Variator loading and control in a V-belt type geared neutral transmission in and around the geared neutral point.

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Abstract

Geared neutral transmissions still hold the promise of unparalleled transmission properties. The mechanism eliminates the need for a starting device and auxiliary clutches and gears for reverse. A very good launch feel without the use of a torque converter should be one of the main benefits. When combined with two clutches, resulting in a two-stage split-torque geared neutral transmission, the system can be augmented such that very high overdrive ratio's can be obtained, allowing low engine speeds at intermediate engine power levels to reduce fuel consumption. The mode shift can be carried out with a synchronised clutch.

One of the main points of concern is the loading of the V-belt-type variator in and around the geared neutral point. It is the purpose of this paper to address this issue in some detail, taking both slip and torque loss into account. Variator control and variator loading in and around the geared neutral point will be addressed.

Geared Neutral Transmission Overview

History and Introduction

Efficiency improvement has been the main reason for developing split-path transmissions based on hydrostatic variators. By transmitting part of the engine power through conventional gears, the load on the variator is reduced, and thereby its parasitic losses. With increasing engine torque levels for passenger cars, the main reason for developing split-path V-belt type CVT's is improvement of CVT's torque capacity [1]. For this case, partial unloading of the variator creates room for higher transmission input torque with comparable endurance life expectation. The price for this increased torque capacity is the reduced ratio range. Usually this is solved by adding two clutches, allowing for a direct LOW mode, significantly extending ratio range at the cost of an, albeit synchronous, shift.

Opposite to this development, there has been significant interest in so-called Infinitely Variable Transmissions (IVT's), comprising a split path in low-mode and a direct path in high mode. The most interesting feature of this type of transmission is that it allows for a so-called Geared Neutral state. In this state, a running engine can be connected to drive shafts of a vehicle standing still, without the use of a slipping clutch. The property of an epicyclic gear to subtract two input angular speeds to generate an output speed is used. By changing one of the input speeds, the output speed starts to deviate from zero, causing the vehicle to take-off.

Nowadays, probably the best known IVT is developed by Torotrak [2] and is based on a full toroidal variator. Another, more recent development, is based on a half toroidal variator [3]. An extensive research study on an IVT based on the PIV-chain variator, called STGN, was reported on in the early nineties [4], [5].

The main advatages and disadvantages on system level over conventional CVT drivelines with a torque converter (TC) or with conventional wet plate clutches are:

- No separate starting device needed.
- No Drive-Neutral-Reverse set required.
- Extra fixed gear set or chain, as well as epicyclic gear required.

- Two clutches required to extend ratio range. Clutches only actuated at or near synchronisation.

Some other characteristics will be covered in the remainder of this paper are running variator under all conditions; lower clamping pressure levels when compared to TC variant; variator not in extreme ratio at take off; better use of hydraulics system; better launch performance.

Since this present paper is concerned with the geared neutral state obtained with V-belt type variators, the layout and transmission parameters of the STGN were taken as reference with minor modifications. Characteristics of the interesting mode shift are beyond the scope of this paper.

Layout

A typical layout of a driveline containing a geared neutral transmission is presented in Fig. 1.

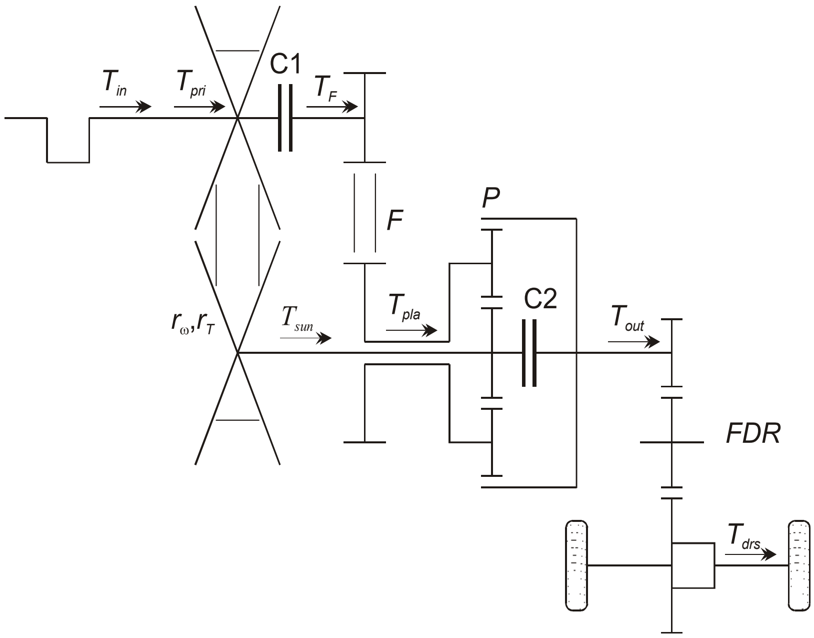

Figure 1. Schematic representation of a typical Geared Neutral Transmission.

The sun of the epicyclic gear is connected to the crankshaft via the V-belt type variator. The planet carrier is also connected to the crankshaft via a fixed-ratio mechanical path and clutch C1. The annulus drives the final drive reduction and the differential gear. By closing clutch C2 and opening C1, the epicyclic gear behaves as a direct link between secondary shaft and output and the transmission behaves like a classical CVT. Within the scope of this paper, C1 is assumed to be closed or slipping and C2 is assumed to be open, since this is the relevant case for geared neutral behaviour. The following parameters were taken directly from [4]:

$P = 2.5$	Ratio of annulus to sun radius of epicyclic gear
$F = 0.44$	Fixed chain ratio
$FDR = 1/4.11$	Final Drive Ratio
$r_{low} = 0.438$	Variator Low ratio
$r_{OD} = 2.28$	Variator OverDrive ratio

Speed, torque, power relationships

Kinematics

Following from the kinematics of the epicyclic gear: $\omega_{sun} = \omega_{pla} + P(\omega_{pla} - \omega_{out})$, with C1 closed in and output speeds are related according: $\omega_{out} = \omega_{in}\frac{1}{P}\left[(1+P)F - r_{\omega}\right]$.

By substituting $r_{GN} \equiv (1+P)F$ we arrive at

$$\omega_{out} = \omega_{in}\frac{1}{P}\left[r_{GN} - r_{\omega}\right]. \tag{1}$$

Here $r_{\omega} = \omega_{sun}/\omega_{in}$ equals the variator speed ratio. The planetary ratio P is defined as the ratio of the ring to sun radii and $F = \omega_{pla}/\omega_{in}$ is the fixed path speed ratio.

Obviously, the output speed equals zero when $r_{\omega} = r_{GN}$.

Torque, power and efficiency relationships

For the following all losses are assumed to originate in the variator.

On noting that $T_{in} = T_{pri} + T_F$, and $T_{sun} : T_{pla} : T_{out} = -1 : (1+P) : P$, a relation between in and output torque can be derived:

$$T_{in} = T_{out}\frac{1}{P}\left[r_{GN} - r_T\right]. \tag{2}$$

Here, r_T denotes the torque ratio of the variator: $r_T = T_{pri}/T_{sun}$.

The relationship

$$T_{out} = -PT_{sun} = -\frac{P}{r_T}T_{pri} = -\frac{P}{r_{GN}}T_F \tag{3}$$

shows that for a positive output torque, T_{pri} should be negative, thus leading to negative power flowing from variator primary to secondary shaft. Therefore, the efficiency of the variator should be defined by $\eta_{\mathrm{var}} = \frac{\omega_{pri}T_{pri}}{\omega_{sun}T_{sun}} = \frac{r_T}{r_{\omega}}$.

In the geared neutral state, the output power is zero, but an output stall torque T_{out} may be present. The circulating power, loading both chain and variator, is given by

$$P_{circGN} = T_{out} \frac{r_{GN}}{P} \omega_{in} . \tag{4}$$

The engine supplies the power that is lost inside the GNT which is equal to

$$P_{lossGN} = \frac{1}{P} \left(r_{GN} - r_T \right) T_{out} \omega_{in} . \tag{5}$$

The efficiency of the GNT is given by: $\eta = \dfrac{\omega_{out} T_{out}}{\omega_{in} T_{in}} = \dfrac{r_{GN} - r_{\omega}}{r_{GN} - r_T}$.

Driveline and Transmission control

In a vehicle stall condition (full brake and accelerator pedal), control of output torque cannot be accomplished by control of engine speed as is customary with conventional drivelines comprising a TC. The relationship between TC-pump speed and pump torque: $T_{pump} = k\omega_{pump}^2$, typical for hydrodynamic couplings, does not exist in the present case: the input speed is a completely free variable in geared neutral.
Engine torque is related to output torque, but depends strongly on variator efficiency, thereby making it unsuitable for control.

Control of output speed to zero by variator speed ratio control will result in small ratio perturbations causing a constant switching between forward and reverse ratios, leading to a "nervous neutral" condition.

Quite a different approach, already mentioned by some authors [4] and [6], results from eq (3) showing a linear relation between T_{out} and either T_F or T_{sun}. This means that each of them would be candidate for output torque control in and around stall condition. In case the transmission contains a sensor for measuring either of these torques, output torque control can be carried out in the conventional way. In most cases, such a sensor is not present. In the following sections a control technique is proposed which relies on the specific friction characteristics of a wet plate clutch or the V-belt type variator.

In the geared neutral point, the engine is only loaded by the transmission losses. Opening the throttle will quickly lead to maximum engine speed, quite opposite to the case with a TC, where the engine will speed up to stall speed. Therefore, the engine speed should be controlled independently.

Clutch Slip Control

Controlling T_F can be accomplished by controlling the transfered torque in the slipping clutch C1. For a wet plate clutch this would be accomplished by applying a fixed pressure onto the cylinder actuating the clutch, corresponding to the torque T_F that is to be transmitted. By selecting clutch slip speed as reference variable for ratio control of the variator, the variator ratio is changed, such that the clutch starts slipping with a controlled slip speed. The slip setpoint should be large enough (Sp1 in Fig. 2a) to avoid the situation that slip control error may cause a zero-slip condition, which will result in unacceptable torque error (Sp2 in Fig. 2a).

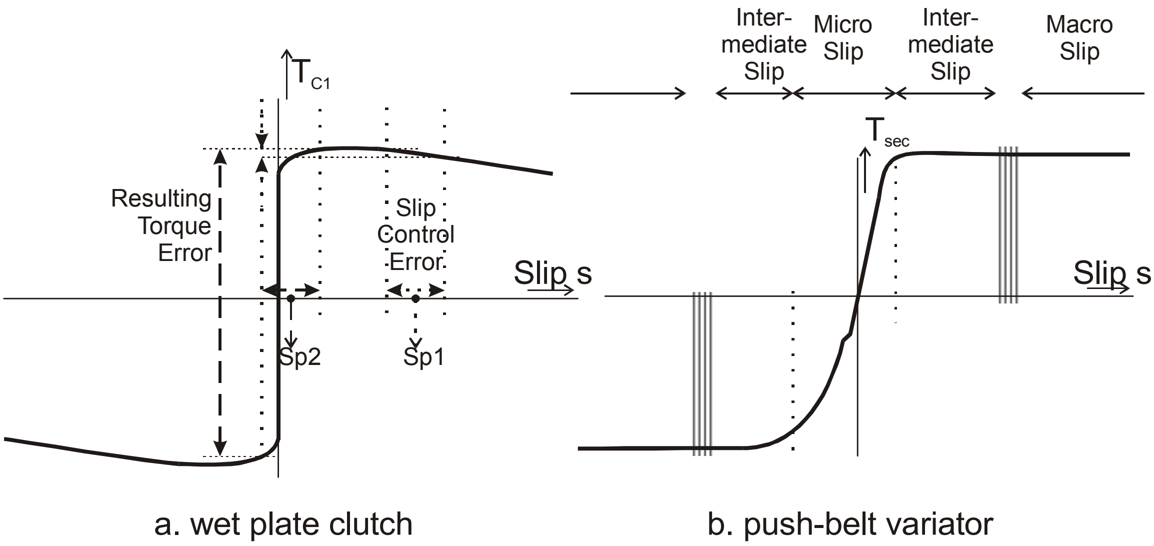

Figure 2. Typical traction curves.

On the other hand, large values for the slip speed setpoint will diminish transmission efficiency and increase the cooling demand of the clutch. Note that an accurate measurement of clutch slip is a prerequisite for this approach.

Variator Slip Control

Controlling T_{sun} can be carried out along similar lines as described in the previous section but differs in some important aspects. The transfered torque in the variator can be controlled by the minimum clamping force (assuming torque independent clamping force control, see section below), by using the following relation:

$$T_{sun}(s) = \frac{2F_{clamp}\mu(s)R_{\sec}}{\cos(\lambda)}. \tag{6}$$

Here $R_{\sec}$ denotes the secondary belt radius, F_{clamp} the lowest of the primary and secondary forces applied to the variator and λ the pulley sheave angle. The slip dependent friction coefficient $\mu(s)$, is frequently referred to as the traction curve.

Unfortunately, variator slip s cannot be defined as easy as clutch slip. A practical way of defining slip s is based on the change of speed ratio upon loading of the variator, while leaving all other variables unchanged: $s \equiv \frac{r_{\omega 0}}{r_{\omega}} - 1$. Here, $r_{\omega 0}$ equals the variator speed ratio at zero variator output torque. A torque curve, valid for a push belt variator and schematically reproduced from [7], is shown in Fig. 2b. By definition, slip is zero at zero variator output torque. The micro slip region can be defined as the region where the torque curve shows approximately linear behaviour and ends where the torque curve shows a clear flattening tendency. The intermediate slip region should be defined as the region beyond the micro slip region where no damage is incurred to either V-belt or pulleys, even after prolonged running. The mere existence of such a region is not yet established. Macro slip occurs beyond the intermediate slip region.

Measuring s is considerably more difficult thn measuring slip of a clutch, since it requires the determination of both r_{ω} and $r_{\omega 0}$. The former, being the ratio of secondary and primary speeds, is rather easy to establish, whereas the latter can only be derived from geometric data and depends primarily on geometric ratio, but also on clamping force level and input speed. Additionally, wear occurring in the variator and temperature changes will have some influence on $r_{\omega 0}$. A rather accurate (<1 %) measurement of $r_{\omega 0}$ is necessary in order to avoid the need for large slip setpoints as discussed in the previous section.

Variator Slip Control Scheme

The control scheme to be used for control of T_{sun} is depicted in Fig. 3.

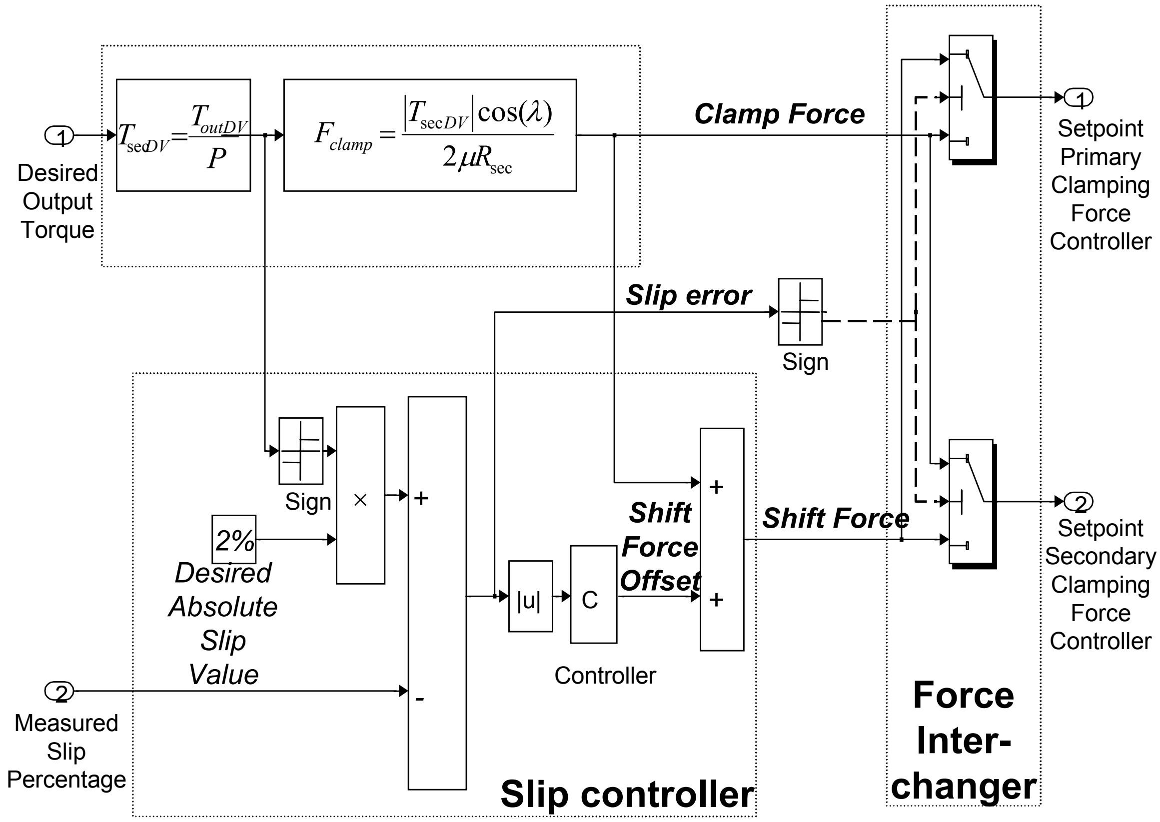

Figure 3. Variator Control Scheme

Prerequisites for this control scheme are torque independent clamping force control (see section below) and availability of a slip signal.

In Fig. 3, reference input for clamping force control is not actual torque, but the desired output torque as it results from the accelerator pedal interpreter. This value is then used for the calculation of clamping force that would just be adequate for transfer of this torque. It is the purpose of the slip controller to obtain variator slip within the Intermediate Slip Region (see Fig. 2). This is realised by shifting the variator. Therefore the shift force, which is the force that causes the variator to change ratio, should become larger when the slip error becomes larger. This is indicated in Fig. 3 by the P-controller. The sign of desired slip is given by the sign of the desired secondary torque, thereby selecting the positive or negative branch in Fig. 2.

The sign of the slip error determines whether an up- or downshift should be made, in order to reduce slip error. Since the shift force should always be positive, so that the clamp force is always the lowest of the primary and secondary actuating forces on the variator, the only way to allow for up- and downshifts is to interchange clamp and chift forces between primary and secondary pulley, depending on the sign of the slip error signal.

Controlling variator slip has several interesting implications. First, and probably most important, is that overclamping is avoided. This means optimal efficiency, as long as slip is small. Additional measurements should be carried out to establish an optimum value for slip speed. Overclamping (often at least 30%) is usually applied, to account for errors in the estimation of variator input torque and friction coefficient. This requires larger piston areas or higher pump pressures than for the case that variator slip control is applied. Due to the small slope of the torque curve in the Intermediate Slip Region, the slip measurement, mentioned above, is insensitive to small offsets.

Contrary to common CVT usage, output torque is controlled by the variator and transmission input speed is controlled by the engine.

An important property of this type of control is that slip is controlled by means of shifting the variator. Therefore, slip can only be avoided as long as the variator is able to shift sufficiently fast. It has been shown [8] that variator shift dynamics is greatly improved when the variator is at the verge of slipping. This will aid in avoiding macro slip, even in cases where fast ratio changes are required.

Torque independent versus torque dependent clamping force control

By Torque Independent clamping force control is meant that the lowest of the primary and secondary actuating forces, which is responsible for the maximum torque that can be transferred, can be controlled, irrespective of the existing torque level. Usually, this is realised by means of a closed control loop with clamping pressure as measured variable. The setpoint for this pressure is derived by means of eq. (6), where the torque level is usually derived from engine controller data. An extra pressure reserve of at least 30 [%] is used in order to avoid variator slip under all conditions. Torque independent clamping force control is a prerequisite for the proposed control system to work. Note that the proposed

system does not require an inpendent measurement of torque in the transmission.
Torque dependent clamping force control, on the other hand, is usually realised by means of a hydromechanical torque sensor and does not allow for independent setting of the clamping force level. In order to be applicable in a geared neutral transmission, a control loop can be constructed, with the realised torque sensor pressure as a torque signal, see [5]. It must be noted, however, that for this case the intermediate slip region cannot be used, since the technology of a torque sensor relies fundamentally upon operation in the micro-slip region.

Transmission properties

Variator loading under maximum stall and creep condition

In the geared neutral state all power from engine to transmission is used for compensating the losses in the transmission. This is comparable to a TC equipped vehicle at rest, where all engine power is dissipated in pumping losses in the TC. For the latter case, the available torque amplification equals $\frac{T_{out}}{T_{eng_{stall}}} = \frac{\mu_{TC}}{r_{low}} \approx 5 \sim 6$ with μ_{TC} denoting the stall torque multiplication of typically 2~2.5. $T_{eng_{stall}}$ represents maximum engine stall torque. For vehicles equipped with a CVT and a TC, however, the actual maximum torque amplification is limited by engine management to values close to $\frac{T_{out}}{T_{eng_{stall}}} = \frac{1}{r_{low}} \approx 2.5$, in order to protect the variator from overloading.

The equivalent torque multiplication for the GNT driveline is given by:
$\frac{T_{out}}{T_{eng_{stall}}} = \frac{P}{r_{GN} - r_T} = \frac{P}{r_{GN}\left(1-\eta_{\mathrm{var}}\right)} \approx 15 \sim 30$. This last value obviously depends strongly on the variator efficiency. Also for this case the torque amplification must be limited, in order to avoid variator overload. Using $T_{out} = -PT_{sun}$ and setting T_{sun} equal to the maximum torque allowed on the variator at r_{GN}, a torque amplification of about $\frac{T_{out}}{T_{eng_{stall}}} \approx 3 \sim 3.5$ can be achieved. The situation at the variator has two main advantages over the conventional case with TC: 1). the variator is running, avoiding stick-slip conditions at variator stand-still, and 2). the V-belt runs at a larger pitch radius, allowing higher torque at comparable clamping

force level. It has to be noted, however, that the variator is transferring considerable circulating power in this stalling condition, see eq (4). By controlling the input speed, the circulating power can still be at acceptable levels. Contrary to the case with the TC, the engine speed is a free variable in the stalling condition, as long as the engine is able to compensate for the losses. The duration of maximum stall is mainly limited by the cooling capacity of the transmission and is expected to be longer than that with a TC, since the dissipated heat is much smaller.

Launch

When the brake pedal is released after a stalling condition, the vehicle will take off, thereby causing the variator speed ratio r_ω to differ from r_{GN}. At launch, engine speed can be regarded constant, and road load resistance can be neglected. Under these conditions, the vehicle acceleration is given by: $a_{veh} = \dot{\omega}_{ds} R_{wheel}$ where $\dot{\omega}_{ds}$ denotes angular acceleration of drive shaft and R_{wheel} is the dynamic wheel radius. Following from eq. (1), vehicle acceleration at launch is given by $a_{veh} = -\dfrac{R_{wheel} FDR}{P} \omega_{pri} \dot{r}_\omega$. This equation shows that launch acceleration is determined by input speed ω_{pri} and by variator ratio change speed $\dot{r}_\omega$. Engine torque does not play a role during launch, but engine speed does, contrary to the stall condition. For most practical situations $|\dot{r}_\omega|$ is limited to about 0.5 to 0.7, which leads to maximum acceleration: $a_{veh\max} =$ 8.6 $[m/\sec^2]$ at 4000 [rpm] input speed. Note that also for this case the torque limitation of the variator has to be taken into account.

Conclusions

- A geared neutral transmission based on a V-belt type variator, calls for unconventional variator control techniques. In case no torque sensor is present, a slipping component, either clutch or variator, may be used for output torque control.
- A novel variator control scheme is proposed, in which slip is not controlled by means of clamping force, but by means of adapting variator ratio.
- The proposed variator control scheme allows for lower demands on the hydraulics system as compared to conventional control methods.
- With a geared neutral transmission a better launch performance can be achieved than with a conventional CVT with TC, without overloading the variator.
- Practical feasibitity of the proposed control scheme and validity of the associated assumptions should be proven by further research.

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