the engine mounts must be shimmed on one side to correct the condition.

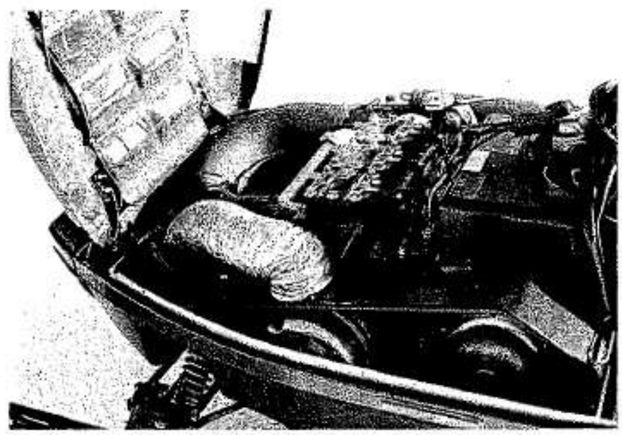


The torque arm installed on a Polaris Indy, tying the engine to the shock tower.

Belts go through a lot of bending and stretching on their normal path, and being bent sideways twice per revolution is the last thing they need. A lot of power can be lost due to bad misalignments, and it's a source for a high rate of belt wear if not corrected.

COOLING

Power loss usually disappears in the form of heat. Heat is generated in the belt from the bending, and between the belt and sheave as power is transferred. The hotter the belt gets, the weaker it gets. Rubber also loses its friction properties after too much heat is developed. The first great step in keeping the belt cool was the introduction of aluminum sheaves. Not only did they remove heat better, they were also lighter and less costly to manufacture. Today all clutches have aluminum sheaves. To remove heat from the sheaves, ribs were added to the driver, and the driver was run open. There is some power loss in this method as you are actually powering two fans spinning up to 10,000 RPM. On some lower power machines "windage shields" are kept on the driver, and some manufacturers go to great lengths to put holes in strategic locations on the cover of their driving clutches.

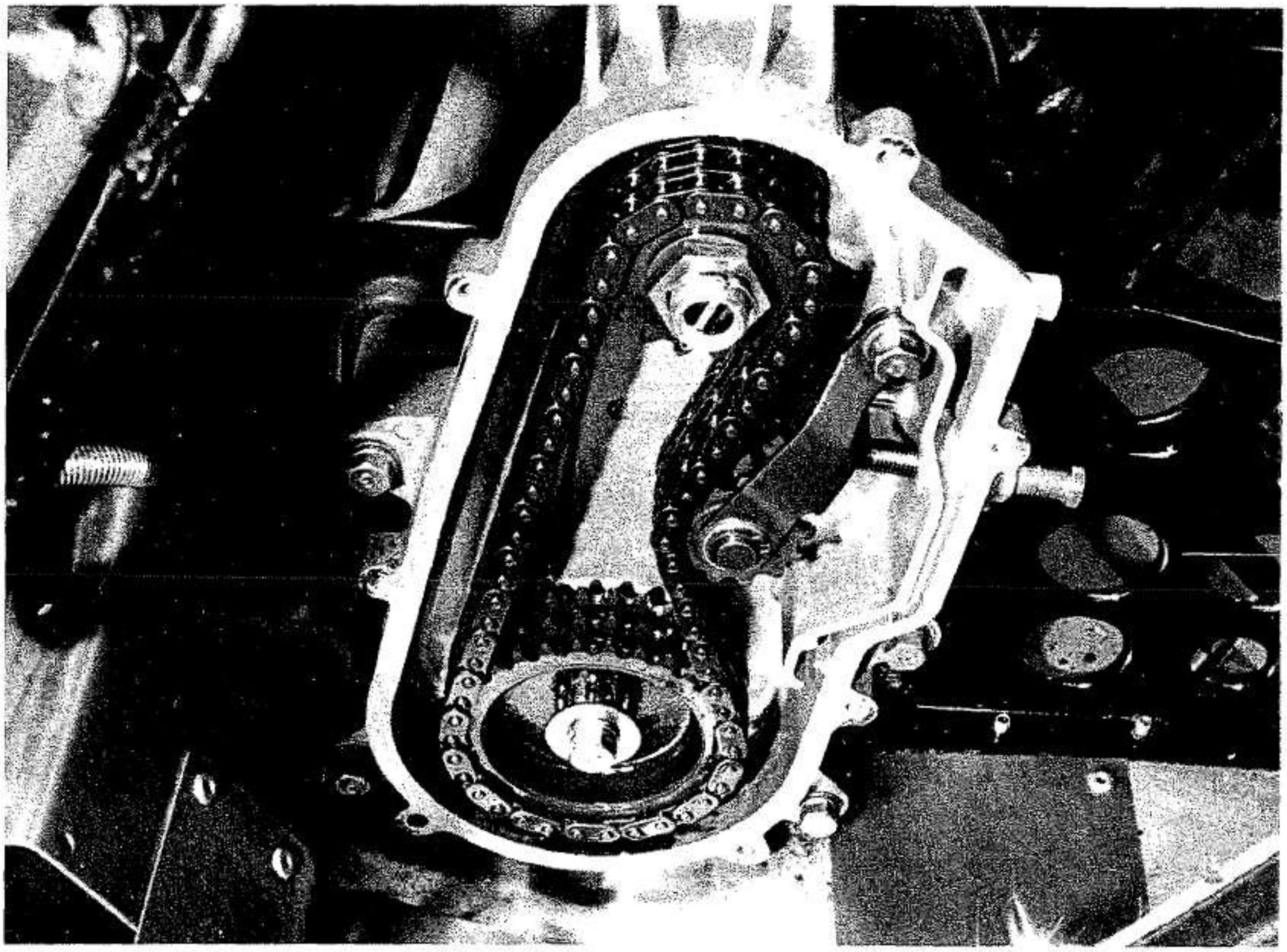


This driver installed an extra cooling duct to make sure the belt stayed cool.

Using air ducts is also popular to get cold air circulating around the clutches. Polaris's new ATV has a fully enclosed transmission. Air is circulated around the transmission with an inlet in the front tube, a fan designed into the back side of the driver, and an outlet up under the tank. This helps keep the transmission cooled and free from mud, dirt and water. High underhood temperatures can also cause deterioration of the belt, especially on fan cooled models. Ducting the hot air from the engine outside with a separate duct not only helps belt life, but gives better performance because the engine runs cooler. Doing whatever you can to keep the clutches and belts cooler will give you more efficiency and better belt life.

DRIVE BELT MAINTENANCE CHART

The Problem	Causes	Treatments
1) Uneven belt wear on one side only	a) Pulley misalignment b) Loose engine mount(s) c) Rough or scratched pulley surface	a) Align pulleys
2) Belt glazed excessively or has baked appearance	Excessive slippage caused by: a) Insufficient pressure on belt sides b) Excessive horsepower for belt and converter c) Excessive oil on pulley surfaces d) Insufficient pre-load on driven spring e) Excessive operation in low gear position	a) Check driver pulley for smooth actuation b) Consult dealer c) Check bearing seals and clean pulley surfaces d) Consult Operator's Manual e) Inspect converter
3) Belt worn excessively in top width	a) Excessive slippage b) Rough or scratched pulley surface c) Improper belt angle d) Considerable use, belt wearing out	a) Check driver pulley for smooth actuation b) Grind or polish pulley(s) c) Consult dealer d) Replace belt
4) Belt worn narrow in one section	Excessive slippage in driver pulley caused by: a) Locked track b) Converter not functioning properly c) Engine idle speed too high	a) Rotate track by hand until free b) Repair or replace converter c) Reduce engine RPM
5) Belt too tight during engine idle	a) Idle speed too high b) Incorrect belt or belt length c) Incorrect drive center distance d) Idler bearing seized	a) Reduce engine RPM b) See Operator's Manual c) See Operator's Manual and reduce center distance d) Replace bearing
6) Concave worn belt side(s)	a) Excessive ride-out on driver pulley b) Drive misalignment c) Rough or scratched pulley(s) surface d) Excessive slippage	a) Repair or replace driver pulley b) Align pulleys c) Grind or polish pulleys d) Repair or replace driver pulley
7) Belt disintegration	a) Excessive belt speed b) Sheave misalignment causing belt flip-over c) Excessive slippage causing heat build-up in belt d) Excessive operation in low gear position	a) Reduce engine RPM at high speed b) Align sheaves c) Inspect converter d) Inspect converter
8) Belt "flip-over" at high speed	a) Pulley misalignment b) Excessive belt speed c) Excessive ride-out on driver pulley d) Incorrect belt length	a) Align pulleys b) Reduce engine RPM c) Repair or replace driver pulley d) See Operator's Manual
9) Belt edge cord broken	a) Pulley misalignment b) Improper belt installation c) Engagement speed too high	a) Align pulleys b) See Operator's Manual c) Reduce engagement speed
10) Flex cracks between cogs	a) Considerable use, belt wearing out b) Bent pulley(s) flange causing belt flutter c) Excessive operation in low gear position d) Extremely low temperature	a) Replace belt b) Repair or replace pulley c) Inspect converter d) Warm up belt slowly
11) Sheared cogs, compression section fractured or torn	a) Improper belt installation b) Belt rubbing stationary object c) Idler bearing seized	a) See Operator's Manual b) Check drive components c) Replace bearing
12) Broken belt	a) Engagement RPM too high b) Belt hanging up in bottom of driven pulley c) Locked track	a) Reduce engagement RPM b) Belt too short; replace c) Rotate track by hand until free



Extra performance can be gained by choosing the correct ratio in the chain case.

CHAPTER 9

Gearing

The correct gearing depends on a number of variables such as engine power, over all weight of machine and driver, and the type of course the machine is used on. Someone going for top end only would use a different philosophy than an Oval or Sno Cross racer. Gearing too low may be as disastrous as gearing too tall.

Drag racers and speed runners are interested in good top end, and their machines are usually light with big engines. These machines end up being geared about 25% taller than the speed they will achieve. If they are shooting for 110 MPH, they may be geared for 130 MPH. There are a couple of good reasons for this. If you look again at the efficiency curve, (Fig. 1) you will see that the efficiency drops off the further into overdrive you get. To get maximum top end you need as much horsepower as possible. If 1:1 ratio is 10% more efficient than overdrive, you would attain more top speed by gearing tall and never getting into overdrive. With a 100 HP engine, 1:1 would give you 90 HP, while overdrive would give you only 80 HP.

Gearing tall may have some penalties in acceleration depending on the power to weight ratio. On super light drag racers acceleration actually increases when gearing tall, because less

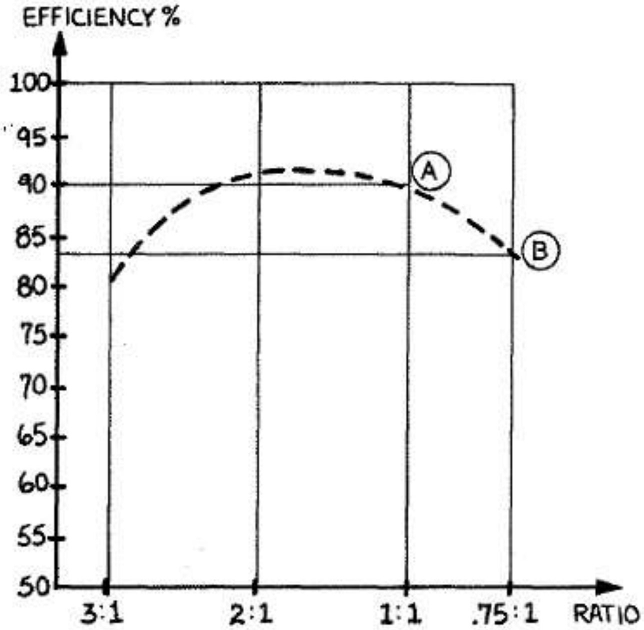


Fig. 1 - For drag racing and speed runs you gear tall to use maximum efficiency. Gearing tall and shifting to A gives you 7% more power than gearing low and shifting to B.

time is spent revving the engine and more on moving forward. Most successful drag racers with a 220 lb. chassis and a 110 HP 440 engine would gear for 130 MPH in 750 feet, although they would only reach 105. Drag racers arrived at this by experimenting with gearing. It turned out that acceleration increased and top end got better because they were on a more efficient part of the shift curve.

The same would not be the case for a 450 lb. stocker with 50 HP. In this case you could not gear as tall because you would lose acceleration.

Our rule of thumb is to gear as tall as you can without losing acceleration.

In Sno Cross and Oval racing you are more concerned about having good power out of the

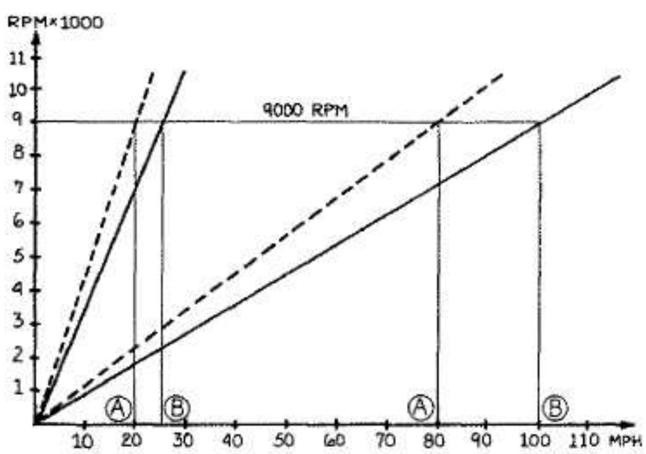


Fig. 2 - With a 4:1 overall shift ratio there is not a great penalty for gearing tall. Changing the gearing from A to B gains 20 MPH on top end but only crosses the 9000 RPM line 5 MPH higher on the low end.

turns and a good start, and you may have better luck with a more "normal" gearing for overall results. This is highly dependent on the course layout. A small track with tight curves obviously needs a lower gearing than a larger super Oval with banked turns and long straights.

A peaky high strung engine needs lower gearing than a low revving torque with a wide power band.

To figure out top speed, shift ratio, or gearing, use the following formulas:

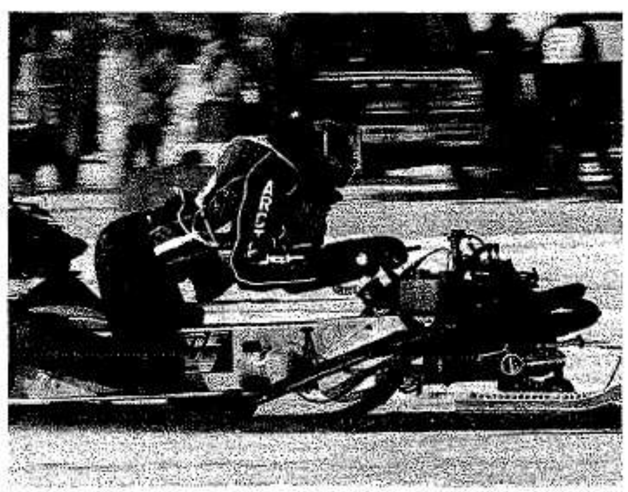
$$(1)\ \text{MPH} = \frac{\text{engine speed} \times \text{sprocket pitch diam}}{\text{shift ratio} \times \text{gearing} \times 336}$$

$$(2)\ \text{Shift ratio} = \frac{\text{engine speed} \times \text{sprocket pitch diam}}{\text{MPH} \times \text{gearing} \times 336}$$

$$(3)\ \text{Gearing} = \frac{\text{engine speed} \times \text{sprocket pitch diam}}{\text{shift ratio} \times \text{MPH} \times 336}$$

Engine speed = RPM (revolutions per minute).
Sprocket pitch diam (inches) = sprocket diameter + track thickness.
Gearing = driven sprockets divided by driving sprocket.
Shift ratio = driven clutch belt radius divided by driving clutch belt radius. Usually 3:1 in low and 0.75:1 in high.

Using the above formulas you should be able to figure out the gearing you want depending on the speed you want to reach and the shift ratio you want to be in.



Drag racers gear tall with the super light machines to get more top end.

Let's take a drag racer with a 440 engine spinning at 10,000 RPM. He wants to hit 105 MPH in 750 ft., and he wants to be in 1:1 shift ratio at the end for maximum efficiency. His track sprocket is 6 inches in diameter and the track is 3/8" thick. Using

Oval racers change gearing according to the track size. Accelerating quickly out of corners is high on the priority list.

formula 3 his gear ratio would then be 1.8.

$$\text{Gearing} = \frac{10{,}000 \times 6.375}{1 \times 105 \times 336} = 1.8$$

With a 36 tooth bottom sprocket his top sprocket would be: 36/1.8 × 20 tooth.

How realistic is the above example? A successful 440 drag sled campaigned to a world series championship was geared 21-39, giving a 1.86 gear ratio. This machine was geared tall to take advantage of the better efficiency in 1:1 ratio, and it was light enough that it did not hurt acceleration. Theoretical top speed in a .75 overdrive would be 105/.75 × 140 MPH. This approach also works on speed sleds running 1000 ft. or quarter mile.

We have tried this same approach on Oval races without much luck. A Formula 1 Oval machine is 150 lbs. heavier and has a smaller 340 engine with a peakier power band. The speed you would gain at the end of the straight would not make up for the distance you lost in acceleration coming out of the turn and down the first half of the straight.

Sno cross machines need to negotiate tight turns and moguls. Quick response and back shifting becomes important, and top speed is seldom ever used.

For the normal trail machine, the gearing is usually set up to be about 10 MPH taller than the machine would reach. This gives good low end, but prevents the belt from spending too much time at high belt speeds which may shorten life.

When the machine has been modified, we face

Sno-Cross racers need plenty of pull low down to accelerate out of tight corners.

another problem. If the engine speed has been increased from 8000 to 9000, the machine is automatically geared 11% higher. The higher engine speed now may require a lower gearing depending on what the original ratio was. Sometimes you may want to drop the top sprocket 1 tooth or more after installing the modified parts. To

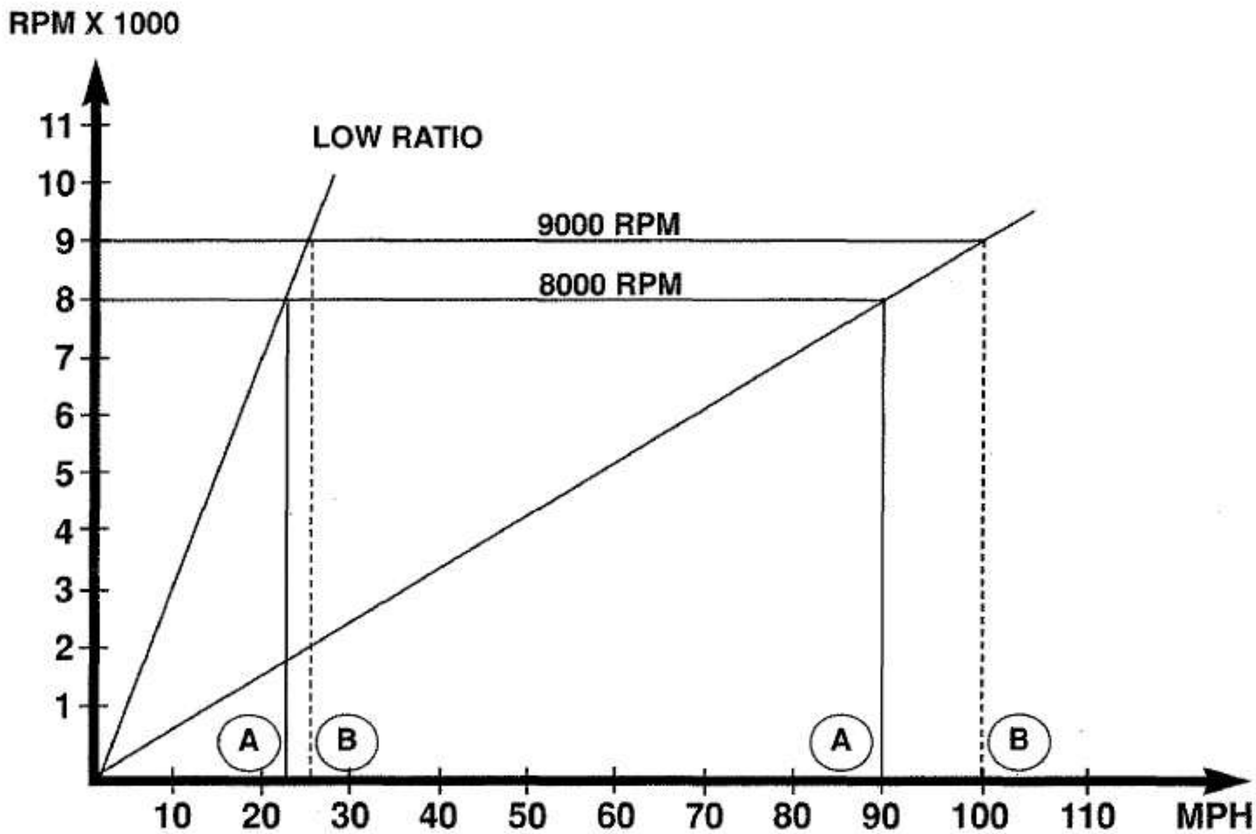


Fig. 3 - Modifying an engine and moving the powr peak from 8000 RPM (A) to 9000 RPM (B) has the same effect as gearing taller. Machines that have been modified with higher RPM sometimes have to install a smaller gear on top to compensate.

find your new ratio it is convenient to use the formulas.

FULL SHIFT OUT PROBLEMS

I often hear guys complain that they can't get the belt to come all the way up on the driving sheave. Many use a black magic marker on the sheave, and by looking at what's left after a little running, they can determine how high up the belt came. If the machine is geared 10% higher than it will ever run, the belt will never get to the top. To get the belt to reach the top, you would have to lower the gearing.

You may however have run into a genuine assembly problem that will not permit the belt to shift all the way out, and some of these can cause serious belt breakage problems. If your secondary does not open all the way up but stops short of a full shift, the belt is not going to shift all the way out. What will happen next is that the engine will over-rev and as the flyweights put more force on the belt, it may stretch and actually snap. You do not want the secondary to stop shifting before the front clutch reaches the end of its travel, because of the extra stress this puts on the belt.

An even worse condition occurs when the front clutch actually travels too far and pushes the belt all the way out of the sheaves. The belt may then flip over which locks up the transmission and the track at top speed. This may occur if the spider has been spaced incorrectly for a widebelt application.

Another source of insufficient shiftout may be springs that have been shimmed so much that they coil bind before the clutch is shifted fully out.

Yet another source of problems can be that the locknut holding the spider is too long and touches the cover bushing before the belt is all the way to the top. Grinding this locknut shorter will cure the problem. Grind it too short and the belt flips out of the sheaves and turns over.

On some clutches the sheaves may touch each other on the bottom before the belt is all the way to the top. The cure for this is to machine small flats on the bottom of the sheaves to make them go together further. Again caution is required to prevent the belt from shifting too far out. To make sure everything is in order, we often assemble the two clutches in the machine without the springs. We then install the belt and move it out to the full shiftout position. This allows you to see where you may have interference problems, and to work out a solution.

A big lake is a good test area for clutch tuning. A radar gun keeps track of the improvements.

CHAPTER 10

Testing

A good testing procedure is a must if you want to get results. The winners are usually those who test the most, and keep accurate records. It is surprising how few people keep records, but the 10% that do, usually also do 90% of the winning.

The basic rules of testing are: **Follow the same procedures every time, and only change one variable at a time.**

Good instrumentation and the right tools makes testing easier and more efficient. The more test runs you can get in, the more variables you can explore for possible performance gains.

INSTRUMENTATION

An accurate tach is necessary for any clutch tuning. If you don't have a good tach, you might as well be guessing. The problem with most tachs is that they are calibrated to read correctly at mid scale. This means that they are correct at 5000 RPM, but may be 250 RPM off at 9000. We prefer a Kroeber tach manufactured in Germany. This tach is made for Grand Prix motorcycle racing, and has a superior rubber mount that shields it from vibration. It also has a built in voltage regulator for protection, and a finely damped needle that stays right on the RPM at all times without lagging or "bouncing over". Do not use digital tachs when clutch tuning. They don't give you the same feel for what's happening to the clutch as watching a needle. The only drawback to this tach is the price which is close to $200.00, but you get what you pay for. A Kroeber tach will last you years without trouble, while in the meantime you went through six or eight cheap tachs, and each one read different and quit when you were in the middle of your most important test.

VDO produces this computerized tach which will replay your run in slow motion. This tach is a must for any serious racer, or anyone doing a lot of clutch tuning. It is available from AAEN Performance.

Your tach is the only communication channel you have with your engine, so make sure it's a good one.

We check tachs in several ways. One way is to use a frequency synthesized calibrator, and another is to check it on the dyno. Checking on the calibration costs $15.00 and takes little time, while running your engine on the dyno takes days and costs thousands of dollars.

Get your tach calibrated to make sure what you read on the tach is actually what the engine is doing.

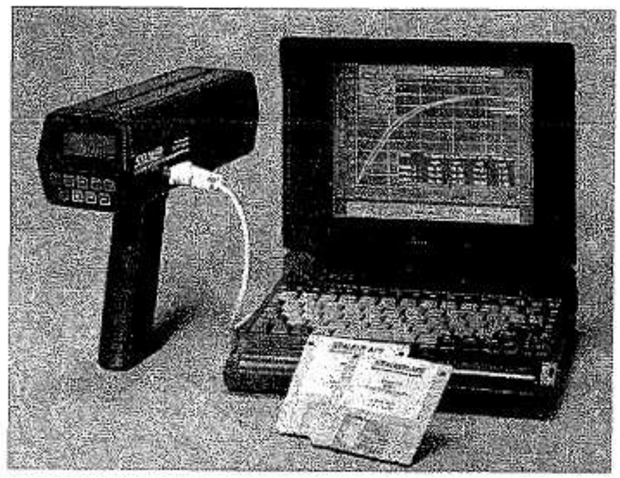
Should you ever have your race engine dynoed, make sure you get your tach checked against it at the same time. Tuning at the wrong RPM because your tach was off is a waste of time.

Reading the tach while you test can many times be hard as you have enough to concentrate on the driving. A good idea is to put a line on the dial with a piece of tape or paint where you want the clutch to shift. You may also want to twist the tach so this line is straight up and down and easy to see.

Some machines have their tach too far down on the console. I recommend that you make a special bracket for it on the handle bar right in your line of sight. This gives you a better chance to concentrate on the driving.

You can now easily observe if the needle is right on your line, or how much low or high, without having to read any numbers.

For a long time this has just been a racers dream, but now it has become an affordable reality. VDO produces a computerized tach with a large dial and sweep needle that can play back your run right on the tach. The sample rate can be set for 50 or 200 seconds, and the tach plays back in 1/3rd speed, which gives you a good view of what's happening to the clutches. We have used this unit for several years now, and have grown almost dependent on it. Testing is now safer and the data much more accurate.



The STATS Testing System for the Stalker Radar System allows you to record the run and have an acceleration curve displayed on a computer. Runs can be saved and compared for analysis.

A good speedometer is handy if you have decent traction, but it is hard to read down to the MPH. If the track is slipping, and it usually is, then a radar gun is useful. There are some good radar guns available now for under 500 dollars, and sometimes you can pick up a rebuilt police radar gun for a reasonable amount.

We also use a set of Crondex lights to measure speed and acceleration. For drag racing we set the lights 250 feet apart and work on dialing in hole shots and shift out problems. A Crondex unit reads out to one ten thousandths of a second, and is therefore preferable to a radar gun which only gives you the closest full MPH.

All this fancy equipment would be nice to have, but in most cases you only have a tach and speedometer, and maybe a radar gun.

Side by side testing is very useful. If you have a second sled which is comparable in speed you can use it as a baseline. Never change anything on this machine as long as you are testing with the other, and always use the same drivers. In side by side testing you can get a good idea about improvements in both acceleration and top end, something that is harder to do when you are just testing one machine by itself. We use a lot of side by side testing, because that is how our products eventually will be tested by the consumer.

TEST AREA

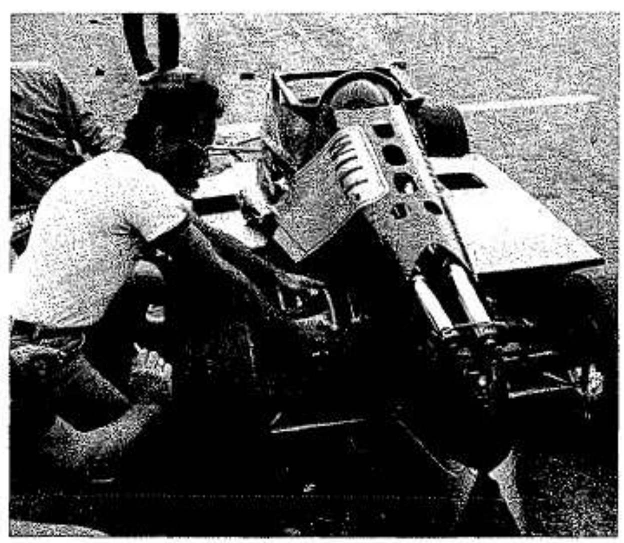
You need a long flat surface where you have plenty of room for at least a quarter mile run plus another quarter mile for shutdown. This pretty much limits you to a lake, an airport, or a drag strip for testing. Before you make any test runs, run over your track at a safe pace to make sure there are no surprises. You may have tested in the same place yesterday and an ice heave came up overnight. You may have tested before lunch, and while you were eating someone drove across your track with a 4 x 4 truck and left big tire ruts. Always use full safety equipment such as helmets and Safe Jacs, and always have someone with you.

Make at least four runs on each combination, to make sure the engine is warm and everything is consistent. A straight run is good to dial in your basic shift curve, but there can be some pitfalls. On a straight line run you will get better acceleration and top end with a loose driven spring pretension. This setting may be useless for trail riding, drag racing or Sno Cross because of poor backshifting. First establish what kind of tension you need for good back shifting with your driven unit, then proceed with your testing. The procedure is a little different for drag racing and speed runs where you want to find out how low of a pretension you can get away with without the belt slipping.

When testing, we recommend the following procedure.

1. Start by filling your record sheet with date, machine type, surface condition, test track info and weather information.
2. Establish the shift speed and engagement speed you want to accomplish, and draw a speed diagram for reference.
3. Establish the driven pretension and cam you feel will give adequate back shifting.
4. Make sure the transmission is in good mechanical shape, correctly aligned and that you have a good belt.
5. Write down belt length and center distance, and mark the belt for rotation.
6. Write down flyweight curvature, weight, spring color and number of spider shims used.
7. Make some runs to check out the track and machine and to warm up the engine.
8. Make four full acceleration runs from a standing start, and observe the shape of your shift curve. Compare result with the curve you want to obtain and make the following changes one at a time.
9. If shift speed is too high go to heavier weight. If shift speed is too low go to lighter weight.
10. If shift speed increases while shifting out, go to spring with less rate. If shift speed decreases while shifting out, go to spring with more rate.
11. If engagement speed is too low go to spring with same rate but higher pretension. If engagement speed is too high go to spring with same rate but lower pretension. The new pretension will also affect shift speed. Retest with heavier or lighter flyweights.
12. If engagement speed still is too low and no better spring can be found, grind flats or notches in weight to obtain higher engagement.
13. When you have reached the theoretically ideal shift curve, try heavier and lighter weights to see if your power peak may be located slightly differently, depending on altitude and temperature.
14. Experiment with different driven pretension and cams to achieve better efficiency.

Due to the torque feedback, less power reduces shift speed, and you may have to compensate with lighter weights. The following variables reduce power: Temperature, barometric pressure, humidity and altitude influence jetting and air density. Jetting and air density influence the power output of the engine, which in turn influences the shift curve of the transmission. Higher temperature reduces power. Higher altitude and lower barometric pressure reduces power. Low humidity reduces power. Jetting that is too rich reduces power.



Have several clutches available and be ready to make quick changes during pit stops or between races.

RACING

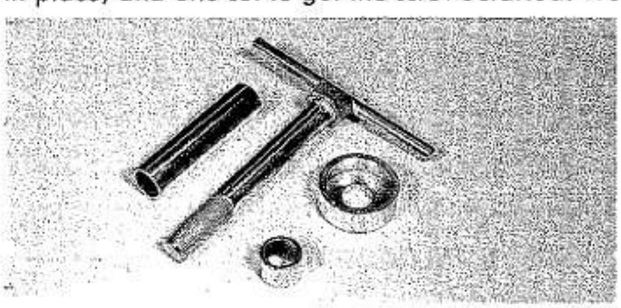
The key to successful racing is to have tested and be prepared when you get to the track. Even the best production can not forsee the variables of the day, but you can prepare to do something about it quickly. Extra clutches and belts, a complete assortment of flyweights, springs and ramps, and good tools to change components quickly should be part of the battle plan.

We usually carry 3 drivers and 3 drivens and 3 broken-in belts with us. One set of clutches is installed on the engine. The two front clutches are set up with lighter and heavier weights than the regular one on the engine. One driven clutch is set up with a different pretension and the same ramp, while the other is set up with the same pretension and different ramp. Air tools, electric impact wrenches and speed wrenches are ready in the right sizes for quick use. Practice sessions should be organized with timing equipment and a clear picture of what variables to try. The driver must get a tach reading at the critical parts of the track, and convey this information to the clutch man so he can make the necessary changes quickly. Make notes of the clutch behavior, race track conditions and lap times. A cold day with hard and fast ice requires heavier flyweights than a warm day with soft ice and slop on top creating higher resistance. In soft sloppy conditions you may also want to run more pretension and a smaller cam angle to keep the clutch back shifting properly.

Experience in reading the weather and track conditions only comes with lots of practice. Some people think that extra clutches are an unnecessary cost, but when you think about how much money and time you spent getting to the race track, the least you can do is to make sure you have the right parts there to make the job most efficient and get you the best results.

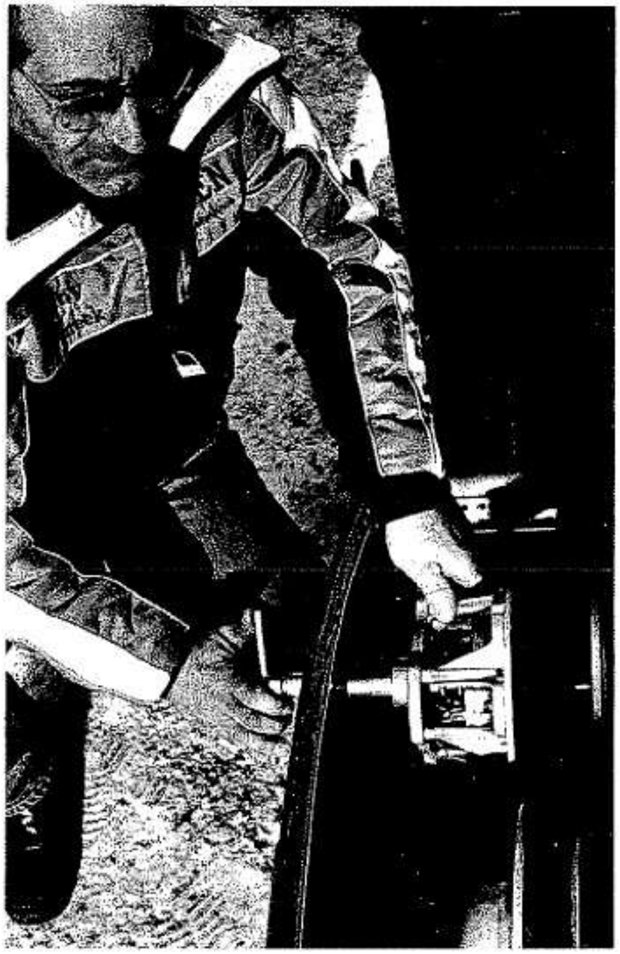
TOOLS

Since you intend to work on your clutches, you should get the correct tools to make the job fast and easy. There are some special tools that we feel are very useful. If you leave the clutch on the machine while changing weights or springs, you know how hard it is to get the cover back on with a new spring. It takes two sets of hands, one set to press the cover in place, and one set to get the screws started. We



This simple clutch service tool saves a lot of work when installing springs and flyweights in the machine.

have developed a Clutch Service Tool which takes care of that problem. It is basically a big bolt with a T-handle, a cup and a number of spacers. The bolt uses the threads for the clutch puller and, together with the right spacer, you can easily screw the cover back in place and start your cover bolts. What used

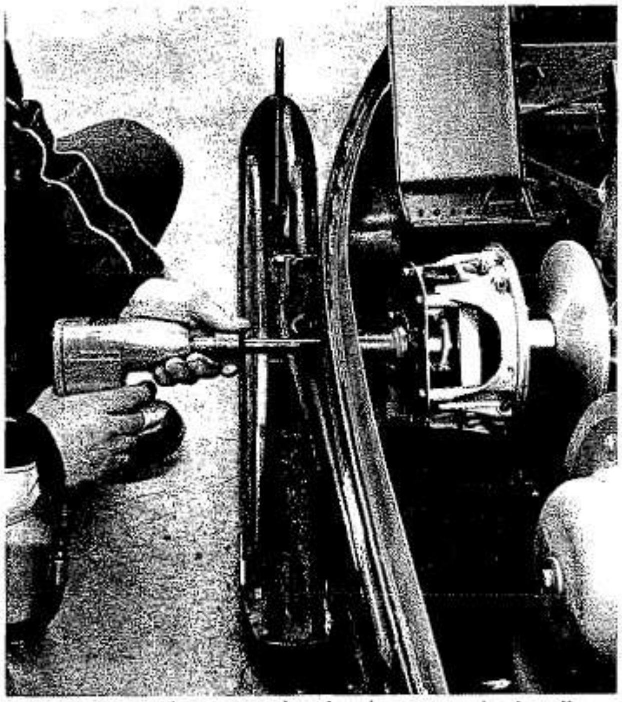


The clutch service tool quickly screws into the clutch to compress the cover or move the flyweights away from the rollers.

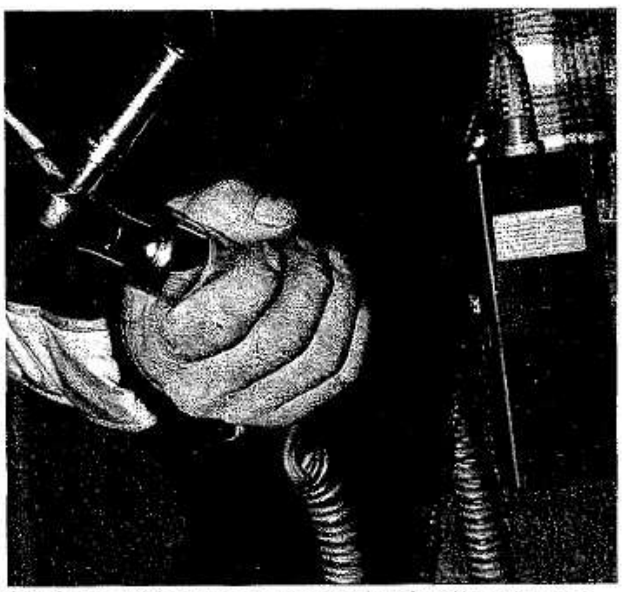
to be a 4 hand job now can be done with one hand. By adding a second spacer the whole movable sheave assembly can be pushed in to move the flyweights away from the rollers and make them easy to change.

If you want to remove the clutch from the crankshaft, you need a good clutch puller. We recommend using an impact wrench on the puller, either electric or air driven. Using normal wrenches may not be enough, and sometimes you need to give the puller a rap with a hammer to "shock" the clutch loose. When you do decide to shock it with a hammer, make sure that you hit it square on so that you do not hurt the crank shaft. Our favorite tool is 1/2" drive Ingersoll Electric Impact Wrench, since electricity is usually easier to come by than air.

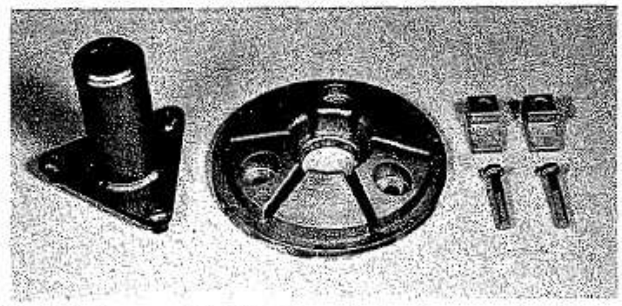
If you want to remove the spider to service the rollers or change the spider spacing, you need special tools. On the Comet and Polaris the spider is screwed on to the shaft, sometimes with plenty of locktite. This means that you need a fixture to hold the stationary sheave while you try to unscrew the spider. Polaris has a die cast fixture you can screw to your work bench or put in a vise. The ribs on the stationary sheave fit into the fixture and are held in



An impact wrench comes in handy when using clutch pullers to remove the clutch.

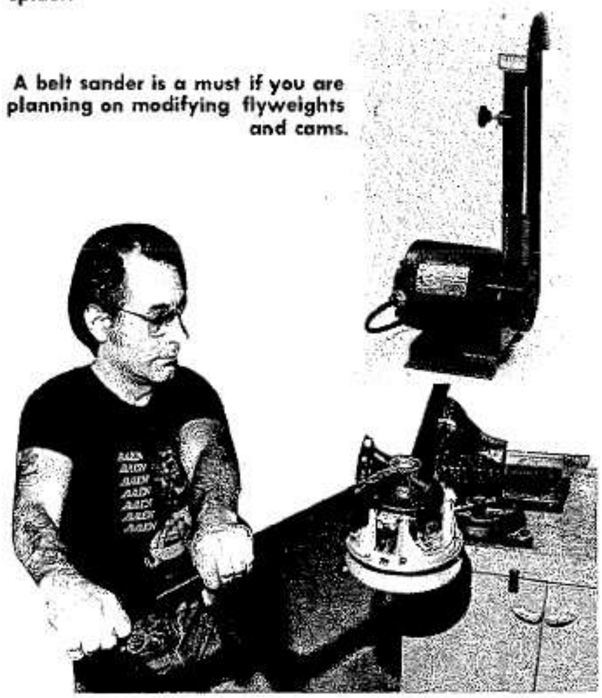


This electric ratchet wrench comes in handy when you are in a hurry to disassemble clutches.

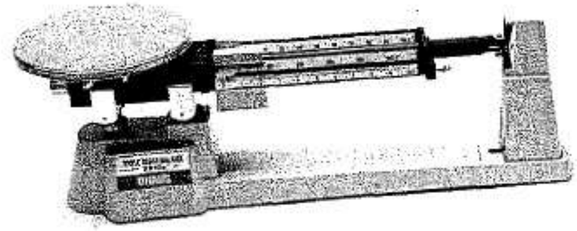


The Polaris assembly fixture saves time when removing the spider.

A belt sander is a must if you are planning on modifying flyweights and cams.



A grunt tool is needed to loosen the spider on a Comet clutch.

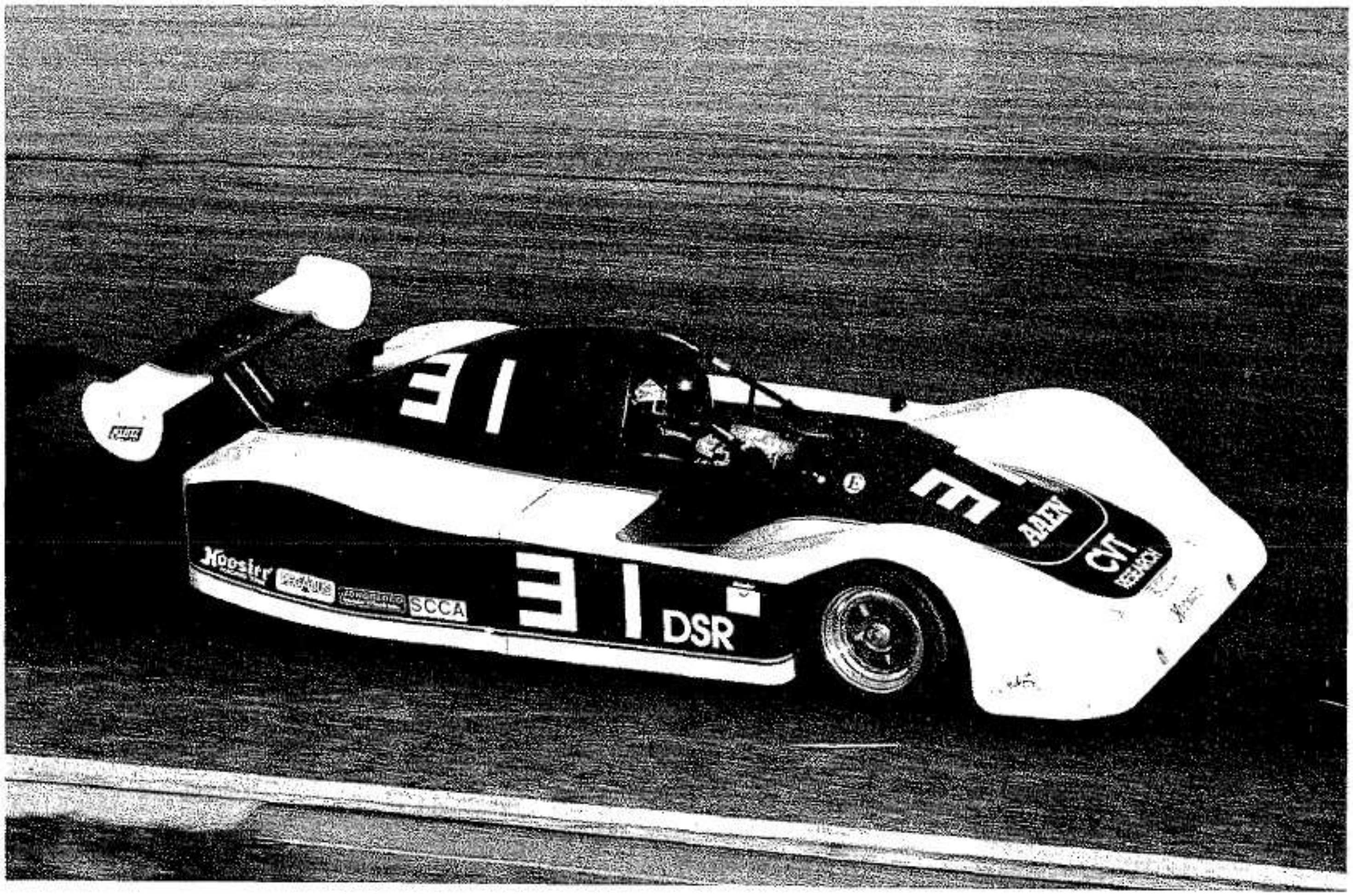


A good gram scale is another necessity when modifying flyweights.

place by the fixture. Then with a three pronged tool, designed to fit the spider (also supplied by Polaris), and a breaker bar or an impact wrench you can loosen the spider. On the Polaris there is also a jam nut that must be loosened before the spider can be removed. On the Comet there is no jam nut, just stubborn locktite. Comet has what they call a "grunt tool" to remove the spider. This is basically two large bars, one attached to the spider and one to the shaft. After you use it you will understand why it is called a "grunt tool". It takes some muscle to loosen the spider. We also use a Polaris type plate to hold the Comet stationary sheave. This works well while tightening the spider down. A Polaris spider wrench can also be used on a Comet with small modifications. We mentioned the use of a gram scale, belt sander and sanding block in our chapter on weight modification. These tools are desirable if you want to get accurate results fast.

You don't need much for your driven clutch, but a good set of snap ring pliers with a 45 degree bent nose makes it easier to get the snap ring out.

To keep the important tools at hand and your springs and weights organized, spend a few dollars on a plastic fishing tackle box. The compartments are ideal for storing clutch parts in an orderly fashion. Stay neat and organized, and clutching will be fun!



CVT in Race Cars

Our D-Sports racer is now transferring 200 HP through a continuously variable belt drive transmission in 40 minute races. The car has recorded many national wins, including the prestigious June Sprints at Road America in 1996.

CHAPTER 11

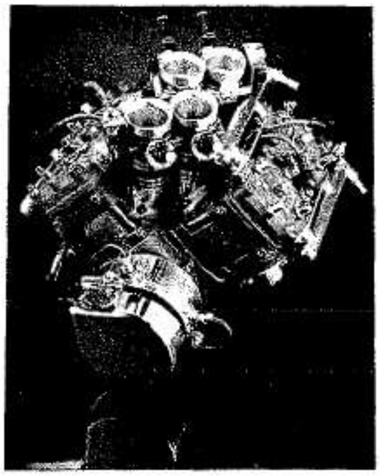
Looking Into The Future

Trying to predict the future always holds the risk of being proven wrong later. What I would like to do is look at the possibilities that the "snowmobile" transmission" holds. The rest of the automotive world has not paid much attention to this transmission system. They somehow seem to assume that a rubber belt is too unreliable although they use much thinner versions to power all kinds of engine accessories.

The simplicity of operation and the high efficiency system should make it very desirable in many forms of automotive applications. Snowmobiles regularly transmit 50 to 100 HP for a whole season on the same belt, and great advances have been made in belt technology. The limit to horsepower transfer lays in the belt design, and I think that even greater strides lay ahead in the near future.

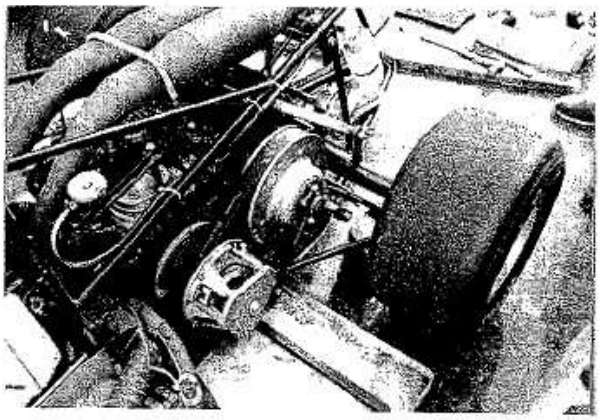
In the last 12 years we have steadily developed the continuously variable belt drive transmission in our SCCA D-Sports racer. 12 years ago we could only transfer 125 HP reliably for a 40 minute race. Today we are transferring 185 HP reliably, and 200 HP should shortly be within reach. We have achieved this result by paying attention to belt design, efficiency and cooling of the drive system.

The now popular F-500 class in SCCA Road



This 1000 cc V-4 engine produces 275 HP and transfers the power successfully through a single 1.5" wide "Top Cog"™ belt.

This Polaris 6 wheel ATV completed the grueling 3500 mile Paris to Dakar race with no problems to its automatic belt drive transmission.



This F-440 transmission has an extra brace on the outside of the driven clutch to keep it from twisting under load.

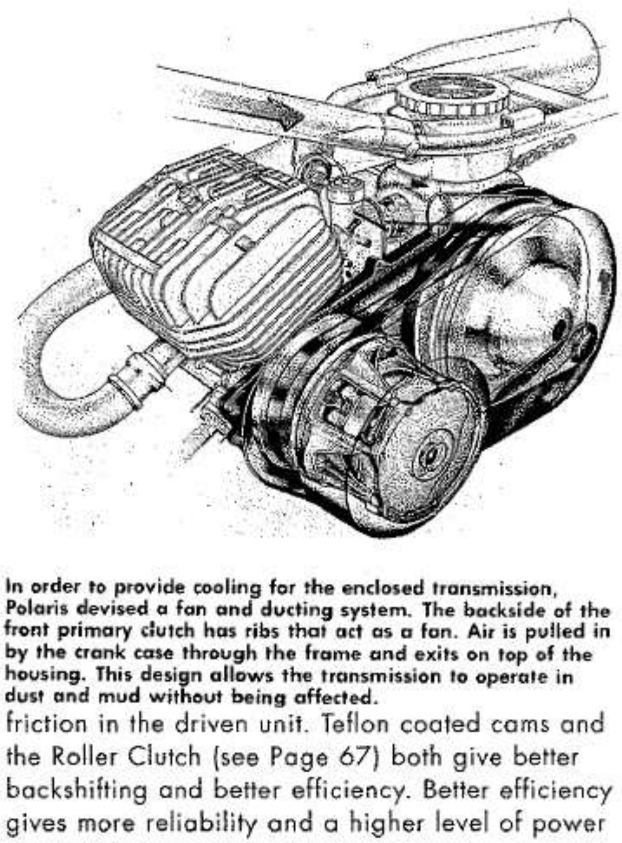
Racing utilizes a snowmobile engine and drive system. With no need to shift gears, these races are becoming popular as the driver can fully concentrate on the driving.

Polaris's new line of ATV vehicles use an advanced transmission design. This transmission is totally enclosed (eliminating water, dirt and mud) and has a forced air circulation system to keep it cool. With 30 HP from a single cylinder air cooled engine, this ATV can accelerate with machines of twice the power. It also rated favorably in hill climbing and pulling through heavy mud where a missed shift could get you stuck. At the 30 HP level belts last forever.

In the 1993 Paris to Dakar Rally, a Polaris ATV became the first ATV to cover the 3500 mile long run. All earlier attempts with conventional transmissions had failed.

As we go up in horsepower we may see a separate slipping clutch in the system, with a belt that is always in engagement. This would make it possible to eliminate belt wear due to high starting torque and belt slip.

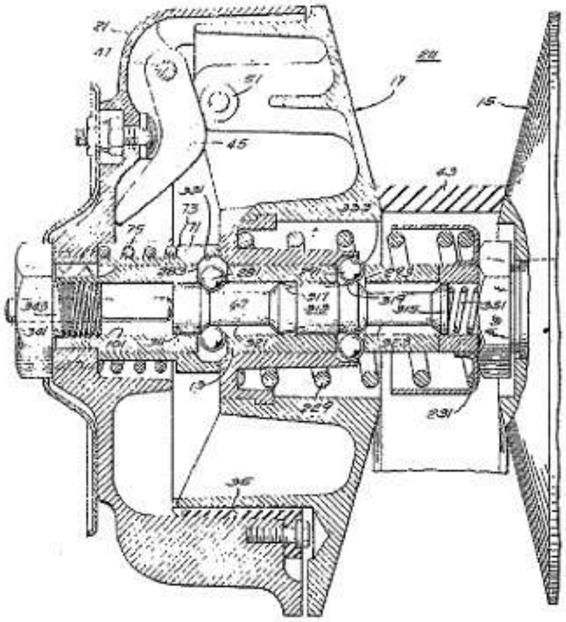
We have also made great strides in reducing



In order to provide cooling for the enclosed transmission, Polaris devised a fan and ducting system. The backside of the front primary clutch has ribs that act as a fan. Air is pulled in by the crank case through the frame and exits on top of the housing. This design allows the transmission to operate in dust and mud without being affected.

friction in the driven unit. Teflon coated cams and the Roller Clutch (see Page 67) both give better backshifting and better efficiency. Better efficiency gives more reliability and a higher level of power transmission.

There is also much work that could be done to make clutches more adjustable and easier to tune for snowmobiles. SkiDoo has given us a hint of this direction with their new clutch. This clutch has an eccentric bolt backing up the shift cam. The cam angle can be adjusted by rotating the bolt from the outside. OMC was a pioneer in clutch design with the first torque ramp, the first wide belt clutch, the first wide ratio over drive unit with a 5:1 total ratio, and the first with a neutral lockout. While I worked at OMC, we developed, in the Advanced Design



Department, a two step transmission which was patented. This transmission had two shift curves. One curve came in with low engagement speed and shifted out smoothly. By pushing a lever you could select a high engagement and more shift speed. This made it ideal for the person who liked to trail ride but every now and then also would participate in a friendly drag race or just needed more power up a hill. This feature was especially well liked for mountain use. This transmission never got out of the prototype stage, as the snowmobile market took a dive and OMC decided to pull its Evinrude and Johnson brands out.

John Deere's TR-800 experienced a smiliar fate after a short selected run, when Deere withdrew from the market. There are lots of good ideas yet to be developed with the variable speed belt transmission. This transmission has all of the good characteristics that a transmission should have and its application for automotive use has only just begun. You are looking at the transmission of the future.

Fig. 1 - Selectomatic Transmission; Inventors: Olav Aaen, Edgar Rose, Anthony Prince, Patent no. 3939720 by OMC. This design allowed the driver to select two different shift curves to suit his driving style. Two springs worked against each other, and by locking one in place a higher engagement and shift speed would result.

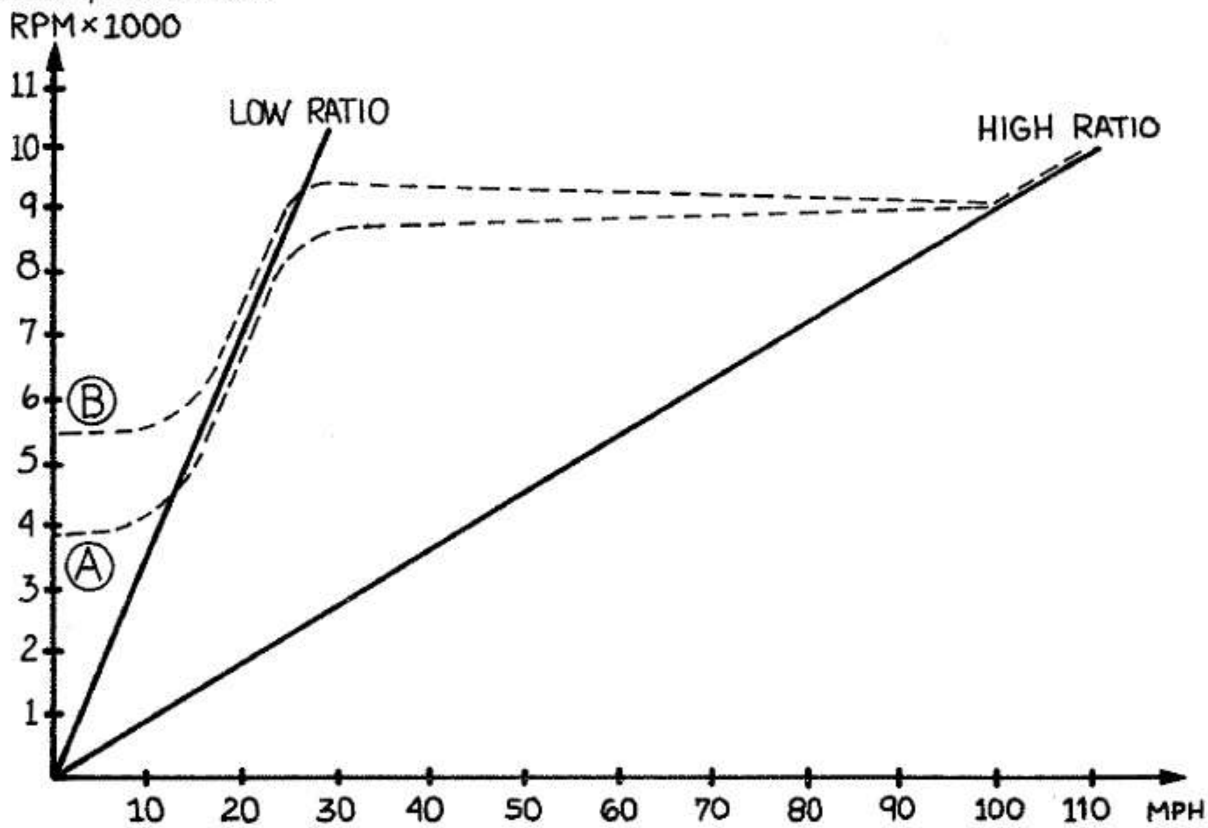
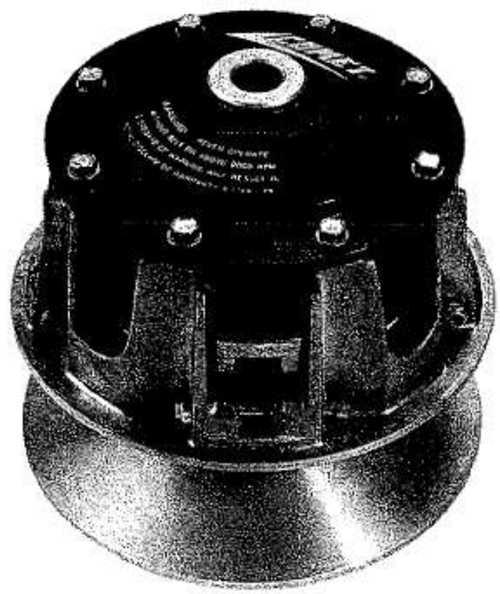


Fig. 2 - The selectomatic shift diagram. Line A shows the shift curve with both springs working. With the small spring working against the large spring you get a lower engagement and shift speed. With the small spring locked in place flyweights have to overcome the full force of the large spring and a higher engagement and shift speed is the result.



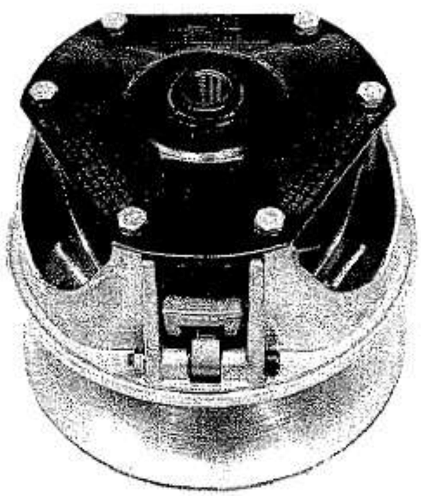
COMETS NEW 4 PRO CLUTCH

COMET

The Comet clutch system is manufactured by Hoffco Industries of Richmond, Indiana. Comet manufactures a wide variety of power transmission products for the recreational industry. Comet now markets two primary clutches for snowmobiles, the 102-108 series and the new Quad clutch. The 102-108 series is a conventional design with three flyweights working against the rollers mounted in a spider. Torque transfer buttons are placed on the outside of the roller pins, as now used in Polaris, Yamaha and Arctic designs. This clutch is found extensively as original equipment on Arctic, Yamaha and Ski Doo sleds in the last 20 years. The 102-108 range of triple flyweight clutches are available as a replacement clutch for most snowmobiles and a large selection of weights and springs make it possible to tune it for any power curve. Comet has now also introduced a new 4 Pro clutch. This clutch is similar in design to the 102-108 series, but it has 4 flyweights instead of 3, and therefore 4 rollers and 8 torque transfer buttons. This clutch was developed to meet the increasing horsepower demand from the larger 800cc-1000cc snowmobile engines which now push over 200 HP. The Comet has been very popular as a racing clutch because of the availability of tuning parts. Polaris and Arctic flyweights can be used in Comet clutches which adds to its versatility. Polaris springs can also be used if a small amount is machined away from the spider and outside slide bushing. Polaris springs have a slightly smaller inside diameter but are very close on pretension and rate.

Care should be taken when the flyweights are replaced. Comet now uses a special bolt for flyweight pivot. This new bolt has a shoulder to stop the nut and prevent over tightening. Use plenty of Loctite and a new nut every time on this bolt. Comet was the first to use an open cover design for their primary unit. This increases cooling of the components and keeps the belt temperature down.

Comet offers three basic cam shapes on the flyweight; a mild, an aggressive, and a Polaris racing style "A" series. Comet starts with a number of base weights such as E, C, B, F, U, N, W, K,and A-1. The flyweight is classified by group name, weight, and engagement profile. Springs are listed by rate, pretension, color code, spring wire thickness and free length. Some of this information may not be the same as other publications. Our information is based on our own measurements of the manufactured part. Comet supplies drivers for a number of belt widths and sheave angles, including 1 1/4, 1 5/16, 1 3/8, and 1 1/2 belts. Make sure you get the right combination for your belt.



COMET 103 HPQ CLUTCH

Comet has introduced the 103 HPQ (High Performance Quad) primary clutch for use on the Polaris ATV platform. This three flyweight unit was developed for racing and calibrates differently than the Polaris ATV primary. The Polaris ATV primary clutch has a different calibration than the snowmobile units, in order to produce a lower engagement speed for slow riding. This is done by spacing out the spider and using more "heel" on the weights to swing them farther out at engagement. The Comet clutch has a more traditional spacing of the spider, making it possible to obtain higher engagement speed and more aggressive shift out during acceleration. As a result, much lighter flyweights are used in the Comet. While the Comet spider can be spaced out and made to perform like a Polaris with lower engagement, it is difficult to convert the Polaris into an aggressive racing clutch, because the spider cannot be moved in without re-machining. As a result, many racers are now finding advantages to using the Comet 103 HPQ for competition.

COMET CAM ARMS

A-1 Cam Arm Part No. 207689A1 48.5 gr. ± .75	Orange A-1 Mod Cam Arm Part No. 209940A1 47.3 gr. ± .75	Hub Yellow A-2 Cam Arm Part No. 208221A1 48.0 gr. ± .75	Red A-3 Cam Arm Part No. 208268A1 44.9 gr. ± .75	Gold A-4 Cam Arm Part No. 212135A1 48.7 gr. ± .75	Blue A-5 Cam Arm Part No. 212639A1 46.8 gr. ± .75	A-6 Cam Arm Part No. 212821A1 43.8 gr. ± .75	End Yellow A-7 Cam Arm Part No. 212822A1 42.3 gr. ± .75
End Red A-8 Cam Arm Part No. 212820A1 44.3 gr. ± .75	End Black A-9 Cam Arm Part No. 213370A1 44.8 gr. ± .75	End Brown A-10 Cam Arm Part No. 213371A1 41.3 gr. ± .75	Purple A-11 Cam Arm Part No. 213400A1 49.0 gr. ± .75	A-12 Cam Arm Part No. 213864A1 45.3 gr. ± .75	A-13 Cam Arm Part No. 213867A1 43.3 Gr. ± .75	A-14 Cam Arm Part No. 214172A1 40.3 gr. ± .75	A-15 Cam Arm Part No. 214255A1 50.0 gr. ± .75
A-16 Cam Arm Part No. 214898A1 50.0 gr. ± .75	A-17 Cam Arm Part No. 215149A1 46.5 gr. ± .75	A-18 Cam Arm Part No. 215400A1 52.7 gr. ± .75	A-19 Cam Arm Part No. 215407A1 46.2 gr. ± .75	A-20 Cam Arm Part No. 251575A1 49.2 gr. ± .75	A-21 Cam Arm Part No. 215865A1 45.5 gr. ± .75	A-22 Cam Arm Part No. 215867A1 46.9 gr. ± .75	A-23 Cam Arm Part No. 216061A1 53.5 gr. ± .75
A-24 Cam Arm Part No. 216065A1 44.0 gr. ± .75	A-25 Cam Arm Part No. 216087A1 49.5 gr. ± .75	A-26 Cam Arm Part No. 216089A1 51.5 gr. ± .75	A-27 Cam Arm Part No. 216091A1 39.7 gr. ± .75	A-28 Cam Arm Part No. 216381A1 46.5 gr. ± .75	A-35 Cam Arm Part No. 217282A1 35.3 gr. ± .75	A-37 Cam Arm Part No. 217285A1 37.3 gr. ± .75	A-39 Cam Arm Part No. 217288A1 39.3 gr. ± .75
A-41 Cam Arm Part No. 217291A1 41.3 gr. ± .75	A-52 Cam Arm Part No. 214897A1 52.5 gr. ± .75	A-54 Cam Arm Part No. 214651A1 54.3 gr. ± .75	Green AA-1 Cam Arm Part No. 208394A1 44.3 gr. ± .75	AA-2 Cam Arm Part No. 208395A1 43.3 gr. ± .75	AB-1 Cam Arm Part No. 208455A1 40.8 gr. ± .75	Silver AB-2 Cam Arm Part No. 208457A1 47.7 gr. ± .75	AC Cam Arm Part No. 215218A1 57.3 gr. ± .75
AC-1 Cam Arm Part No. 215242A1 52.3 gr. ± .75	AC-3 Cam Arm Part No. 215922A1 54.3 gr. ± .75	B-1 Cam Arm Part No. 207801A1 46.0 gr. ± .75	B-2 Cam Arm Part No. 208232A1 41.3 gr. ± .75	B-3 Cam Arm Part No. 209431A1 44.7 gr. ± .75	B-1 Cam Arm Part No. 211331A1 42.7 gr. ± .75	C1-M Cam Arm Part No. 208967A1 38.0 gr. ± .75	C-2 Cam Arm Part No. 208968A1 43.4 gr. ± .75
C-2M Cam Arm Part No. 209403A1 37.6 gr. ± .75	D Cam Arm Part No. 203088A1 42.6 gr. ± .75	E Cam Arm Part No. 203089A1 40.0 gr. ± .75	F-2 Cam Arm Part No. 211333A1 54.0 gr. ± .75	HE-1 Cam Arm Part No. 205353A1 51.7 gr. ± .75	HE-2 Cam Arm Part No. 205357A1 54.1 gr. ± .75	HE-3 Cam Arm Part No. 205385A1 45.7 gr. ± .75	HE-4 Cam Arm Part No. 206995A1 55.9 gr. ± .75
HE-5 Cam Arm Part No. 207820A1 45.7 gr. ± .75	K-1 Cam Arm Part No. 209011A1 59.7 gr. ± .75	K-2 Cam Arm Part No. 213337A1 54.0 gr. ± .75	L Cam Arm Part No. 203837A1 38.6 gr. ± .75	N-1 Cam Arm Part No. 209642A1 45.5 gr. ± .75	P-1 Cam Arm Part No. 209782A1 41.8 gr. ± .75	Q Cam Arm Part No. 206911A1 56.9 gr. ± .75	R Cam Arm Part No. 205249A1 59.6 gr. ± .75
R-1 Cam Arm Part No. 207674A1 57.7 gr. ± .75	T Cam Arm Part No. 205137A1 48.1 ± .75	U Cam Arm Part No. 205169A1 63.0 gr. ± .75	U-1 Cam Arm Part No. 207753A1 61.5 gr. ± .75	W Cam Arm Part No. 205257A1 59.5 gr. ± .75	X Cam Arm Part No. 207270A1 56.7 gr. ± .75	X-1 Cam Arm Part No. 207331A1 49.7 gr. ± .75	

COMET WEIGHTS

Cam Arm	Weight (gr) ±1	Engagement Profile	Part No.
A-1*	48.5	Standard	207689A1
A-1 mod*	47.3	Standard	209940A1
A-2*	48.0	Standard	208221A1
A-3*	44.9	Standard	208268A1
A-4*	48.7	Standard	212135A1
A-5*	46.8	Standard	212639A1
A-6*	43.8	Standard	212821A1
A-7*	42.3	Standard	212822A1
A-8*	44.3	Standard	212820A1
A-9*	44.8	Standard	213370A1
A-10*	41.3	Standard	213371A1
A-11*	49.0	Standard	213400A1
A-12*	45.3	Standard	213864A1
A-13*	43.3	Standard	213867A1
A-14*	40.3	Standard	214172A1
A-15*	50.0	Standard	214255A1
A-16*	50.0	Standard	214898A1
A-17*	46.5	Standard	215148A1
A-18*	52.7	Standard	215400A1
A-19*	46.2	Standard	215407A1
A-20*	49.2	Standard	215575A1
A-21*	45.5	Notch	215865A1
A-22*	46.9	Notch	215867A1
A-23*	53.5	Notch	216061A1
A-24*	44.0	Notch	216065A1
A-25	49.5	Notch	216087A1
A-26*	51.5	Notch	216089A1
A-27*	39.7	Notch	216091A1
A-28*	46.5	Notch	216381A1
A-35*	35.3	Standard	217282A1
A-37*	37.3	Standard	217285A1
A-39*	39.3	Standard	217288A1
A-41*	41.3	Standard	217291A1
A-52*	52.5	Standard	214897A1
A-54*	54.3	Standard	214651A1
AA-1*	44.3	Standard	208394A1
AA-2*	43.3	Standard	208395A1
AB-1*	40.8	Flat	208455A1
AB-2*	47.7	Standard	208457A1
B-1*	46.0	Standard	207801A1
B-2*	41.3	Standard	208232A1
B-3*	44.7	Flat	209431A1
B-4*	42.7	Standard	211331A1
C-1M*	38.0	Notch	208967A1
C-2*	43.7	Standard	208968A1
C-2M*	37.6	Notch	209403A1
D	42.6	Standard	203088A1
E	40.0	Standard	203089A1
F-2	54.0	Standard	211333A1
HE-1	51.7	Flat	205353A1
HE-2	54.1	Standard	205357A1
HE-3	45.7	Notch	205385A1
HE-4	55.9	Flat	206995A1
HE-5*	45.7	Flat	207820A1
K-1	59.7	Standard	209011A1
K-2	54.0	Standard	213337A1
L	38.6	Flat	203837A1
N-1*	45.5	Notch	209642A1
P-1*	41.8	Flat	209782A1
Q	56.9	Notch	206911A1
R	59.6	Standard	205249A1
R-1	57.7	Standard	207674A1
T	48.1	Notch	205137A1
U	63.0	Standard	205169A1
U-1	61.5	Standard	207753A1
W	59.5	Standard	205257A1
X	56.7	Notch	207270A1
X-1	49.7	Flat	207331A1

*Available from Comet only, with performance bushings, factory pre-installed. For replacement bushings order kit # 208342A

COMET SPRING CHART

Color	Part No.	O.D.	Height	Spring Rate	Engine Preload	Full Shift
Yellow /Red	208238A	2 1/6"	2.29/32	146	93	275
Silver/Orange	209936A	2.06"	3.07"	142	99	283
Yellow/Green	208228A	2"	3 13/32"	130	134	325
Red	207877A	2 1/16"	3 1/4"	130	92	254
Red/Blue	209833A	2.06"	3 1/16"	129	74	236
Yellow/White	211361A	2.06"	3"	123-129	76	233
Gold	208175A	2"	3 1/2"	123	129	296
Black	204115A	2"	3 1/8"	118	50	198
Brown	205040A	2"	2 7/8"	116	50	195
White	203474A	2"	3 1/8"	100	65	187
Silver/Red	209677A	2.03"	3.11"	97	67	189
Purple	207888A	2"	3 15/32"	96	106	226
Yellow	203475A	2"	3 1/2"	96	96	224
Pink	203473A	2"	3"	92	45	142
Purple	207758A	2"	4 1/8"	92	136	254
Silver/Black	209696A	2 1/32"	3 23/64"	91	85	203
Silver	204818A	2"	3 15/32"	84	85	190
Silver/Green	209935A	2.03"	3.638"	78	99	196
Pink	202789A	2"	3 1/2"	72	45	130
Yellow	202551A	2"	4 1/16"	69	100	187
White	202467A	2"	3 3/4"	62	82	160
Blue	202552A	2"	4 7/8"	55	118	187
Orange	203472A	2"	3"	40	25	74
Blue	203071A	2"	4"	110	106	25

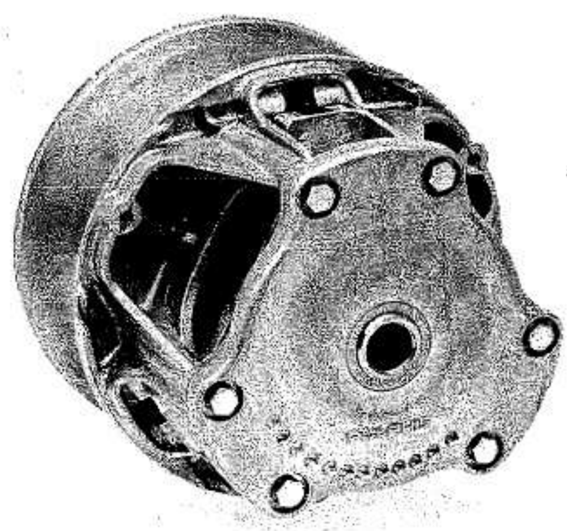
AAEN PRIMARY RACING SPRINGS

Color	Part No.	O.D.	Height	Spring Rate	Engine Preload	Full Shift
Orange	AAS11054	2.06"	3.7"	127	120	290
Purple	AAS10034	2.06"	3.9"	127	150	320
Yellow	AAS10038	2.06"	3.4"	172	150	380
Alm/Bl	AAS11039	2.06"	4.3"	92	165	290
Yel/Red	AAS10040	2.06"	4.3"	116	165	310
White	AAS10043	2.06"	4.1"	128	170	330
Maroon	AAS10041	2.06"	4.7"	108	185	320
Gold	AAS10042	2.06"	4.3"	116	90	335

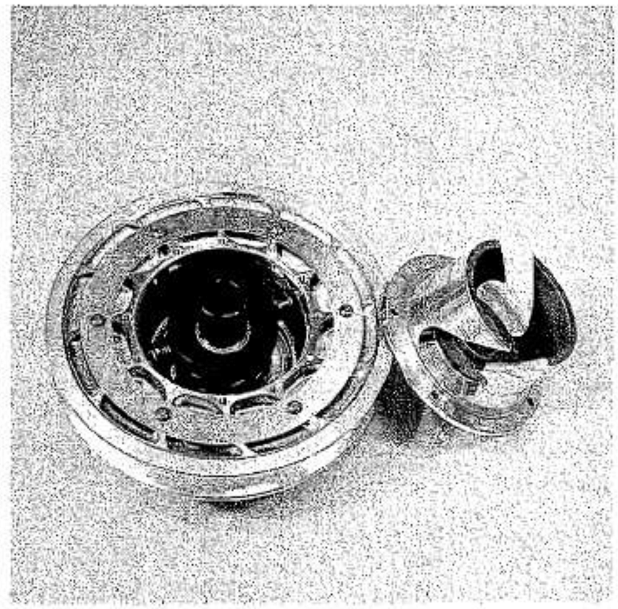
COMET 103 HPQ POLARIS ATV CLUTCH

Clutch Part No. 218875A Comet Calibration Suggestions

Year/Model	Flyweights (3)	Spring	Engagement
85-99 250 Trail Boss	A-1 207689A	Pink 203473A	2400
90-03 250 Trailblazer	A-1 207689A	Pink 203473A	
00-02 250 Explorer	A-1 207689A	Pink 203473A	
94-97 400 Sportsman 4X4	A-54 214651A	Pink 203473A	2650
98-01 400 Explorer 4X4	A-54 214651A	Pink 203473A	
95-02 400 Scrambler 4X4	A-54 214651A	Black 204115A	3400
00-01 400 Scrambler 2X4	A-54 214651A	Black 204115A	
95-02 400 Sport	A-54 214651A	Black 204115A	
500 Scrambler 2X4 / 4X4	A-1 207689A	Pink 203473A	2400



Polaris P - 85 Primary Clutch



TEAM Industries Secondary Roller Clutch

POLARIS

Polaris has been one of the leaders in snowmobile transmission design. They were one of the first to use a torque transfer button on the spider, which greatly improved backshifting compared to the splines or hex bushings used on earlier primary clutches. Polaris was also one of the first to use a wide 1.375" belt for their higher horsepower machines. The early race clutches had the spider mounted on splines, which made it possible to remove the complete movable sheave assembly to replace belt spacing washers, or quickly throw on a new assembly with different calibration at the race track. This clutch also had set screws installed behind the heel of the flyweight, to adjust the engagement speed to track conditions. By turning the set-screw in, the engagement would decrease. The belt clearance would change with the set- screw position, but since the removable assembly could be quickly shimmed up again, the racer could easily get both correct engagement speed and belt clearance. Early model primaries had a full cover, but in the early 80's this was changed to an open clutch with only a flat spring cover to improve cooling of the clutch. Present production clutches have the spider screwed on to the center shaft, and locked in place with an extra locking collar. The spider can be shimmed for correct belt clearance with special washers between the spider and the shaft. The spider buttons can also be shimmed to give a more accurate clearance with the sliding surfaces.

Polaris offers three basic flyweight curvatures, trail, aggressive trail, and racing. The trail curvatures are widely used in standard snowmobiles, while the racing curvature was developed as a part of the Sno Pro program in the late 70's. They are now also used on some of the high performance models. Weight is normally changed by varying thickness of the flyweight, although some weights are available with material ground off the tail. There are also flyweights with notches for higher engagement, and the flyweights are listed with either a standard curvature or a notch.

There is a large selection of primary pressure springs available from the factory, as well as numerous choices from the aftermarket. In addition, Comet and Arctic springs can be used. Aaen aftermarket springs are listed with with the Polaris choices. In 2002 and newer models, some larger capacity models have a new cover with .125" deeper spring pocket. Stronger springs with a longer installed length is used in these applications.

Polaris secondary clutches are known for their efficiency and tunability, and are widely used by racers. A big selection of springs and helixes are available both from the factory and aftermarket. This unit was the first to be rollerized by the aftermarket in the early 90's.

Team Industries, the Polaris transmission supplier has now developed their own secondary roller clutch. This unit has been sold as an aftermarket clutch for several years, and has now found its way into production on 2004 models. Each helix has two sets of ramps, running on only two rollers. The rollers are mounted on a floating spider, which ensures contact with the rollers at all times. On the electric reverse models, only a single set of ramps are available in the helix, in order to add a notch which prevents the clutch from shifting out in reverse (pictured above). Numerous springs and helixes are available for this unit.

Polaris Secondary Tuning Data
Polaris Secondary Cam Angles

Degrees	Description	Type	Part #
34°	34	P85-Snow	5130896
34°	34M*	P85-Snow	5130751
36°	36	P85-Snow	5130895
36°	36M*	P85-Snow	5130717
38°	38	P85-Snow	5130723
40°	40	P85-Snow	5130724
42°	42	P85-Snow	5130725
44°	44	P85-Snow	5130726
40°-36°	R9*	P85-Snow	2900043
40°-36°	40-36	P85-Snow	2900034
40°-35°	R5*	P85-Snow	2900039
40°-32°	R1*	P85-Snow	2900035
42°-36°	R10*	P85-Snow	5131296
42°-34°	R6*	P85-Snow	2900040
42°-32°	R2*	P85-Snow	5131288
42°-36°-34°	T-1*	P85-Snow	5131013
45°-36°	R11*	P85-Snow	5131297
45°-34°	R7*	P85-Snow	2900041
45°-32°	R3*	P85-Snow	2900037
50°-36°	R12*	P85-Snow	5131298
50°-36°	R49*	P85-Snow	5133023
50°-34°	R32*	P85-Snow	2900046
50°-34°	R8*	P85-Snow	5131294
50°-32°	R4*	P85-Snow	5131290
40°-38°-36°	40-38-36	P90-Sno/AT	5131161
38°-36°	38-36	P90-Sno/AT	5131162
38°-36°-34°	38-36-34	P90-Sno/AT	5131163
34°	34	P90-Sno/AT	5131164

*Cam has a .060" deeper cutaway in the snap ring pocket. For use with the 1 7/16" belt, install two additional (.030") washers.

Aaen Teflon-Coated Cams for Polaris Secondary

Degrees	Description Angle Cut	Type	Aaen Part #
36°	Straight	P85-Snow	PCP12401
38°-34°	Progressive*	P85-Snow	PCP12406
42°-34°	Progressive	P85-Snow	PCP12407
46°-38°	Progressive	P85-Snow	PCP12409
50°-36°	Progressive	P85-Snow	PCP12417

*Progressive CV cams change angles in a continuous curve.

Driven Spring Data

Torsional	Part No.	Torsional Rate Lbs./ Rev	Compression Rate Lbs./In.
Aaen Brown	AAS12416	428	65.0**
Aaen Green	AAS12412	421	63.0
Black/Yellow	7042181	380	56.0
Black/Green*	7042066	380	56.0
Blue/Orange	7042022	322	30.2
Blue	7041296	312	31.0
Aaen Blue	AAS12410	308	45.0
Aaen White	AAS12417	306	47.0***
Silver Blue	7041646	286	27.6
Silver	7041499	286	26.7
Gold	7041501	247	22.2
Black	7041782	236	22.2
Red	7041198	213	21.3

* Notes Electric Reverse Spring Only
** 1" Taller than Aaen Green Spring for more side pressure
*** 1" Taller than Aaen Blue Spring for more side pressure

Polaris Flyweights
Unbushed Early Models (1970-1995)

Weight ± 1 Gr	I.D Letter	Shift Profile	Engagement Profile	Part No.
53	Bold O	Trail	Standard	5630174
51	o	Trail	Standard	5610088
48	A	Trail	Standard	5630080
47.5	08	Trail	Standard	5630245
47	A/P	Trail	Standard	5630094
43	B	Trail	Standard	5630084
41	G	Trail	Standard	5630063
57.5	04	Race	Notch	5630229
55.5	15	Race	Standard	5630274
55	10A	Race	Standard	5630286
54	N	Race	Standard	5630080
53.5	05	Race	Standard	5630234
52	07	Race	Notch	5630244
50	06	Race	Standard	5630243
49	02	Race	Standard	5630225
46	M1 Mod	Race	Standard	5630301
44	J-1	Race	Standard	5630065
42	P-1	Race	Standard	5630089
39	K-1	Race	Standard	5630144
37.5	W	Race	Standard	5630109
34	U	Race	Standard	5630107
32.5	03	Notch	Standard	5630227
10 Series with Fiber Bushings				
74	10-74 B	Ag. Trail	Standard	1322429
72	10-72 B	Ag. Trail	Standard	1322428
70	10-70 B	Ag. Trail	Standard	1322414
68	10-68 B	Ag. Trail	Standard	1322427
66	10-66 B	Ag. Trail	Standard	1321584
64	10-64 B	Ag. Trail	Standard	1321585
62	10-62 B	Ag. Trail	Standard	1321586
60	10-60 B	Ag. Trail	Standard	1321587
58	10-58 B	Ag. Trail	Standard	1321588
56	10-66 B	Ag. Trail	Standard	1321684
55	10A	Race	Standard	1321589
54	10-54 B	Ag. Trail	Standard	1321685
53	10-AL	Race	Standard	1321531
51.5	10	Race	Standard	1321526
49.5	10M	Race	Standard	1321528
47.5	10 MB	Race	Standard	1321529
46	10 MW	Race	Standard	1321527
44	10 MR	Race	Standard	1321530
11 Series with Fiber Bushings				
57	S5711	Ex. Ag.	Standard	1322005
55	S5511	Ex. Ag.	Standard	1322004
53	S5311	Ex. Ag	Standard	1321759
51	S5111	Ex. Ag.	Standard	1321731
49	S4911	Ex. Ag.	Standard	1321730
47	S4711	Ex. Ag.	Standard	1321851
45	S4511	Ex. Ag.	Standard	1321850
43	S4311	Ex. Ag.	Standard	1321849

Weights out of production but available from dealer inventories

X - 40 gr.	01 Notched - 48.5 gr.	L1 - 34 gr.
N1 mod - 51.5 gr.	M1 - 47 gr.	L1 Mod - 33 gr.
09 - 51.5 gr.	Q1 - 38 gr.	T1 - 30 gr.
H - 51 gr.	Y - 35 gr.	Z - 29 gr.

Polaris Primary Spring Chart

Color	Part #	Engagement Load - Lbs.	Full Shift Load Lbs. - P85	Full Shift Load Lbs. - P90	Spring Rate Lbs. / In.	Free Length / In.
Almond/Black	7041816	165	310	273	108	3.75
Almond/ Red	7041988	165	310	273	108	4.29
Aaen Yellow	AAS10038	150	380	322	172	3.40
Almond	7041566	150	320	286	136	3.65
Aaen Purple	AAS10034	150	320	286	136	3.90
Almond/Blue	7041922	150	310	270	120	3.75
Almond Gold	7041645	150	270	246	96	4.00
Black/White	7041818	140	320	275	135	3.52
Black/Green	7042083	120	340	275	175	3.38
Dark Blu/Wht	7041781	120	310	262	142	3.52
Dark Blue	7041526	120	300	264	144	3.52
Aaen Orange	AAS11054	120	290	256	136	3.70
Blue/Gold	7041080	120	293	258	138	3.50
Red	7041083	120	235	212	92	3.77
Pink	7041065	115	192	177	62	4.69
Gold	7041148	100	267	234	134	3.25
Red/White	7041150	100	215	192	92	3.59
Black	7043077	80	340	288	206	2.90
Silver/Gold	7041286	78	280	240	162	3.05
Silver	7041062	78	265	228	150	3.12
Purple	7041063	72	138	125	53	4.37
Orange	7041060	70	195	170	100	3.37
Plain	7041021	70	125	114	44	4.38
Brown	7041061	69	212	170	115	3.06
Aaen Green	AAS12418	60	200	172	112	3.38
Yellow	7041102	50	178	152	102	2.92
Black	7041022	45	75	69	24	4.25
Green	7041168	42	135	117	75	3.05
White	7041132	35	130	111	76	2.92
Blue/Green	7041157	20	110	92	72	2.50

*P85 clutches are used in Polaris snowmobiles; P90 clutches are used in smaller Polaris snowmobiles & ATV's.

TEAM Industries Rapid Reaction Secondary Clutch Twin Trax Helix & Spring Chart

Twin Trax Helix

Part #	Track 1	Type	Track 2	Type	Part #	Track 1	Type	Track 2	Type
420400	60-38	0.46	60-38	0.46	420528	68-44	0.46	68-40	0.46
420401	56-40	0.46	56-38	0.46	420529	64-44	0.46	64-40	0.46
420402	70-40	0.46	70-38	0.46	420531	46-34	F	48-34	F
420403	60-44	0.46	60-42	0.46	420532	64-48	0.46	54-44	F
420404	66-40	0.46	66-38	0.46	420533	66-44	0.56	54-42	F
420422	70-46	0.46	70-44	0.46	420534	68-44	0.46	54-42	F
420423	34	S	36	S	420535	54-34	0.46	64-42	0.46
420430	58-42	F	62-44	F	420536	54-34	0.46	48-34	F
420437	54-44	0.46	44	S	420537	66-44	0.56	66-44	0.46
420438	56-38	0.46	56-40	0.46	420538	66-44	0.46	66-40	0.46
420440	36	S	38	S	420539	68-44	0.46	52-42	F
420451	50-42	0.46	50-44	0.46	420540	70-48	0.46	54-44	F
420453	42	S	44	S	420545	54-40	0.46	54-42	0.46
420500	70-48	0.46	66-44	0.46	420546	58-40	0.46	58-42	0.46
420506	50-36	F	56-36	0.46	420547	52-42	F	66-42	0.46
420507	48-34	F	56-34	0.46	420548	54-40	0.46	64-42	0.46
420508	46-42	F	52-42	0.46	420549	54-44	F	68-44	0.46
420509	52-40	F	58-40	0.46	420551	70-50	0.46	72-52	0.46
420510	52-36	0.46	50-34	0.46	420552	70-46	0.46	72-48	0.46
420512	54-38	0.46	54-36	0.46	420554	52-40-4	0.46	52-40-2	0.46
420513	49-40	0.46	49-38	0.46	420561	66-44	0.46	70-44	0.46
420514	54-40	0.46	54-38	0.46	530562	38	S	42	S
420515	58-40	0.46	58-38	0.46	420563	40	S	44	S
420516	54-44	0.46	54-42	0.46	420564	42	S	46	S
420517	58-44	0.46	58-42	0.46	420565	44	S	48	S
420518	60-40	0.46	60-38	0.46	420566	46	S	50	S
420519	60-48	0.46	58-44	0.46	420567	48	S	52	S
420520	54-36	0.46	54-34	0.46	420568	50	S	54	S
420521	74-48	0.46	70-48	0.46	420569	52-42	F	52-40	F
420522	74-44	0.46	74-40	0.46	420570	50-40	F	48-38	F
420523	70-44	0.46	70-40	0.46	420571	54-42	F	54-44	F
420524	66-48	0.46	66-40	0.46	420572	56-44	F	56-46	F
420525	62-44	0.46	62-40	0.46	420573	70-44	0.46	70-42	0.46
420526	58-44	0.46	58-40	0.46	420574	70-48	0.46	72-44	0.46
420527	72-44	0.46	72-40	0.46					

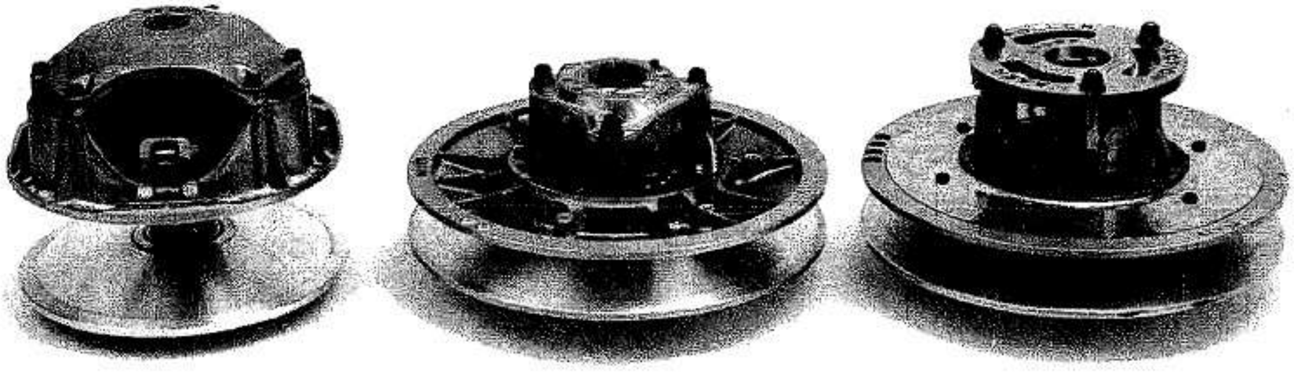
Electric Reverse Twin Trax Helix

Part #	Track 1	Type	Track 2	Type
420405	54-38	0.46	54-40	0.46
420406	54-40	0.46	54-42	0.46
420407	58-38	0.46	58-40	0.46
420408	58-42	0.46	58-44	0.46
420409	64-40	0.46	64-44	0.46
420410	68-40	0.46	68-44	0.46
420411	74-40	0.46	74-44	0.46
420412	70-44	0.46	70-42	0.46
420434	36	S	38	S
420435	42	S	38	S
420436	54-44	0.46	44	S
420439	56-38	0.46	56-40	0.46
420452	50-42	0.46	50-44	0.46
420478	42	S	44	S

S: Straight helix angle
F: Full progressive angle
0.46 : Partial progressive angle

Secondary Springs

Color	Part #	Eng. Preload	Full Shift
Red/Yellow	210035	100	150
Red/Gray	210063	125	175
Red/White	210084	100	200
Black/Yellow	210106	120	200
Red/Green	210085	120	220
Red/Dark Bl	210064	140	200
Black/Green	210114	140	220
Red/Black	210086	140	240
Black/Red	210100	155	222
Black/White	210134	160	260
Black/Purple	210193	160	240



Yamaha YVX Primary and Secondary. YVX with Roller Kit conversion.

YAMAHA

Yamaha manufactures their own clutches, except for some older SRV and V Max models using Comets. There are four clutch models: the SRX, YPZ, YXR, and YVX. The primary clutches are a continuous development of eachother, as improvements have been made from year to year. All the primary clutches feature three top-pivoted vertical style flyweights as in Comet and Polaris clutches. The flyweights have long featured holes that can be filled with a number of different steel or aluminum rivets to tune the overall mass of the flyweight. Rivets are available in increments from .8 to 4.5 grams. Yamaha also briefly had special racing flyweights for the YPZ clutch where additional tuning washers could be screwed into the backside of the flyweight to change the tuning mass. The old SRX clutches had the spider mounted with a spline, but all later models have had the spider cast solidly to the shaft. To remove the moveable sheave on these designs, you must unscrew the stationary sheave from the shaft. Yamaha has separate torque transfer towers that slide in channels with plastic shoes in the moveable cover on earlier models, or on the towers on later models. Since there are no torque transfer buttons on the roller towers, the rollers are held in place by a sleeve secured by a bolt and nut. Five different roller sizes are available: 14.5 mm, 15 mm, 15.5 mm, 16 mm, and 16.5 mm, with either 8 mm or 9 mm inside diameter. The smaller rollers will result in more aggressive shiftout, and since the rollers can be changed from the outside without disassembling the clutch, they can easily be used to change the tuning.

Yamaha has an unusually large selection of primary springs, although many springs listed eventually get phased out if they are not used in current models. Yamahas color identification code for newer models bracket the color code for the engagement, with the color code for the rate. The first and last color will always be the same, and assures that there is no confusion about the code depending on which way you hold the spring. For the 1999 season Yamaha has introduced a new "long" springcap which allows an 8 mm longer spring to be used. This allows stronger springs and more spring shims to be used without risk of coilbinding on full shift.

Early Yamaha secondary clutches have been of a conventional design, where the moveable sheave has been supported by an outside bushing riding on the helix cam. The cams have been splined to the center shaft, which was prone to wear and would allow the helix cam to "rock" on the spline and cause inconsistencies in the shifting. Yamaha solved this problem by nickel plating the helix cam on later models. This eliminated the wear, and reduced the friction between the sliding bushing and the helix. If you are racing one of these secondaries, use the nickel plated cams for best consistency.

In 1995 Yamaha introduced the YVX secondary. This clutch features a "reverse" cam design, with the cam on the outside where it can be changed in the machine without taking the clutch off. This design also offers a wide spacing between the sliding bushings which now both slide on the shaft. This new design has proven to be an excellent performer with much improved backshifting performance over the earlier conventional models. The cams have 4 holes for the spring mountings and the sheave has 3. This gives 12 possible combinations which produces preload twist angles from 10° to 120° (see table). Yamaha has 5 different cams available, all with straight angles (39, 41, 43, 45, & 47). The aftermarket has a number of different cams with progressive cuts and teflon coating on the button surfaces. The 47-41, 45-41, and 45-39 are the most used combinations.

Yamaha offers 4 secondary springs with a rate increase of 18% between each one from the softest to the stiffest. A number of aftermarket springs are also available which expands the tuning possibilities for this popular secondary. For the ultimate performance, roller conversion kits are also available from a number of aftermarket companies. (See Yamaha Roller Kit, Pg. 67)

YXR FLYWEIGHTS

Year	#	Weight (gram)	Rivet (gram)	Roller Dia. (mm)
'92	88R00	41.2	10.3	15.6
'92	89R00	43.3	13.3	15.6
'92	89F00	43.8	13.3	15.6
'92	89A00	46.7	13.3	15.0*
'93	8AY00	39.8	10.3	15.6
'93	89L00	43.0	13.3	15.6
'93	8AT00	44.0	13.3	15.6
'93	89A10	47.5	10.3	15.6

* 15.0 mm roller O.D. Do not use with '88R weight. It may cause the weight to fly past the roller.

YPZ FLYWEIGHTS

Description	Weight	Models
85L	43.8	CS340
8V8	45.4	SS440
82M90	45.6	'87 Exciter
8V0	45.8	PZ, SRV, XLV
85T	47.2	'88-'89 YPZ
84M	47.6	'89 Exciter
82M	48.8	'88 Exciter
87F	49	'90 PZII
83R	54.3	VK540

YAMAHA YPZ & YXR PRIMARY SPRINGS

Part Number	Spring Rate (Kg.mm)	Preload (Kg)	Color	Free Length (mm)	Model Used On
90501-507G2-00	1.5	20	Gold-Blue-Gold	8.7	PZ480STS, PZ480STT
90501-527G1-00	1.75	20	Red-Blue-Red	76.8	PZ480S, ES, T, ET, Std
90501-556G5-00	2.25	20	White-Blue-White	74.3	VX750S Hi Alt; VT480S
90501-607G0-00	2.75	20	Green-Blue-Green	72.7	
90501-606G9-00	3.00	20	Pink-Blue-Pink	72.1	PZ480, Hi Alt.
90501-524G5-00	1.50	25	Gold-Yellow-Gold	82.1	VX750S Std.
90501-524G4-00	1.75	25	Red-Yellow-Red	79.7	EX570T, ET (Alternate)
90501-553G0-00	2.00	25	Blue-Yellow-Blue	78	Ex 570T, ET (Alternate)
90501-553G6-00	2.25	25	White-Yellow-White	76.5	EX570R, VX750S
90501-605G7-00	2.75	25	Green-Yellow-Green	74.1	EX570T, ET (Alternate)
90501-604G0-00	3.00	25	Pink-Yellow-Pink	73.3	EX570R, ER Mid & Hi Alt
90501-521J6-00	1.50	30	Gold-Pink-Gold	85.4	EX570STT Std.
90501-550J8-00	2.25	30	White-Pink-White	78.7	VX750T,EX570STT
90501-582J1-00	2.50	30	Yellow		EX570SXT, EX570 Std.
90501-602J0-00	3.00	30	Pink-Pink-Pink		EX570SXT Mid & Hi Alt

YAMAHA SECONDARY SPRINGS

Part Number	Spring Rate (kg./m)	Free Length (mm)	Color	Model Used On
90508-40341-00	4.86	85	White	SRX440
90508-40080-00	3.19	105	None	ET250, ET300, EC340,ET340
90508-50654-00	11.2	85	Yellow-Blue	PZ480
90508-45763-00	*477	83	Red	CS340N, EN
90508-40666-00	*187	127	Gold	ET340TR, ET400TR
90508-50793-00	*571	86	Green-Yellow	EX570, VT480, PZ480
90508-50654-00	*641	85	Light Blue-Yellow	EX570, VT480, PZ480
90508-50571-00	*721	85	Pink	EX570, VT480, PZ480
90508-50746-00	*721	93.5	White	EX570, VT480, PZ480

*These springs are in kgmm/rad, instead of kg/mm as the other springs are rated. The kgmm/rad rating is a torsional (winding) spring rate.

YAMAHA YPZ & YXR TORQUE HELIX CAMS

Cam Angle	YPZ Part #	YPZ Sheave	YXR Part #	YXR Sheave
37	--------------	------------	8AT-17684-00	-------------
39	88R-17684-00	80L, 89N	88R-17684-00	88R
42	89N-17684-00	80L, 89N	89N-17684-00	88R
42	88N-17684-00	80L, 89N	--------------	88R
45	8V7-17684-00	80L, 89N	--------------	-----------
47	8V0-17684-00	80L, 89N	87X-17684-00	88R
39-45	8K4-17684-00	80L, 89N	-----------	------------

YVX Primary Springs

Yamaha Part Number	Preload Kg.	Preload Lbs.	Total Force Kg.	Total Force Lbs.	Spring Rate Kg.	Spring Rate Lbs.	Spring Type Description	Spring Color
90501-524G5-00	25	55	74	163	1.5	84	Short	Gold-Yellow-Gold
90501-524G4-00	25	55	82	180	1.75	98	Short	Red-Yellow-Red
90501-553G0-00	25	55	90	198	2.0	112	Short	Blue-Yellow-Blue
90501-553G6-00	25	55	98	216	2.25	126	Short	White-Yellow-White
90501-581 J7-00	25	55	106	233	2.5	140	Short	Yellow
90501-605G7-00	25	55	113	249	2.75	154	Short	Green-Yellow-Green
90501-604G0-00	25	55	122	268	3.0	168	Short	Pink-Yellow-Pink
90501-525 J8-00	30	66	79	174	1.5	84	Short	Gold-Pink-Gold
90501-550 J8-00	30	66	103	227	2.25	126	Short	White-Pink-White
90501-585 J3-00	30	66	119	262	2.75	154	Short	Green-Pink-Green
90501-602 J0-00	30	66	128	282	3.0	168	Short	Pink-Pink-Pink
90501-605 J5-00	30	66	135	297	3.25	182	Short	Orange-Pink-Orange
90501-555 J9-00	35	77	110	242	2.25	126	Short	White-Silver-White
90501-586 J0-00	35	77	118	260	2.5	140	Short	Yellow-Silver-Yellow
AAS12419	40	88	130	285	2.75	154	Short	Aaen Black Orange
90501-551L3-00	30	66	96	211	2.0	112	Long	Blue-Pink-Blue
90501-583L5-00	30	66	104	229	2.25	126	Long	White-Pink-White
90501-581L5-00	30	66	113	249	2.5	140	Long	Yellow-Pink-Yellow
90501-581L6-00	30	66	121	266	2.75	154	Long	Green-Pink-Green
90501-601L7-00	30	66	129	284	3.0	168	Long	Pink-Pink-Pink
90501-601L8-00	30	66	137	301	3.25	188	Long	Orange-Pink-Orange
90501-551L9-00	35	77	101	222	2.0	112	Long	Blue-Silver-Blue
90501-583L4-00	35	77	109	240	2.25	126	Long	White-Silver-White
90501-582L1-00	35	77	118	260	2.5	140	Long	Yellow-Silver-Yellow
90501-582L2-00	35	77	126	277	2.75	154	Long	Green-Silver-Green
90501-602L3-00	35	77	134	295	3.0	168	Long	Pink-Silver-Pink
90501-602L4-00	35	77	142	312	3.25	188	Long	Orange-Silver-Orange
90501-552L5-00	40	88	106	233	2.0	112	Long	Blue-Green-Blue
90501-582L6-00	40	88	114	251	2.25	126	Long	White-Green-White
90501-582L7-00	40	88	123	271	2.5	140	Long	Yellow-Green-Yellow
AAS10045	40	88	130	285	2.75	154	Long	Aaen Black Yellow
90501-602L8-00	40	88	131	288	2.75	154	Long	Green-Green-Green
90501-602L9-00	40	88	139	306	3.0	168	Long	Pink-Green-Pink
90501-583L0-00	45	99	111	244	2.0	112	Long	Blue-White-Blue
90501-583L1-00	45	99	119	262	2.25	126	Long	White-White-White
90501-603L2-00	45	99	128	282	2.5	140	Long	Yellow-White-Yellow
90501-603L3-00	45	99	136	299	2.75	154	Long	Green-White-Green

Spring **PRELOAD** controls clutch engagement RPM. Spring **RATE/TOTAL FORCE** controls clutch shift RPM.

Short springs for (8BV-CAP) (8DJ-00) Cover — Long springs for (8DF-CAP) Cover

Spring Shims For engagement Preload

Yamaha # 90201-45592 Thickness .62 mm, .025 in.
Yamaha # JH116 Thickness 1.85 mm, .075 in.

YAMAHA YVX Secondary Springs

Part Number	SPRING RATE Torsion (kgf/mm/rad)	Compression (kgf/mm)	Color
90508-500B1-00	613	0.63	Brown
90508-536A9-00	729	0.74	Red
90508-556A2-00	848	0.87	Green
90508-556A7-00	965	1.04	Silver
90508-60012-00	1211	1.256	Pink
90508-60007-00	1290	1.372	White

YAMAHA YVX Flyweight Rivets

Part Number	Length (mm)	Weight (gm)	Material
90261-06033-00	17.2	4.5	Steel
90269-06006-00	17.2	3.6	Steel w/Hole
90261-06034-00	13.9	3.6	Steel
90261-06019-00	13.3	3.1	Steel
90266-06001-00	13.3	0.85	Alum w/Hole
90266-06002-00	13.3	2.44	Steel w/Hole
90261-06017-00	11.3	2.7	Steel
90261-06015-00	10.3	2.44	Steel
90261-06028-00	10.3	0.85	Aluminum

*Heavier rivets increase shift force.*13.9 steel should not be customer installed in 8FA weight. It will not peen over properly. *10.3 & 11.3 are not long enough for RX1 weight width.

YAMAHA YVX TORQUE HELIX CAM

Cam Angle	Yamaha Part No Coating	Aaen Part Teflon Coated	I.D. Mark	
39°	8BV-17604-91	YAC12054	8BV-91	Quicker backshift under load.
41°	8BV-17604-11	YAC12055	8BV-11	
43°	8BV-17604-31	YAC12056	8BV-31	↑
45°	8BV-17604-51	--------------	8BV-51	
47°	8BV-17604-71	--------------	8BV-71	
45°-39°	----------------	YAC12059	45-39	
45°-41°	----------------	YAC12058	45-41	
47°-41°	----------------	YAC12057	47-41	↓
51°-43°	8FA-17604-00	--------------	8FA-00	Quicker upshift during acceleration.
52°-45°	----------------	YAC12090	52-45	
52°-49°	----------------	YAC12103	52-49	
54°-47°	----------------	YAC12092	54-47	

Larger angles give aggressive upshift, smaller angles produce higher shift speeds

Secondary Spring Preload (Twist Angle)

Fixed Sheave	Helix Cam Hole Number			
Hole Number	0	3	6	9
1	10°	40°	70°	100°
2	20°	50°	80°	110°
3	30°	60°	90°	120°

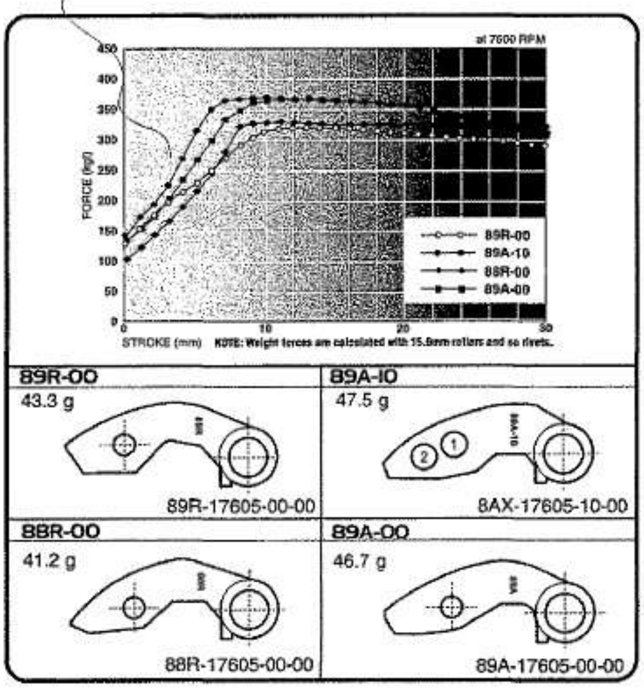
Yamaha YVX Primary Rollers

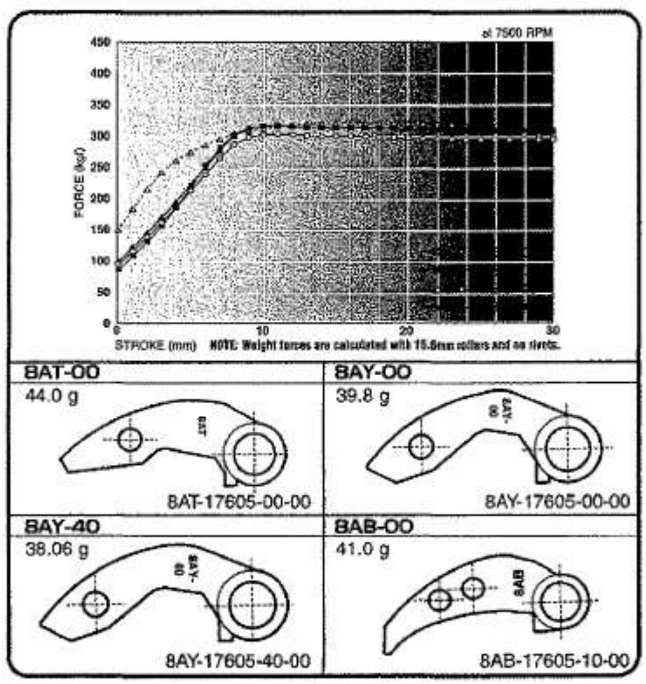
8 mm Bushing: 90380-08221-00		9 mm Bushing: 90380-09245-00	
Roller Dia. (mm) O.D.	Part # 8 mm I.D. Roller w/Bushing	Part # 9 mm I.D. Roller w/Bushing	Force Effects
14.5	8AB-17624-40-00	8CR-17624-00-00	Increased
15.0	8AB-17624-00-00	8CR-17624-10-00	↑
15.6	8AB-17624-10-00	8CR-17624-20-00	
16.0	---------------------	8CR-17624-30-00	↓
16.5	8AB-17624-30-00	8CR-17624-40-00	Decreased

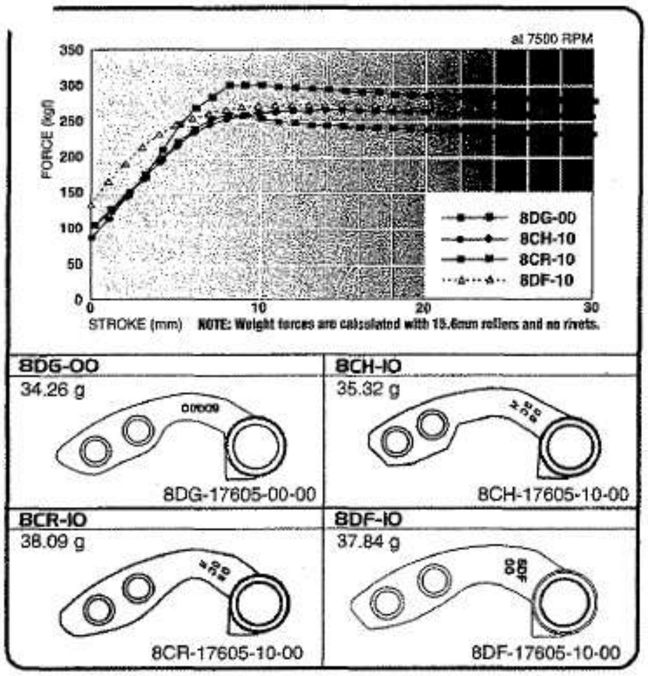
Yamaha YVX Flyweights

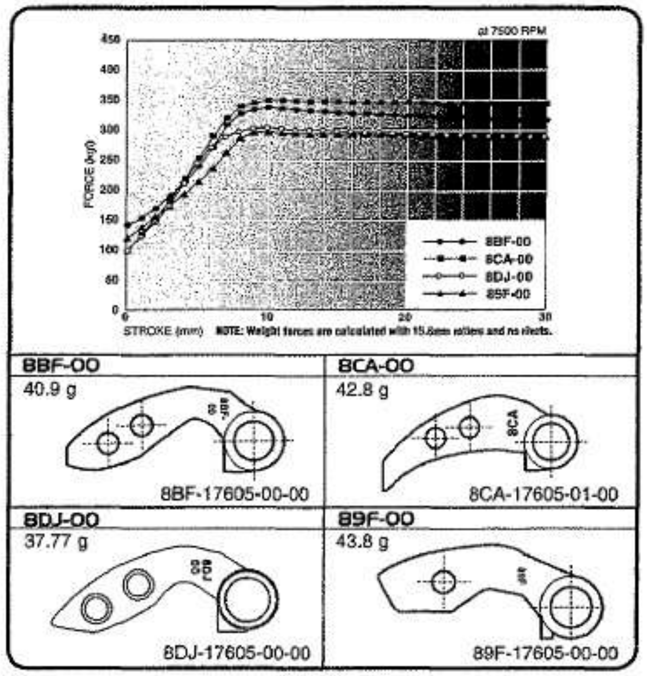
I.D. #	Weight (gr)	Yamaha Part #
8D6-00	34.3	8D6-17605-00-00
8CH-00	35.3	8CH-17605-10-00
8DJ-00	37.8	8DJ-17605-00-00
8DF-00	37.9	8DF-17605-10-00
8CR-00	38.1	8CR-17605-10-00
8AY-40	38.1	8AY-17605-40-00
8EK-00	39	8EK-17605-00-00
8DN-10	39.8	8DN-17605-10-00
8AY-00	39.8	8AY-17605-00-00
8BF-00	40.9	8BF-17605-00-00
8AB-01	41	8AB-17605-01-00
88R-00	41.2	88R-17605-00-00
8CA-01	42.8	8CA-17605-01-00
89L-00	43	98L-17605-00-00
89R-00	43.3	89R-17605-00-00
89F-00	43.8	89F-17605-00-00
8AT-00	44	8AT-17605-00-00
8DN-00	44.3	8DN-17605-00-00
8BU-10	45.41	8BU-17605-10-00
89A-00	46.7	89A-17605-00-00
89A-10	47.5	89A-17605-10-00
8BV-00	48.7	8BV-17605-00-00
8FA-10	69.43	8FA-17605-10-00
Four-Stroke Weights		
8FS-00	65.5	8FS-17605-00
8FP-00	67.8	8FP-17605-00
8FN-00	75.3	8FN-17605-00

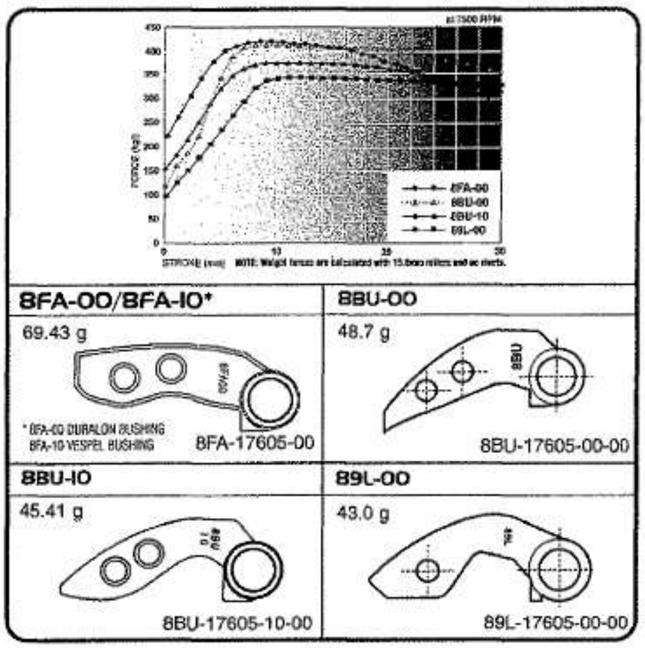
Flyweights have been grouped in families. The flyweights in each group are the most likely to be used in tuning equal size engines. In the tables following, the stock weight of popular models are listed with other compatible weights.

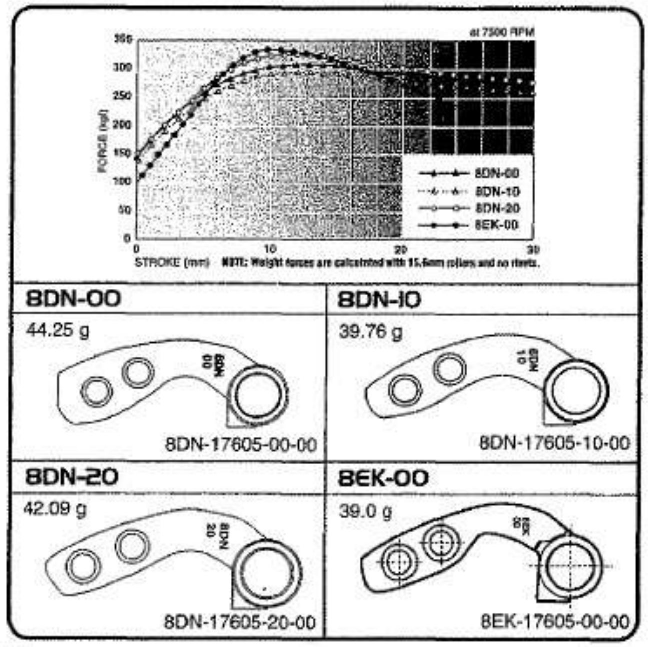












Model	Year(s)	Stock wt.	Compatible weights	Model	Year(s)	Stock wt.	Compatible weights
RX-1, RX-1 Mountain	'03-'04	8FA-00	8BU-10, 8BU-00, 89L-00	SX Viper R/ER/S	'02-'04	8EK-00	8DN-00, 8DN-10, 8DN-20
SRX 700	'00-'02	8DN-20	8DN-00, 8DN-10, 8EK-00	SX Venom 600	'04	8DG-00	8CH-10, 8CR-10, 8DF-10
SRX 700	'99	8DN-10	8DN-00, 8DN-20, 8EK-00	700 SX	'97-'99	8CH-10	8DG-00, 8CR-10, 8DF-10
SRX 700	'98	8DN-00	8DN-10, 8DN-20, 8EK-00	600 SX	'99	8DG-00	8CH-10, 8CR-10, 8DF-10
Mountain SRX 700	'98	8DN-00	8DN-10, 8DN-20, 8EK-00	Phazer 500	'99-'01	8DJ-00	8BF-00, 8CA-00, 89F-00
SRX 600	'98-'99	8DF-10	8DG-00, 8CH-10, 8CR-10	Phazer 480	'97-'99	8BF-00	8CA-00, 8DJ-00, 89F-00
SX-R 700	'00-'01	8DF-10	8DG-00, 8CH-10, 8CR-10	SX-R 500	'00-'02	8DJ-00	8BF-00, 8CA-00, 89F-00
SX-R 600	'00-'03	8DG-00	8CH-10, 8CR-10, 8DF-10	500 SX	'99	8CR-10	8DG-00, 8CH-10, 8DF-10
SX Viper	'03-'04	8DN-10	8DN-00, 8DN-20, 8EK-00	V Max 800	'97	8BU-10	8FA-00, 8BU-00, 89L-00
Mountain Viper	'03-'04	8DN-10	8DN-00, 8DN-20, 8EK-00	Mountain Max 800	'97	8BU-10	8FA-10, 8BU-00, 89L-00

ARCTIC CAT

Arctic Cat's early primary clutches were of the "hex shaft" design. This referred to the hex shaped fiber bushing riding on the hex center shaft and acting as the torque transfer point. The hex clutch had a roller arm working against a stationary cam, and was easy to tune by changing weights on the roller pin. When engine power increased, Arctic changed to a Comet drive clutch.

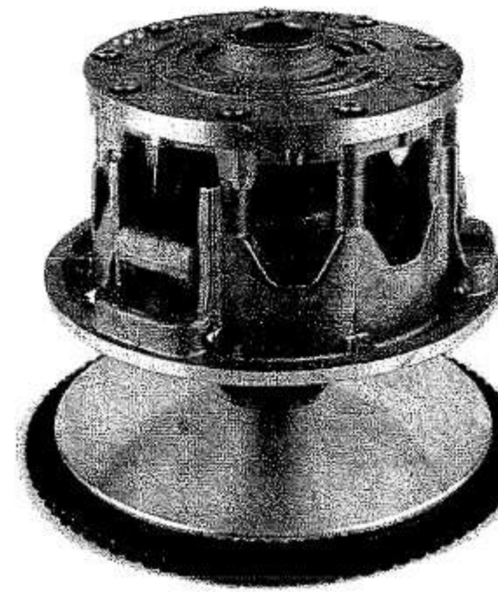
The Comet clutch remained standard in the Arctic lineup for many years, until Arctic developed their own version with three additional support towers for the cover to stiffen up the assembly to take the additional torque loads of the larger twin and triple engines. Although the design follows the same proven layouts as pioneered by Polaris, Comet, and Yamaha, there are subtle differences in the relationship between the rollers and flyweight pin. This results in a softer shiftout in the beginning of the sheave travel. Race drivers who want a more aggressive acceleration often re-grind the shift curve or use a smaller roller. The Arctic and Comet weights are interchangeable, but the springs are not. The Arctic springs have a smaller inside diameter.

For the 2002 models, Arctic introduced a new six tower primary clutch for the bigger twin cylinder models. This clutch has a more aggressive relationship between the flyweights and rollers, and more space for stronger pressure springs. The castings are heavily reinforced to take higher loads from the powerful twin engines.

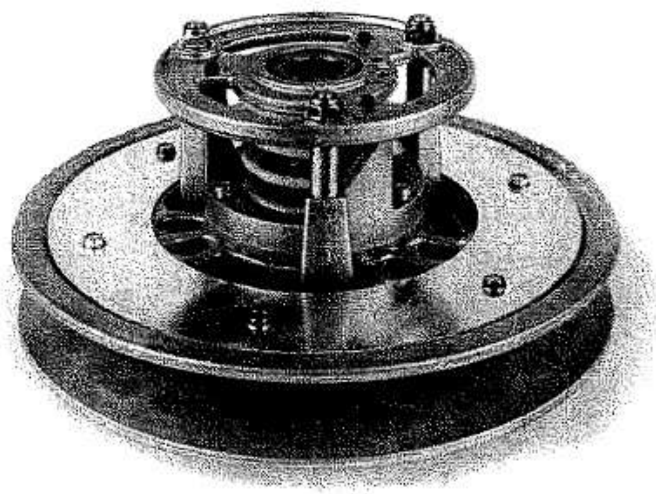
Arctic has long been known for its "reverse cam" secondary design. This design locates the cam assembly on the outside and away from the tunnel. Arctic's force philosophy utilizes stiffer springs and larger cam angles than Polaris, and the clutch has often been a favorite with drag racers because of its hard shiftout and good acceleration properties.

The regular "reverse cam" Arctic secondary uses plastic buttons riding against a die cast aluminum cam. Lots of cams are available as tuning components, both from Arctic or in billet cut form from a number of aftermarket companies. Teflon coated versions of the billet cams are also available for improved backshifting and acceleration.

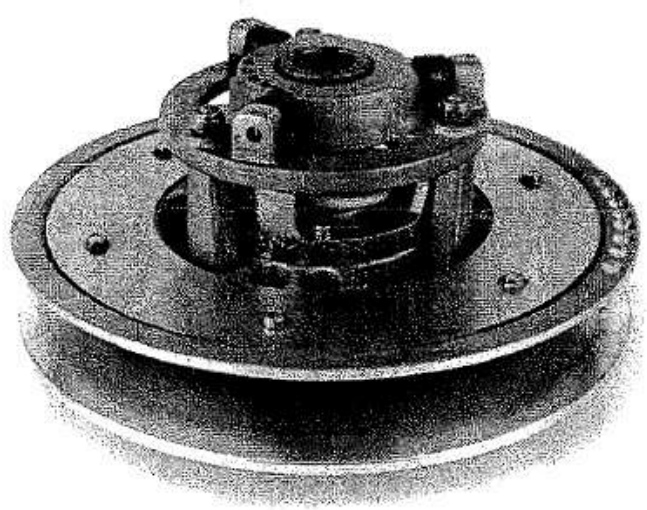
Arctic was the first manufacturer to rollerize their secondary. The first two designs used different covers than the present design, but for the 98 season, Arctic designed a full-purpose roller cover which locates the plastic rollers on the outside of the cover. This clutch has shorter mounting towers than the older unit, and is often referred to as the "short tower" version. Arctic driven cams can be used on either unit. Steel roller and steel cam conversions for the Arctic secondary can be found on page 67.



Arctic Nine Tower Primary



Arctic"Tall Tower" Reverse Cam Secondary



Arctic"Short Tower" Roller Secondary

Popular Arctic Cat Flyweights

Weight Grams	Arctic Part #	Weight Grams	Arctic Part #
71.5	0746-640	48.5	0746-582
70.5	0746-634	48	0746-638
68	0746-633	48*	0746-546
67	0746-650	48	0746-566
66	0746-616	47.5*	0746-533
64	0746-614	47.5	0746-608
63.5	0746-653	47	0746-570
63	0746-636	47	0746-611
62	0746-615	47*	0746-619
60	0746-617	47	0746-523
58	0746-635	46.5*	0746-528
57	0746-646	46.5	0746-540
56.5	0746-655	46.5	0746-579
55.5	0746-559	46.5	0746-583
55	0746-560	46	0746-576
54.5	0746-588	46	0746-549
54	0746-530	46	0746-607
54	0746-500	45.5	0746-593
53	0746-539	45*	0746-536
52.5	0746-581	45	0746-584
52.5	0746-535	45*	0746-586
52	0746-645	45	0746-589
52	0746-609	44.5	0746-502
52	0746-592	44.5	0746-531
52	0746-605	44.5	0746-594
51.5	0746-618	44.5	0746-596
51.5*	0746-534	44	0746-525
50.5	0746-642	44*	0746-532
50.5	0746-527	44	0746-591
50	0746-610	44	0746-598
50	0746-561	43.5	0746-565
50	0746-568	43.5	0746-597
50	0746-587	42.5	0746-602
50	0746-600	42	0746-526
49.5	0746-538	42	0746-562
49.5*	0746-529	41.5	0746-595
49	0746-658	41.5	0746-604
49	0746-647	41	0746-503
49	0746-585	40.5*	0746-590
48.5	0746-606	39.5*	0746-537
48.5	0746-501	* Denotes a notch for higher engagement.	
48.5	0746-524		

Popular Arctic Comet Flyweights

Weight Grams	Arctic Part #	Weight Grams	Arctic Part #
56.5	0646-098	46.5	0646-021
54	0646-099	46.5*	0646-199
54	0646-163	45	0646-078
53	0646-117	45*	0646-156
52.5	0646-100	44.5	0646-027
51.5*	0646-165	44.5*	0646-250
50.5	0646-102	44.5	0646-530
50.5	0646-105	44	0646-002
50	0646-249	44	0646-079
49.5	0646-124	44	0646-525
49.5*	0646-164	44	0646-162
48.5	0646-031	43.5	0646-015
48.5	0646-080	43.5	0646-018
48	0146-514	42	0646-019
47.5*	0646-157	41	0646-030
47	0646-123	40	0646-085
46.5	0646-009		

Arctic Cat Drive Clutch Springs

Color	Engagement Load-lbs.	Full Shift Load-Lbs.	Spring Rate Lbs./In.	Arctic Part #
Or/Wht	143	290	118	0646-248
Yel/Wht	122	285	130	0646-229
Purple	121	240	95	0646-155
Yel/Grn	114	267	153	0646-147
Red	74	228	154	0646-149
Silver	72	188	93	0646-150
Blue	68	224	179	0646-148

Arctic Cat - Comet Drive Clutch Springs

Color	Engagement Load-lbs.	Full Shift Load-lbs.	Spring Rate Lbs./In.	Arctic Part #
Purple	136	254	92	0646-154
Yel/Grn	134	264	104	0646-147
Red	92	222	104	0646-083
Silver	85	169	67	0646-096
White	78	178	80	0646-084
Blu-Red	68	197	129	0646-097

Aaen Primary Racing Springs

Color	Engagement Load-lbs.	Full Shift Load-lbs.	Spring Rate Lbs./In.	Aaen Part #
Gold	190	335	11	AAS10042
Maroon	185	320	108	AAS10041
White	170	330	128	AAS10043
Yel/Red	165	310	116	AAS10040
Alm/Blu	165	280	92	AAS10039
Yellow	150	380	172	AAS10038
Purple	150	320	127	AAS10034
Orange	120	290	127	AAS11054

Arctic Driven Springs

Color	Sideload at 1.43" - Lbs	Arctic Part #
Green	157	0748-025
Red/Wht	116	0648-114
Yellow	92	0148-227
Blue	79	0648-012
White	58	0648-010
Black	42	0148-176

Popular Arctic Driven Cam Angles

Cam Angle	Part No.	Cam Angle	Part No.
42°	0648-026	53°- 51°	0648-126
45°	0148-180	55°	0648-005
47°	0648-025	55°- 53°	0648-424
48°- 44°	0648-011	57°	0648-006
49°	0648-014	57°- 50°	0648-016
51°	0148-222	57°- 50°*	0648-107
51°- 49°	0648-125	62°- 54°	0648-656
52°- 44°	0648-001	70°- 35°	0648-655
53°	0648-002	77°- 32°	0648-773

* .100" thicker at base than 0648-016

ROLLER ACTION CLUTCHES

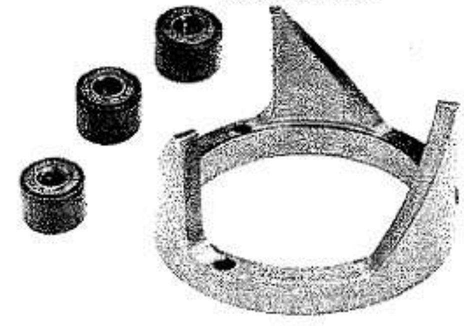
STEEL NEEDLE BEARINGS ROLLING ON A HARDENED STEEL CAM: THE ULTIMATE IN PERFORMANCE

ARCTIC

Pre 98 Roller Cover

Adjustable Spring Cover

99-05 Roller Kit



Arctic Roller Action Kits bolt directly on the reverse cam Arctic secondary clutch in just minutes. The hardened steel cam and the steel needle bearing roller eliminates any sliding friction or flex as found with plastic rollers working against aluminum cams. The result is ultra quick backshifting and acceleration which out-performs any roller clutch on the market. The Pre-98 Roller Kit consists of a roller cover and a hardened steel cam.

Complete Pre 98 Kit Cam . . . .Give model & cam angle

Spare Parts

Pre 98 Cover. ACP12709
Hardened Steel Cam . . . . See Chart
Needle Roller Bearing Pre 98 (each) ACP12640

Aaen Brown Spring

This secondary spring has an extra heavy torsion rate and is longer for more initial belt pressure. AAS12416

The new 99-05 short tower kit consists of a hardened steel cam and three needle bearing steel rollers to replace the plastic roller in the stock cover.

Complete Kit 99-05 (cam/ rollers) . . . .Give model & cam angle

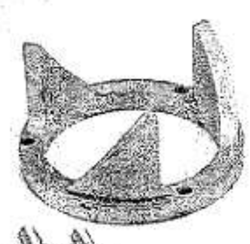
HARDENED STEEL CAMS

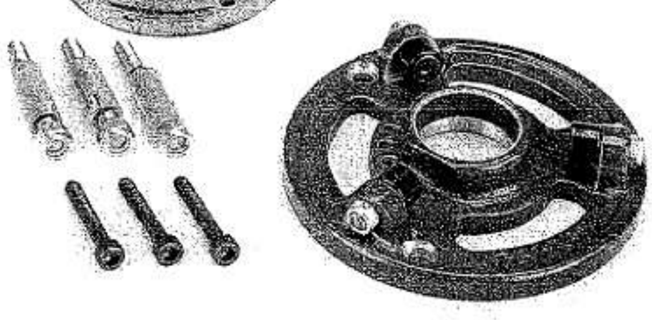
Pre 98	44°, 46°, 48°, 50°, 48°-44°, 50°-46°, 52°-48°, 54°-50°, 56°-40°, 56°- 50°
98 & Up	48°, 51°, 50°-46°, 52°-48°, 56°- 44°

YAMAHA

*HELIX CAM CHART

46° Cam	YCP12095
48° Cam	YCP12103
50° Cam	YCP12096
52-48° Cam	YCP12074
54-46° Cam	YCP12098
56-44° Cam	YCP12099





YVX "Roller Action" Secondary Kit

- **Faster acceleration, quicker backshift**
- **Cooler running belts and less belt wear**
- **Uses the original YVX secondary clutch**
- **Uses hardened steel cam and steel needle rollers to give superior reaction**

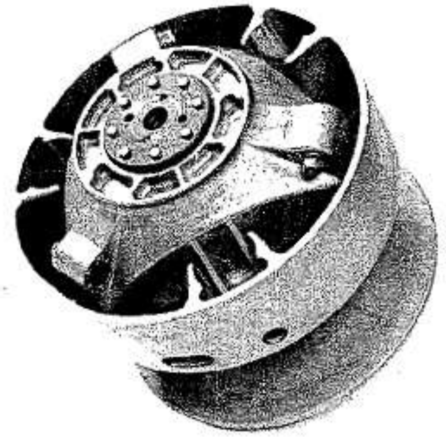
The kit consists of a hardened steel cam, a "Roller Action" cover with cam follower needle bearings, a hardware kit and a spring. We now also offer the Adjustable Spring Tensioner for quick pretension adjustment.

To install the kit, the moveable sheave has to be machined to remove the button towers and provide a mounting surface for the cam. This is a very simple machining procedure that can be done by anyone with a lathe. Complete machining instructions are included in the kit. We also offer complete machining and set-up of your clutch with our "Roller Action' kit. Send us your secondary clutch, quick turn around !

Machining & Set-up LBR12073
Yamaha YVX Roller Action Cover YCP12102
Hardware Kit YCP12072
Roller Action Cover w/Spring Adjuster YCP12110
AAEN Brown Secondary Spring AAS12416
Spare Bushing, Roller Cover Only ACP12721
Spare Bushing, Adjuster Cover ACP12784
Cam Spacer Plate (.100"Thick). ACP0148237

***See Chart For Helix Cam Selection**

SKI DOO TRA CLUTCH



Ski-Doo's TRA (Total Range Adjustable) clutch has a number of unique features that should make it easy to tune and work on.

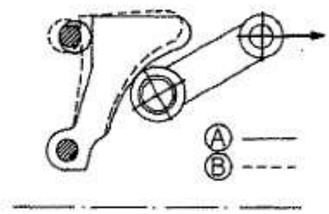


Fig. 1 The adjustable cam in principle. A roller arm is mounted in the movable sheave. The cam pivots on the bottom and rests against an eccentric shaft on the top. Depending on the position of the eccentric, the angle of the ramp changes. Line shows the ramp all the way forward, which gives higher engagement and shift speed. Line B shows the ramp all the way back which gives lower engagement and shift speed.

The Ski-doo TRA (total range adjustable) clutch has proven to be a good performance clutch with a number of unique features that makes it easy to tune. An adjustable ramp is mounted in the outside cover, and the roller arm assemblies are mounted in the movable sheave. Torque is transferred to the movable sheave through spring loaded buttons located in ears on the stationary cover. The springs eliminate noisy rattles at lower engine speeds, but adds some friction during shifting. Racers often eliminate the button springs to improve down shifting. When assembling the spring loaded buttons you need a special tool to hold the buttons in place.

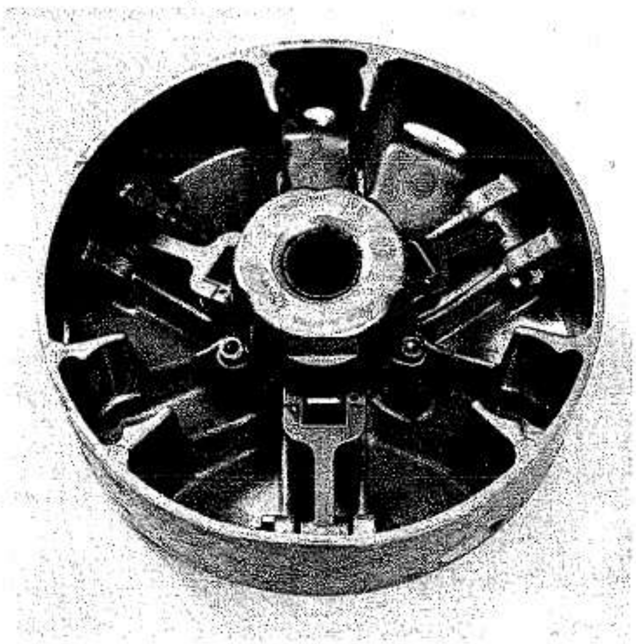
In 1996 Ski-doo introduced a torsional dampener to the outside cover. While the outside cover had earlier been riveted directly to the steel center mounting plate, the new design has a rubber dampening disc sandwiched between the cover and the mounting plate. This makes the outside cover mass act as a torsional dampener and improves both clutch and engine component life.

Tuning with the TRA

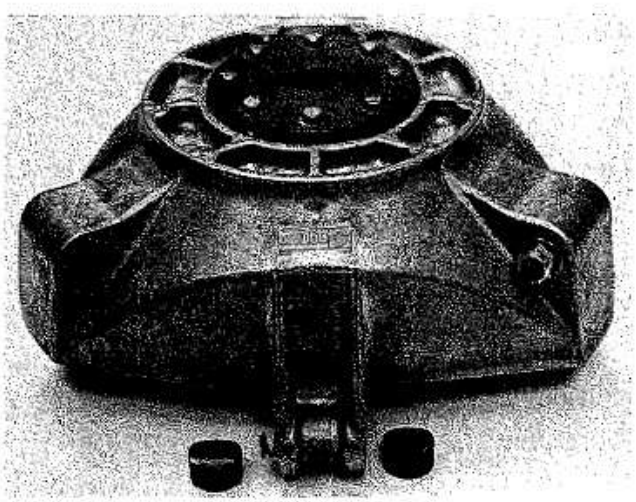
Ski-doo now has available a large assortment of cams and springs. In addition, the aftermarket also offers special ramps, springs, lightweight arms, different weighted pin and washers, and an assortment of rollers. With all these component choices, plus the six adjustments on the cam position, this clutch has more tuning choices than any other on the market. In fact, it is easy to get lost if you don't keep good notes. The geometry of the shift mechanism tends to favor hard initial acceleration, and most Ski-doo engines produce good midrange torque to take advantage of the TRA clutch. The common tuning procedure is to first try different positions on the cam, and then adjust engagement or shift speed with different springs. Fine tuning is done by adding or subtracting weight on the pins. If none of this produces satisfactory results, the procedure starts over with a new ramp. Fine tuning can also be accomplished by selecting different helix angles and secondary springs.

There is a large selection of ramps available, but good results are obtained by using the 280, 286, or 300 ramps on large twins or triples with strong midrange acceleration. On higher revving mod sleds the 281 is sometimes used, but the CFI is the most popular ramp for racing and can be used in many applications. The CFI was developed by Ski-doo specifically for racing and is more expensive than regular ramps. Goodwins GP3 ramp is a similar high RPM mod ramp at a reasonable price, while the Goodwin GP1 and GP2 ramps are used for aggressive shifting with lower revving high torque engines. High engagement is accomplished by moving the engagement portion away from the roll pin as in the CFI or by providing a notch as on the 281 ramp. Higher shift RPM is produced by keeping the final shift angle closer to parallel with the clutch shaft (Line A in fig. 1).

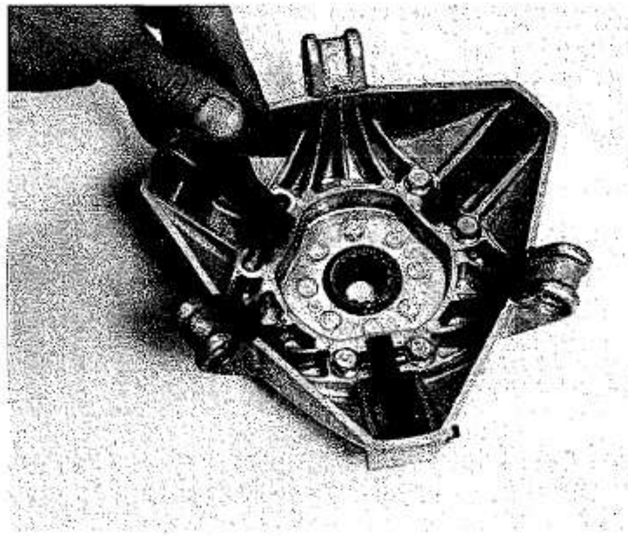
Ski-doo has a wide selection of springs available for tuning. Engagement preloads vary from 70 to



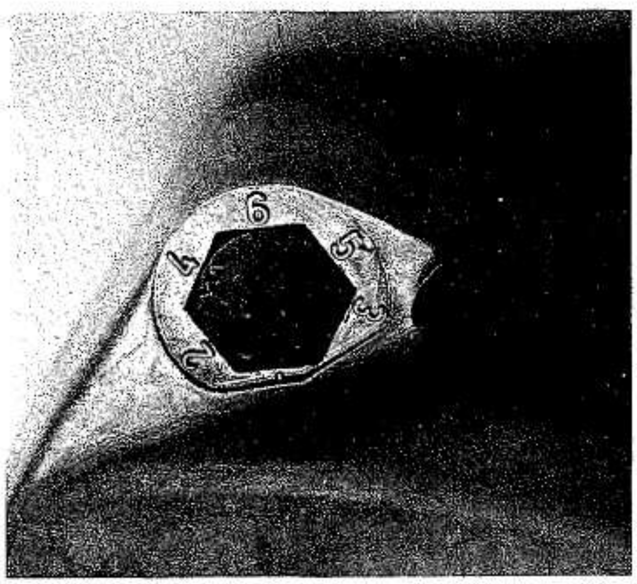
The movable sheave holds the roller arm assembly. The spring is contained by a cast cup with the outside sliding bushing. This makes it easy to take the clutch apart. Torque transfer takes place against machined surfaces on the outside diameter.



Torque transfer buttons are located on ears on the stationary casting. The buttons are spring loaded.



The ramp sits in the stationary casting and pivots on the bottom. An eccentric cam on the top moves the cam in or out to change the angle. The casting is mounted to the shaft on a spline, making it easy to remove. No special tools are need to work on this clutch.



The eccentric cam is turned by changing the bolt head to different positions. This operation is simple and can be done quickly from the outside.

310 lbs., and full shift loads from as little as 170 lbs., to as much as 510 lbs. Arms come in several designs, from a magnesium racing arm, to the standard aluminum arms, and also several after-market billet and composite arms.

Rollers can be had in two sizes from Ski Doo, while Thunder Products offers two larger sizes, one and two mm larger than stock. Pins are available from aluminum to solid steel, and also with threaded centers where set screws can be added by as little as a half a gram at a time.

Ski Doo's new secondary for electronic engine reverse models has helixes with opposing cam surfaces for forward and reverse operation. Springs on this model are the straight pressure type, not torsionally wound. Newer models also have a rollerized secondary for increased belt life and better backshift.

TRA Pin Kits

Aluminum threaded pin w/set screws, 4.5 - 9.0 grams	SCP12010
Steel threaded pin w/set screws, 10.8 -15 grams	SCP12014
Heavy steel threaded pin w/set screws 16.5-22 grams	SCP12015
Solid steel, 23 grams	SCP12009
Solid steel, 24 grams	SCP12008

TRA Components

Cam Arm Combination	Gr. Wt. -No. of Set Screws				
	4	3	2	1	0
Std. Arm-Std. Roller-Steel pin w/Screws	57.84	57.13	56.42	55.71	55.00
Std. Arm-375 Roller-Steel pin w/Screws	55.84	55.13	54.42	53.71	53.00
Std. Arm-Std. Roller-Alum. pin w/Screws	----	51.24	50.82	50.40	49.98
Std. Arm-375 Roller-Alum.pin w/Screws	----	48.69	48.27	47.85	47.43
Mag Arm-Std. Roller-Steel pin w/Screws	46.78	46.07	45.36	44.65	43.94
Mag Arm-375 Roller-Steel pin w/Screws	44.21	43.50	42.79	42.08	41.37
Mag Arm-Std. Roller-Alum. pin w/Screws	----	39.42	39.00	38.58	38.16
Mag Arm-375 Roller-Alum. pin w/ Screws	----	37.15	36.73	36.31	35.89

Ski-doo Performance Secondary Cams

Hi Performance Formula Secondary Cam

Straight and progressive angle X-Team helix Aerospace coating for the optimum in ultra smooth performance and durability • Tapered "lead-in edge" for easy installation • Spring holes chamfered to prevent spring bind

Formula secondary clutch only

Angle	Part No.	Angle	Part No.
40°	860424800	47°-44°	860426300
42°	860424900	50°-37°	860426400
44°	860425000	50°-40°	860426500
47°	860425100	50°-42°	860426600
50°	860425200	50°-44°	860426700
53°	860425300	50°-47°	860426800
40°-37°	860425400	53°-40°	860426900
40°-44°	860425500	53°-42°	860427000
42°-37°	860425600	53°-44°	860427100
43°-47°	860425700	53°-47°	860427200
44°-37°	860425800	53°-50°	860427300
44°-40°	860425900	56°-44°	860427400
47°-37°	860426000	56°-47°	860427500
44°-40°	860426100	56°-50°	860427600
47°-42°	860426200		

Secondary Springs

Color	Part #
Titanium	406 1040 00
Beige	414 558 900
White	504 152 070
Violet RER	414 987 300
Black	417 126 687

Popular TRA Ramps

Model	Shift Profile	Part #
CF1	Hi Eng, Med Shift, Hi RPM	415 0238 00
293	Hi Eng, Light Shift, Hi RPM	417 0052 93
281	Hi Eng, Med Shift, Hi RPM	420 4802 81
228	Hi Eng, Med Shift, Hi RPM	417 0052 28
280	Hi Eng, Aggr Shift, Med RPM	417 0052 80
285	Med Eng, Med Shift, Med RPM	417 0052 85
286	Med Eng, Med Shift, Med RPM	417 0052 86
300	Med Eng, Aggr Shift, Med RPM	417 222 381

Formula RER Secondary Cam

Angle	Part Number
47°	417 126 683
48°-44°	417 126 685
50°	417 126 716
50°-47°	417 126 680

Anodized RER Formula Cam

Angle	Part Number
44°	417 126 747
47°	417 126 748
48°-44°	417 126 750
50°	417 126 749
50°-47°	417 126 751

Anodized, Formula VSA Cam

Angle	Part Number
44°	417 126 718
48°-44°	417 126 719
50°	417 126 704
50°-47°	417 126 720

Anodized HPV, VSA Cam

Angle	Part Number
44°	417 126 445
47°	417 126 577
47°-40°	417 126 724
47°-44°	417 126 385
50°-40°	417 126 721
50°-47°	417 126 580

High Performance HPV™ 27 RER™ Secondary Cam

Specifically designed for the RER HPV-27 and HP-27 VSA secondary clutch • Aerospace coating for the optimum in smooth performance and durability

Angle	Part Number
47°	860 428 000
50°	860 428 000
52°-47°	860 428 100
50°-47°	860 428 200
50°-44°	860 428 300
47°-44°	860 428 400

TRA Components

Part	Wt. in Grams
Std. lever w/flanged bushing	34.02
Mag lever w/flanged bushing	22.40
Standard Roller	9.64
.375 Roller	7.09
Standard pin	9.92
Aluminum pin (empty)	5.1-6.8
Standard shim	.14
Thick shim .125	.27
Flange bushing	.57
Nylon shim	.28
Cotter pin	.57

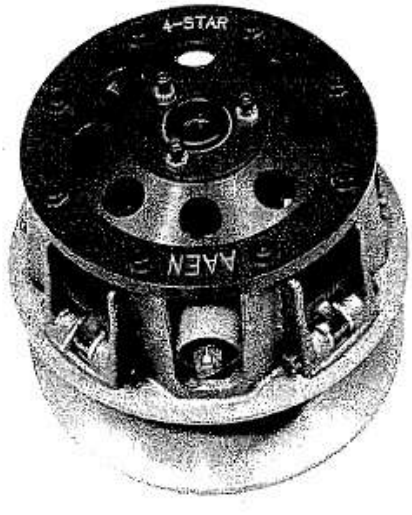
Ski-Doo OEM Drive Springs

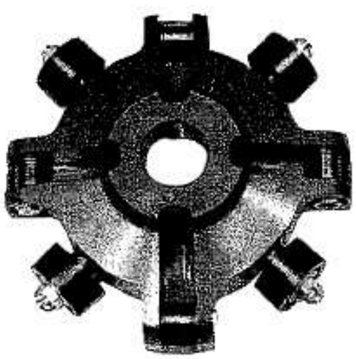
Color	Eng . Load-lbs.	Full Shift Load-lbs.	Part No.
Rd/Rd	70	170	414 6898 00
Rd/Or	70	200	414 0152 00
Rd/Yl	70	230	414 8175 00
Rd/Gr	70	260	414 6892 00
Rd/Bl	70	290	414 6915 00
Rd/Pu	70	320	414 7010 00
Yl/Rd	100	170	414 9930 00
Yl/Gold	100	200	414 6897 00
Yl/Yl	100	230	414 7496 00
Yl/Gr	100	260	414 7421 00
Yl/Bl	100	290	414 8180 00
Yl/Pu	100	320	414 6784 00
Bl/Or	130	200	414 6390 00
Bl/Yl	130	230	414 6895 00
Bl/Gr	130	260	414 8177 00
Bl/Bl	130	290	414 6894 00
Bl/Pu	130	320	414 8178 00
Bl/Pi	130	350	414 9163 00
White	150	240	414 6056 00
Pu/Yl	160	230	414 0153 00
Pu/Gr	160	260	414 0154 00
Yellow	160	270	414 6055 00
Pu/Bl	160	290	414 0349 00
Pu/Pu	160	320	414 8179 00
Pur/Pk	160	350	414 9495 00
Black	185	410	415 0195 00
Titanium	185	410	486 1032 00
Pi/Gn or Gr/Bl	200	290	414 7682 00
Gr/Pu	200	320	414 7628 00
Green or Gr/Pi	200	350	414 7569 00
Gr/Wh	200	380	414 222 371
Yl/Rd or Pi/Pu	230	320	414 7542 00
Bl/Rd or Pi/Pi	230	350	415 0748 00
Rd/Wh or Pi/Wh	230	380	415 0194 00
Green	230	390	415 0196 00
Red	230	410	415 0197 00
Titanium	240	370	486 1035 00
Blue	240	430	415 0198 00
Wh/Wh	250	380	417 2220 04
Orange	250	420	415 0200 00
Pink	250	460	415 0199 00
Titanium	260	380	416 1033 00
Wh/Sil	260	420	417 2221 64
Titanium	280	410	486 1034 00
Gr/Gr	280	420	415 0201 00
Rd/Rd	280	460	415 0202 00
Bl/Bl	280	510	415 0203 00
Pk/Pk	310	460	415 0204 00
Gold/Gold	310	510	415 0205 00

NEW TECHNOLOGY

FRONT ROLLER CLUTCH (Driver)

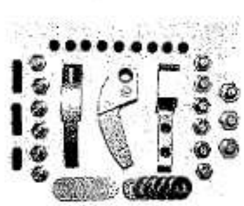
Aaen 4-Star Roller Clutch





If secondary clutches benefit from removing friction by substituting roller for plastic buttons, why not do the same with the torque transfer buttons on the engine driving clutch? We have experimented with different versions since 1995, and the results have been encouraging. The first front roller conversion was made on a Ski Doo TRA clutch, and the results were excellent, but component life was short. Improving on the TRA life would have meant a new cover casting, but this would have limited the application to just Ski Doo's. Instead we decided to rollerize a Comet 4-Pro primary clutch, since this is available for all brands. The rollers run in channels, with a small clearance so they only contact one side at a time. With four rollers instead of three, it can also take higher loads, and the mounting on the spider is stronger. Component life has been excellent with the 4-Pro rollers. The rollers are are made of a shock resistant plastic with ball bearings inside. This gives the rollers a chance to absorb some of the shock loads, and still roll freely. After five seasons of use, the feedback on the 4-Star Roller has been good. Quicker acceleration, improved backshift, and increased top speed has been recorded. The increased top speed seems to be from the ability of the clutch to shift into a higher ratio when the opposing fricti~~on-~~forces are removed. Users also comment on a much smoother engagement and take-off, and the improved ability to lug hard loads at low speeds without excessive belt slip. A billet cover with adjustable engagement speed is now also available.

Heavy Hitters



The ultimate in high performance weights. Can be adjusted in low range while on the clutch. Adjustable in midrange and top end with 0.4 gram resolution. Fits all models, no machining required. Perfectly gram weight match balanced. Comes with a complete set of tuning components and tuning manual. You can add 13 grams to each weight. Tungsten washers available to add 3.4 gr. more weight per arm.

50 Gram Base Weight Kit CLATSKHH50
55 Gram Base Weight Kit CLATSKHH55
60 Gram Base Weight Kit CLATSKHH60
65 Gram Base Weight Kit CLATSKHH65
Yamaha Base Weight Kit .... CLATSKHHYAM
Quad Cam Base Weight Kit CLATSKHH4QU
Tungsten Washers (ea) CLATSK500

Heel Clickers

This unique design makes it possible to use heavy force for acceleration and still maintain top RPM without the use of radical helix angle splits. Fully adjustable by adding up to 7 grams to the tip or washers to the heel portion. Billet cover for use with bigger springs is also available. A conversion kit to use Heel Clickers in the Ski Doo TRA clutch results in a substantial top end speed increase. Specify model & 3 or 4 tower.

Base Kit	Arctic Polaris	Comet	Yamaha
	3 Tower	4 Tower	** 3 Tower Direct
45 Gram	CLASTS2HC45-7	CLASTS4HC45-7	bolt-on. ISR Stock
50 Gram	CLASTS2HC50-7	CLASTS4HC50-7	Legal. 40 gr. +10.
55 Gram	CLASTS2HC55-7	CLASTS4HC55-7	CLASTS33HC40-10Y

Ski Doo 55-70 Gram Kit CLASTS1TRA45-10
Billet Cover -Polaris CLASTSHC2POL
Billet Cover -Arctic Cat CLASTSHC2AC

Belt Drive Transmissions for ATV's

When Polaris introduced their entry into the ATV market the machine had an automatic "PVT" (Polaris Variable Transmission) belt drive transmission. This was a natural step for the Minnesota snowmobile manufacturer, since a belt drive transmission is the only acceptable transmission for a machine that needs to keep constant power to the track to stay on top of the snow. Polaris reasoned that the proven belt transmission with its easy operating features would be a bonus on the new ATV.

ATV's proved to have a few more challenging conditions than the cold snow environment. First of all the transmission needed to be completely enclosed to protect it from dirt, mud, and water. The enclosed environment quickly produced heat that affected belt wear, so an air circulating system was needed. This was accomplished by adding a fan to the back of the engine clutch, and ducting the air supply into this area from a tall snorkel with its intake located up under the tank. An outlet hose also exited the used air high up, and in the case of the air-cooled models the outlet air was directed onto the head cooling fins.

The Polaris primary (engine) clutch was a snowmobile unit with a smaller diameter which was shortened down to take a narrower belt. The roller spider was spaced further out to permit more angle on the flyweight in order to get a lower engagement speed, smoother takeoff and better low speed performance on tight trails. At first the automatic belt transmission was looked down on by the California crowd who raced "real" ATV's with gearboxes, but as Polaris introduced more powerful models, the convenience of the automatic belt drive system started to gain converts.

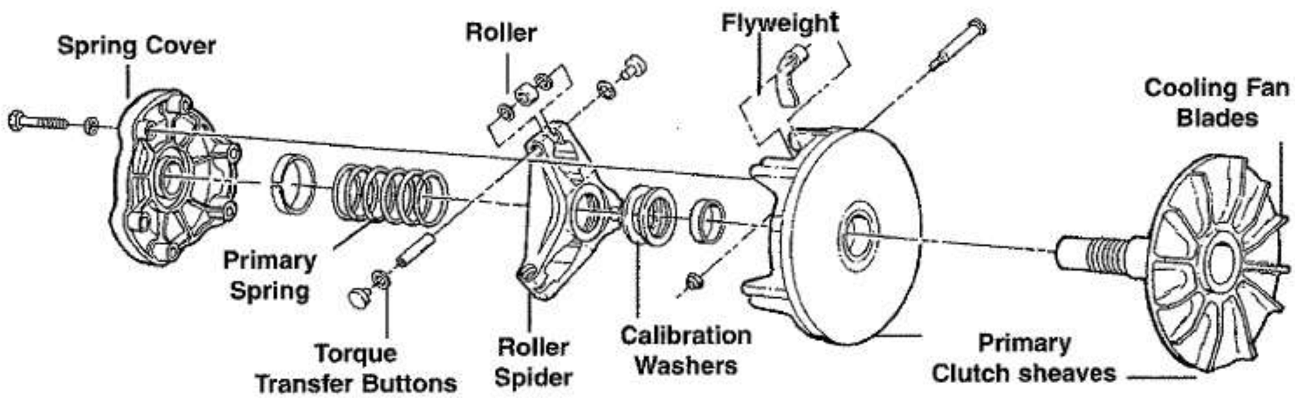
The simplicity of just a throttle and a brake for operation, with the power always at the right gear ratio without frantic clutching and shifting, appealed to a large segment of the booming ATV market. As Polaris conceded some points to the race crowd and introduced it's popular Predator gearbox version, the competition recognized the advantages of the automatic belt drive transmission, and several manufacturers introduced versions of their own.

One of the complaints about the early Polaris system was a lack of engine braking on down hills, and Polaris solved this problem with its EBS (Engine Braking System). On the Polaris engine braking system, the belt has longitudinal ribs on the inside drive cogs which match the ribs on the center hub of the primary clutch. One-way sprague clutches inside the hub transfer the belt drive to the motor during reverse pull, while still permitting freewheeling and a regular engagement speed during normal driving.

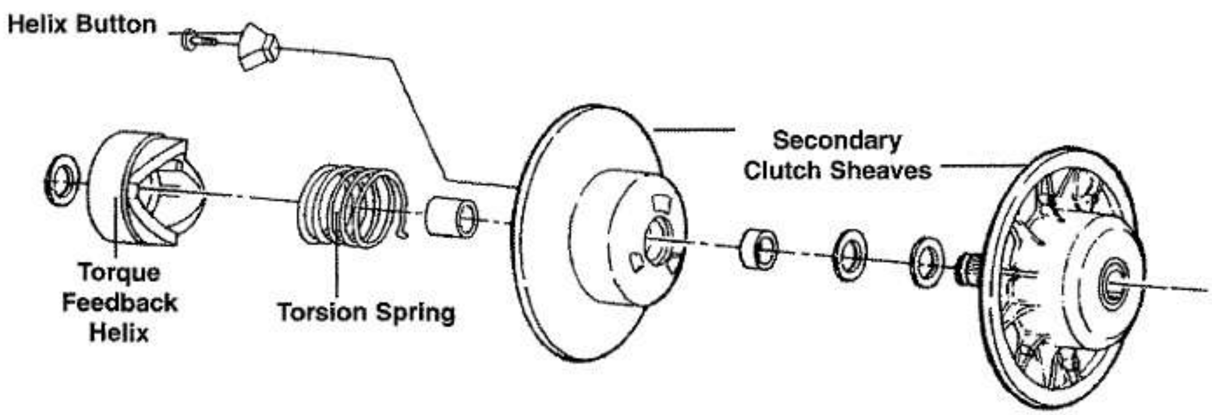
Another complaint about the belt drive transmission was a lag in the response when letting off the throttle and then hitting it again coming out of tight corners or balancing on the throttle up hills. This lag is due to friction between the drive components in the secondary clutch as the shifting action changes direction. Most belt transmissions use relatively inexpensive plastic sliding buttons riding against helix angles to provide side tension on the belt in order to transfer the torque of the engine. This was solved in snowmobiles by introducing rollers to eliminate any sliding friction. Because the rollers can change direction without getting stuck by friction, the response to up shifting or downshifting thru the ratios improved greatly. Not only did the lag in response disappear, but since the belt now was under constant pressure, belt slippage was reduced, the belt life tripled and the efficiency improved. Most production roller secondaries utilize plastic rollers, but there are also racing designs that go to the extra expense of using hardened billet steel helixes and ball bearing rollers in order to totally eliminate any friction within the rollers themselves.

Tunability is an important point with automatic transmissions as you change the power by adding pipes, carbs or other power boosters. You need to adjust the transmission to run at the correct engine speed where the power is produced. This is done by changing springs, flyweights and helixes, and the key is a good availability of these components. Polaris has a large selection of flyweights and springs from their 50 years of producing belt drive transmissions, as does Comet for their after-market primary clutch

Primary Clutch



Secondary Clutch



These exploded drawings show the typical components and tuning parts in an ATV beltdrive transmission. The main tuning parts in the primary are the primary spring, the flyweights and the spider calibration washers. The main tuning parts in the secondary are the torque feedback helix and the torsion spring. An extensive listing of these parts can be found in the Polaris ATV section.

for Polaris and Kawasaki models.

The newer belt drives from companies such as Kawasaki, Suzuki, Arctic and Can-Am-BRP, have improved on several problem areas. Better die-cast covers as part of the engine castings provides improved protection against water, although the downside is the much longer time it takes to remove them if you want to change a belt or work on the clutches. Several models also have the belt in constant contact with the pulleys to provide engine braking, while a separate slipper clutch takes care of the engagement function.

Ultimately, the performance of a belt drive system comes down to how well the manufacturer has reduced or eliminated all the possible friction areas. Availability of tuning parts is also important to compensate for changed power levels due to added after-market products, as well as conditions such as altitude or temperature changes due to seasonal variations.

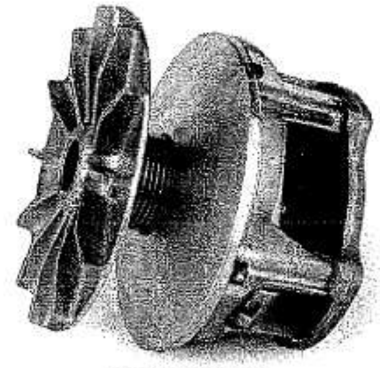
The belt drive transmission is here to stay, both because of its convenience and the constant power delivery to the driving wheels in mud or other difficult conditions. With more manufacturers committing resources to development, we are due to see considerable improvements in designs that will enhance both performance and durability.

POLARIS P-90 ATV CLUTCHES

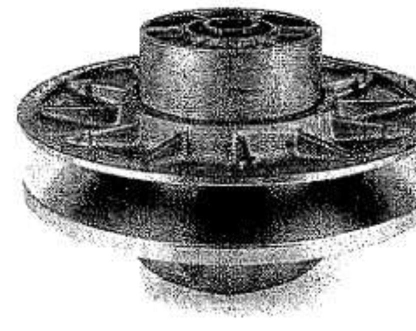
The P-90 primary clutch is a machined-down version of the P-85 snowmobile unit. The sheave diameter is smaller (7.25 in.) and the travel is 1.1" as compared with the 1.25" on the P-85. The belt is 1 3/16" wide, which is adequate to over 60 HP. The P-90 uses standard Polaris flyweights and springs, but the P-90 full shift numbers are lower because of the shorter travel. Polaris also has an EBS (engine braking system) primary. This unit has a larger ribbed center hub where the belt has matching ribs cut in the cogs. The belt is in constant tension with the hub, but the hub has a one way Sprague clutch in it that allows it to spin freely before the sheaves engage it. When the clutch shifts all the way down in low again for going down steep hills, the one way clutch engages and allows the belt to drive the engine and use it for braking. The EBS uses shorter flyweights to clear the hub.

The P-90 ATV secondary clutch is completely different from the P-85 snowmobile clutch. The P-90 is smaller in diameter (9.5") and the moveable sheave moves only on center bushings. The helix has a forward and reverse angle, and is mounted with spines instead of a key. With a double angle helix, the clutch can absorb reverse shock loads when the ATV comes off jumps, etc. The P-90 secondary unit uses standard Polaris springs, and selected Polaris cams are also available with teflon coating for improved back-shifting.

There is also a roller conversion kit available. The buttons in this kit are replaced by a double angle steel cam, and a cover containing 3 ball bearings replaces the helix. This kit is a direct bolt-on-- no machining is needed.

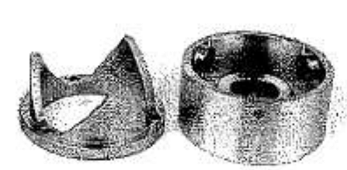


P-90 EBS Primary



P-90 Secondary

Aaen Teflon Coated Polaris Helix



Aaen "Roller Action" Kit

Polaris ATV Flyweights

Part No.	Desc.	Grams	Part No.	Desc.	Grams
5630295	Q-1mod	35.5	1321529	10MB	47.5B (HI-A)
5630292	K-1mod	38.5	5630514	G	48
5630144	K-1	39	5630418	C	50
5630280	16 mod	40	5630513	10MH	50.5B
5630279	16	43	5630095	S53	53
5630709	10RH	44B	5631214	20-54	54B
1321530	10MR	44B (HI-A)	5630509	S55	55
5630515	F	45 (HI-A)	5631215	20-56	56B
5630710	10WH	46B	5630581	S58	58
1321527	10MW	46B (HI-A)	5631216	20-58	58B
5630711	10BH	47B	B = Bushed HI-A = High Altitude		

Polaris Aluminum - Aaen Teflon ATV Helix

Polaris Part No.	Angle	Aaen Teflon Coated Part No.
5131164	34	------------
5130383	36.5	------------
5131162	38-36	------------
5131163	38-34	------------
5131446	40	ATC30013
5131161	40-36	------------
5132344	44-36	ATC30014
------------	48-40	ATC30015

Aaen "Roller Action" Steel Helix

Part No.	Angle
ATC30019	36
ATC30018	40
ATC30020	42-38
ATC30050	44-38
ATC30045	42-38 Enclosed
ATC30016: Roller Housing With Steel Ball Bearings	

Polaris ATV Primary Pressure Springs

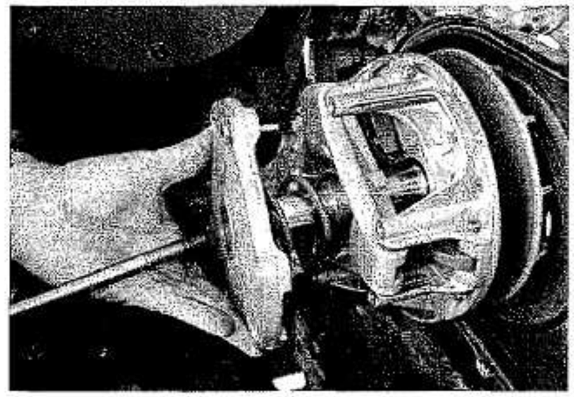
Part No.	Description	Engagement Load - lbs.	Rate lbs. / in.	Full Shift Load - lbs.
7041150	Red / White	100	92	192
AAS12420	Aaen Red	90	140	230
7041062	Silver	78	150	228
AAS12418	Aaen Green	60	112	172
7041061	Brown	50	106	180
7041132	White	35	76	111
7041157	Blue / Green	20	72	92

Polaris ATV Secondary Spring

Part No.	Description	Torsion Rate in. / lbs. - REV	Compression Rate in. / lbs.
AAS12410	Aaen Blue	308	45
AAS12417	Aaen White	360	47
7041646	Silver / Blue	286	28
7041501	Gold	247	23
7041198	Red	213	22

CHANGING TUNING COMPONENTS

Changing tuning components can usually be done quickly if you have the correct tools and know some short cuts. Here are some tips.



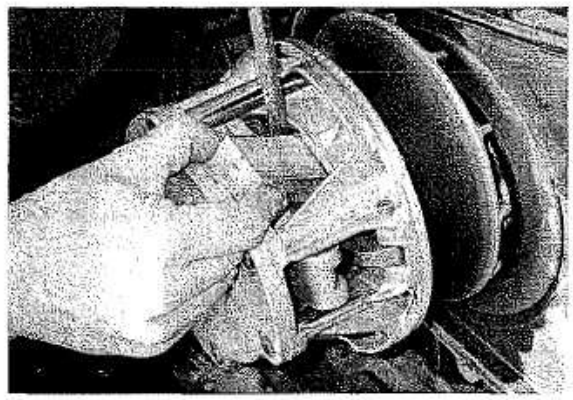
Removing primary pressure spring.

CHANGING PRIMARY PRESSURE SPRING

Remove the drive belt. Remove the center clutch mounting bolt. Remove the bolts holding the outside cover in place, and remove the cover. Change springs or add/remove spring washers. To get the cover back on against the spring pressure, use an extra large washer on the mounting bolt and screw the cover back in place; then install the cover screws. Remove the mounting bolt with the large washer, and install it again with the correct washer. Retighten to factory specs.

CHANGING PRIMARY FLYWEIGHTS

Remove the drive belt. Use a screwdriver or a lever between the spider and moveable sheave, and push the sheave away from the spider. Insert a 1/2 " block between the spider and sheave to hold them apart, or use a clip over the two sheaves. Use an allen wrench and an open end wrench to remove the flyweight bolt. Make sure all thrust washers are accounted for. Install new weights and the thrust washers and make sure you install the flyweight bolt against the direction of rotation of the clutch. (this puts the load on the bolt head and not the nut)

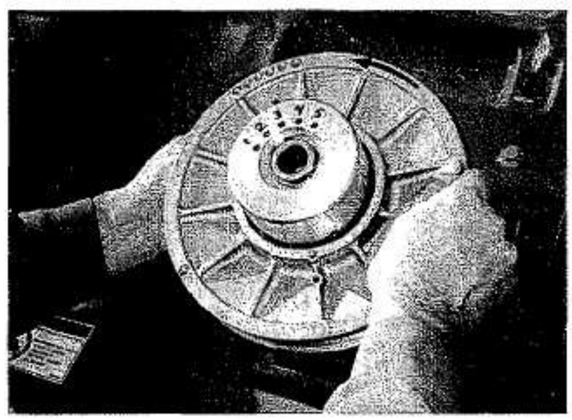


Insert block to move the spider away from the flyweight.

WARNING: Do not start the engine until the spacer block or clip has been removed, and the belt and beltguard installed, otherwise they will be flung out of the clutch and will cause damage to property and people. Stand on opposite side of the machine (away from the clutches) when the engine is started.

CHANGING SECONDARY TUNING COMPONENTS

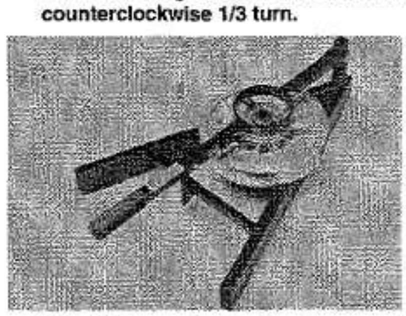
Removing the cam on Polaris and Skidoo clutches requires a snapring plier. Remove the secondary from the machine and place it on a table with the cam facing up. Rotate the moveable sheave counterclockwise and press the cam down while the snapring is removed. Release the pressure on the cam slowly to prevent it from flying off due to the spring pressure. On Arctic and Yamaha reverse cam designs, the cover/cam has to be removed by unscrewing the three tower nuts. To tighten up the torsional spring, look at the spring holes and rotate the spring tab counterclockwise to the next hole.



When installing helix, turn movable sheave counterclockwise 1/3 turn.

To install the spring and cam again, it is handy to use a special clutch service tool that allows you to push the cam back down with leverage, as some springs are very stiff.

When installing the cam and spring again, first make sure to wind the moveable sheave at least a quarter turn counterclockwise to make sure the helix cam towers are on the tension side of the buttons or rollers. Failure to do so will result in no pretension and a sluggish shift out. This is one of the most usual mistakes for beginners. Before installing the clutch, check the pretension by rotating the moveable sheave counterclockwise by hand or use a fish scale to measure the pretension.



This clutch service tool makes assembling the secondaries easier.

USE SAFETY GLASSES WHEN WORKING ON CLUTCHES

A QUICK TUNING SESSION

To do any intelligent clutch tuning, you need a tach. You are interested in two things; your "shift speed" and how it varies from low ratio to full shift out. You also need to know the "engagement speed," where the vehicle first starts moving.

There are four tuning components that affect the calibration of your transmission. In the primary clutch you can change the pressure spring and the flyweights. In the secondary, you can change the torsion spring and the helix cam. When tuning, only one component at a time should be changed, and the result noted.

The primary clutch components change the calibration in the following ways:

Pressure Spring: The pressure springs are listed by three properties: engagement load, full shift load, and rate. Engagement load is the load the flyweight has to overcome to engage the belt. The full shift load is the load the flyweight has to overcome at the end of the shift when the machine runs at top speed. The rate of the spring controls the shift RPM during the shift, and influences whether the RPM increases, decreases, or stays in the same (straight) during the shift.

The Flyweight: The flyweight has two properties used for tuning; the weight of the flyweight and the curvature. The weight determines how much RPM is needed to overcome the loads of the pressure spring and secondary. Light flyweights need higher RPM, heavier flyweights less RPM. The curvature determines the engagement speed and the aggressiveness of the acceleration. Flyweights are listed by weight in gram, curvature and engagement profile.

The secondary clutch components change the calibration in the following ways:

Torsion Spring: The torsion spring has a torsional rate and a compression rate. Depending on installed length and torsional load, the total spring load acting through the helix cam determines the shiftout point, low ratio and the shift curve. Increasing pretension load by moving the spring to different holes in the helix to give more angle pretension increases the RPM where the belt starts moving into the secondary sheave. The rate of the spring influences how the RPM changes during the shift. Higher rates will increase the RPM at top speed. Springs are usually listed by torsional rate.

Helix Cam: The helix cam has only one property we are interested in; the cam angle. Angles are listed either as straight or progressive. In most cases the cams start out with a large angle to give harder acceleration at the beginning of the shift. At the end of the shift the angle is reduced to keep the engine at a higher RPM. By changing the angles, the shift curve can be changed. Small angles give a higher shift speed, larger angles give a more aggressive shift out. Cams are listed by angles.

The influence of the tuning component on your tuning objective is listed below. Some of these changes may result in decreased efficiency and belt life.

CLUTCH COMPONENT INFLUENCE ON TUNING OBJECTIVE
CHANGE ONE COMPONENT AT A TIME

TUNING OBJECTIVE	PRIMARY CLUTCH		SECONDARY CLUTCH	
	PRESSURE SPRING	FLYWEIGHT	TORSION SPRING	HELIX CAM
Increase Shift Speed	Same Rate Higher Engagement Load Higher Full Shift Load	Lighter Flyweights	Same Rate More Pretension	Less Cam Angle
Decrease Shift Speed	Same Rate Less Engagement Load Less Full Shift Load	Heavier Flyweights	Same Rate Less Pretension	Larger Cam Angle
More RPM on Top End	More Rate Same Engagement Load	Less Aggressive Curvature	More Rate Same Pretension	Less Angle at End of Shift
Less RPM on Top End	Less Rate Same Engagement Load	More Aggressive Curvature	Less Rate Same Pretension	More Angle at End of Shift
More Aggressive Acceleration Less RPM at Beginning of Shift	More Rate Less Engagement Load Same Full Shift Load	More Aggressive Curvature	More Rate Less Pretension	More Angle at Start of Shift
Less Aggressive Acceleration. More RPM at Beginning of Shift	Less Rate More Engagement Load Same Full Shift Load.	Less Aggressive Curvature Grind Flat and Extend it Into Shift Curve	Same Rate Higher Pretension	Less Angle at Start of Shift
Increase Engagement Speed	Less Rate, Add Shim More Engagement Load Same Full Shift Load	Grind Flat or Notch	No Change	No Change

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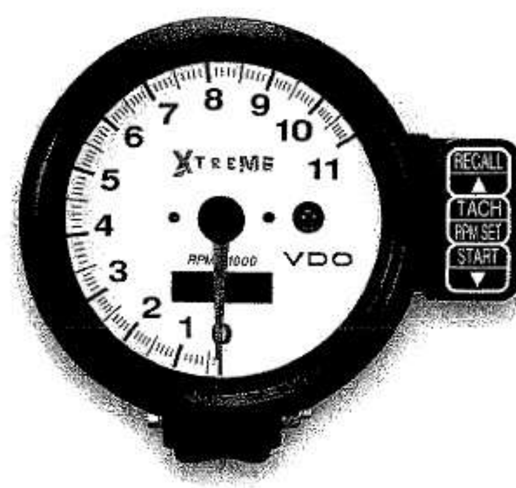
VDO XTREME TACH

Replays the RPM'S of your run in slow motion!

AVAILABLE WITH WHITE OR BLACK FACE. SPECIFY COLOR WHEN ORDERING.

Record your engine RPM during your run. Works on both drag strips and oval tracks. A built-in computer records engine RPM at two different speeds. You can replay the run directly on the dial at a third of the real speed or at 1:1 real time to catch any small variations in clutching. The tach has a 4" dial face and a red shift light which can be set to come on right below the engagement point to show you when the RPM is close for take-off. Two additional shift points can be programmed in to indicate shift speed or over-rev conditions. Store up to 4 runs. Specify year, make, and model.

Xtreme White TGT14804
Xtreme Black TGT14805
Battery TGT14801



Avenger I

The new Avenger I by Racepak records 1 RPM and 2 EGT temperatures, and displays them one at a time on a backlit LCD with .800" digits. You can replay 6 minutes of data or simply display peak readings during any run. The Avenger I can also be programmed for an alarm mode where a red light will flash when user programmed maximum limits are exceeded. Compact & waterproof, it fits in a standard 2 1/16 gauge cutout. Kit comes complete with 2 quick response EGT probes & remote switch.

Avenger I Kit TGT14823
Quick Response EGT . . TGT14824

Avenger II

Avenger III

Technology from the race track on a trail riders machine. Built-in user setable alarms on the EGT,max recalls, and 25 seconds of recording playback. Computer/printer download available. Waterproof, so it will not fog up in wet or changing weather conditions.Operates reliably down to -40° F. **(Probes sold separately--see below)**

Avenger II Base Unit (3 temperature inputs & 1 RPM) TGT14580
Avenger III Base Unit (4 temp inputs & 1 RPM) TGT14581
Exh. Temp. Probe TGT14579
2nd RPM Input TGT14582
Warning Lite TGT14583
Remote Switch TGT14584

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Mount this timer on your sled and record lap times in oval or snocross. With an extra beacon it is also perfect for drag racing and speed run tests. Only 4" x 2" x 1" in size, and operates on 9V and AA batteries. Provides 32 data memory with .01 sec. accuracy. Beacon is threaded for easy tripod mounting. High tech, vibration resistant circuit design. Used by professional auto, cycle, & snowmobile racers. Purchase extra beacons for split timing.

Oval & SnoX Timer TGT14830
Drag & Speedrun TGT14831
Extra Beacon TGT14832

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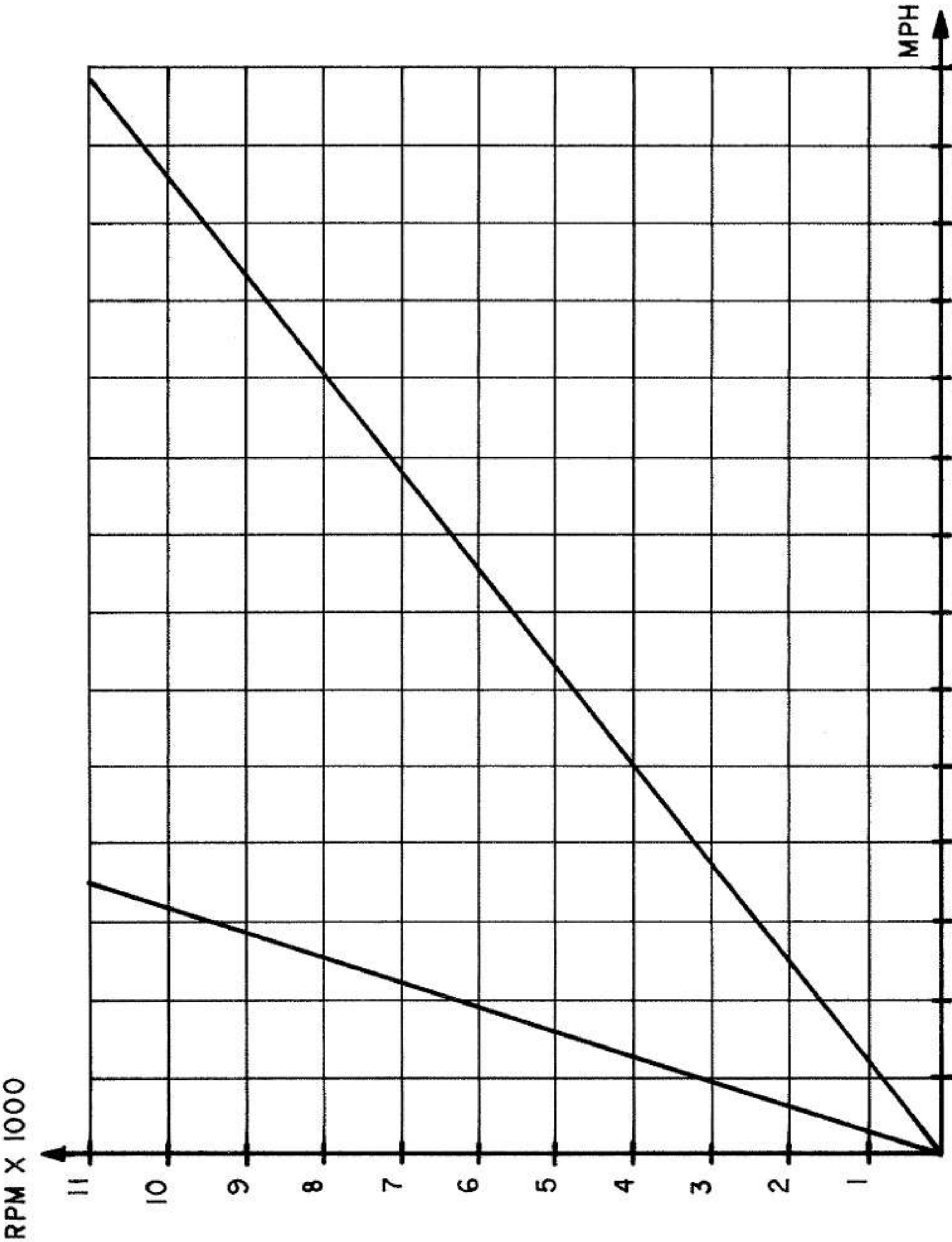
TRANSMISSION TUNING DATA

DATE: MACHINE: MODEL: SHEET NO:

TEST AREA: TEMP: SURFACE COND: TIMING EQUIP:

CENTER DISTANCE: GEARING: DRIVER WEIGHT: CLUTCH TYPE:

	ITEMS	TEST 1	TEST 2	TEST 3	TEST 4	TEST 5	TEST 6
DRIVEN DATA	DRIVEN RAMP ANGLE						
	SPRING						
	PRETENSION SETTING						
	PRETENSION PULL						
	WASHERS						
DRIVER DATA	DRIVER FLYWEIGHT NO.						
	WEIGHT						
	CURVATURE						
	SPIDER SPACERS						
	SPRING						
	SPRING SPACERS						
BELT DATA	BELT BRAND						
	LENGTH						
	WIDTH						
SHIFT DATA	SHIFT INFO ENGINE SPEED						
	SHIFT SPEED						
	OVER-RUN IN LOW						
	SHIFT SPEED INCREASE						
	SHIFT SPEED DECREASE						
	OVER-RUN IN HIGH						
	AVERAGE TIME FOR RUNS						
	COMMENTS						


RPM X 1000
11
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130
140

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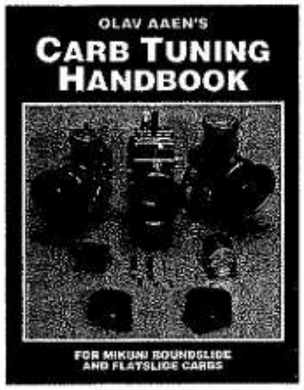

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Our updated Carb Tuning Handbook covers the basics in Carb Tuning, testing and racing. Mikunis Roundslide Carbs, Flatslide Carbs and the new TMX are discussed in detail. A must for those who want to learn to get the most out of their engine. Learn how to diagnose your carbs performance, use correct testing procedures and calibrate with the right tools. B-16002

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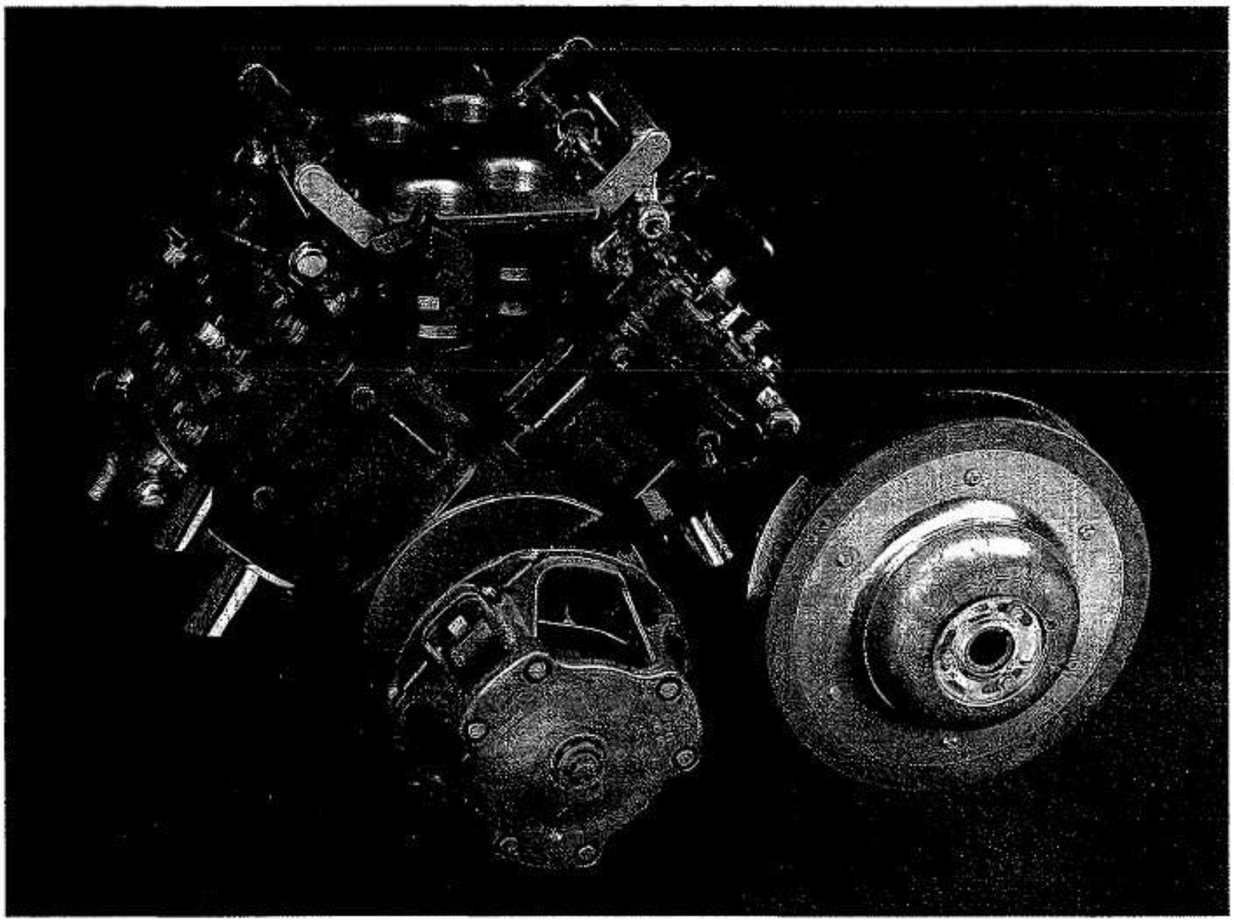
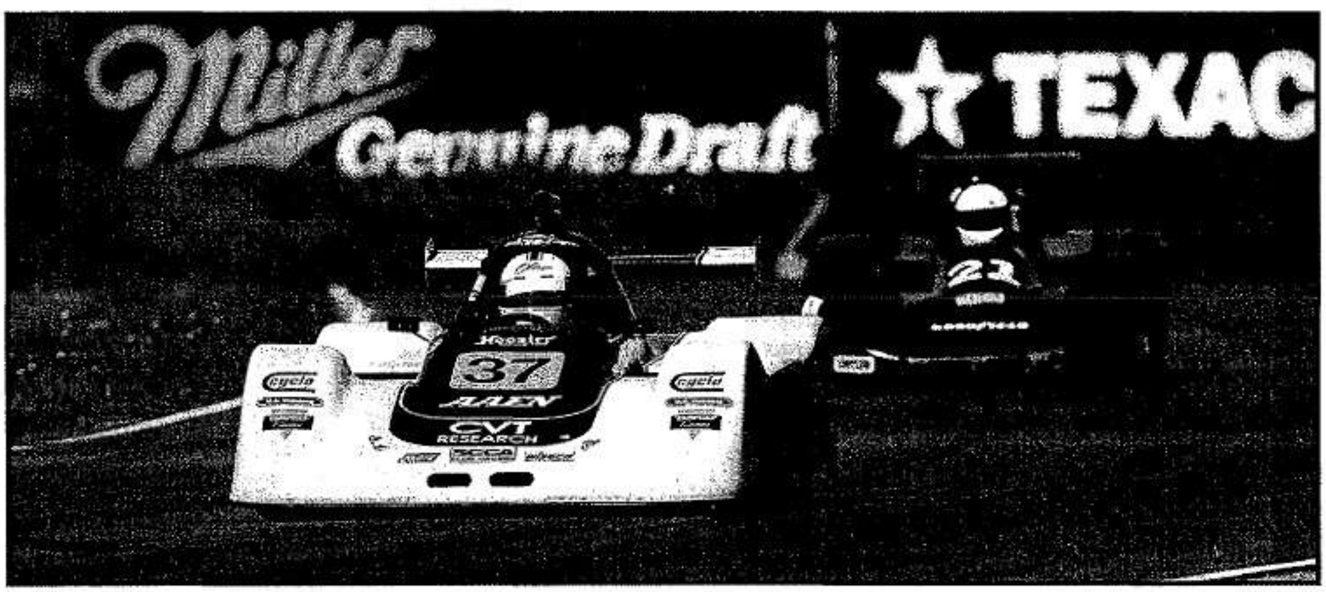
CVT Research





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