

Toothed Chain CVT: Opportunities and Challenges

Chengwu Duan, Jian Yao, and Ying Huang
General Motors LLC

ABSTRACT

A toothed chain continuously variable transmission concept is studied. By designing positive engagement at top overdrive ratio, we explored the potential to improve CVT mechanical efficiency. The low cost solution could improve fuel economy by 0.7% in FTP composite cycle. Preliminary multi-body dynamic simulation is also completed using VL-Motion to concept-proof the technical feasibility of disengagement and engagement. To address the noise issue resulted from abandoning the random pitch design in production chain, we proposed an alternate chain pitch sequence but more experimental data is required to validate the design.

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INTRODUCTION

By enabling engine to operate under sweet spots, friction-based continuously variable transmission (CVT) carries fuel economy advantage over positive engagement based transmissions like AT and DCT, although the latter ones have better mechanical efficiency in general. Several studies have been done to combine the merit of positive engagement and continuously variable ratio functionality [1-2]. Recently, a toothed chain CVT concept emerged as a low cost solution. Fixed teeth are conceivably added to the pulleys to mate with the fixed teeth on a chain, as shown in Figure 1. The tooth profile can be designed accurately to avoid gear lash and to achieve smooth torque transfer. The CVT continuously ratio changing functionality is maintained except at top overdrive ratios for which positive engagement is designed on the output side. When CVT operates in positive engagement ratios, its mechanical efficiency can be improved due to reduced clamp pressure and pump loss. This forms the motivation of this study.

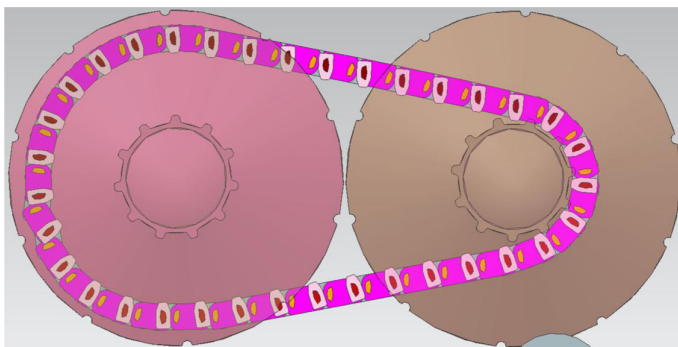


Figure 1. Schematic of TC-CVT

EFFICIENCY IMPROVEMENT AND FUEL ECONOMY BENEFIT

For a conventional friction-based CVT, the secondary pulley clamp force is used to prevent chain from slipping on pulley and the primary pulley clamp force is to maintain the ratio [4]. The ratio between primary and secondary clamp force is defined as $KpKs$ [4], which is greater than 1.0 at overdrive condition. As in equation (1), the actual clamp force on the secondary pulley F_s includes a safety factor SF .

$$F_s = SF \cdot F_{clamp}^* \quad (1)$$

The theoretical clamp force F_{clamp}^* is defined in the absence of dynamics.

$$F_{clamp}^* = \frac{T_p \cos \beta}{2r_p \mu} = \frac{T_s \cos \beta}{2r_s \mu} \quad (2)$$

Here, T_p and T_s are the input torque at primary pulley and output torque at the secondary pulley respectively; r_p and r_s are the chain pitch radius on primary pulley and secondary pulley; β is the pulley nominal half angle and μ is the theoretical friction coefficient between chain and pulley. Further, the primary clamp force F_p is as below:

$$F_p = KpKs \cdot F_s = KpKs \cdot SF \cdot F_{clamp}^* \quad (3)$$

Study at LuK has indicated that CVT efficiency can be improved by reducing clamp force [5]. Also, our experimental study of chain CVT over-clamping effect showed that efficiency was improved by 2.5% if the clamp force was reduced by a factor of 1.5, as in Figure 2.

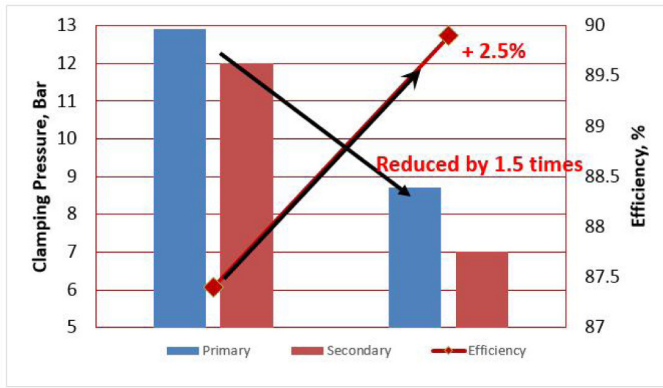


Figure 2. Experimental Over-Clamping Study (1000 rpm 100 Nm, Ratio = 0.7)

However, the clamp force scenario for the final design of Figure 1 is different from conventional friction drive CVT at the operating point of top overdrive ratio. Because the secondary pulley transmits torque through positive engagement, there is no possibility of slipping between chain and pulley and F_s could be as small as possible. On the other hand, CVT should maintain the top drive ratio if there is no slip between chain and primary pulley. So, the required primary clamp force F_p' should be equal to original F_s .

$$F_p' = SF \cdot F_{clamp}^* \quad (4)$$

Compare equation (3) and (4), the primary clamp force is reduced by a factor of $KpKs$, which varies from 1.5 to 2.0 depending on different designs [4]. Further, it is reported that for conventional friction CVT, there is around 0.5%-1% slip between chain and pulley at steady-state condition [6]. Although power loss should include both slip loss and torque loss, we can still assume approximately 0.5%-1% efficiency loss given the fact that torque loss is small thanks to advanced bearing design and low viscosity CVT fluid. The positive engagement shall reduce the slip to 0 at the secondary pulley, which would eliminate around 0.5% efficiency loss. In total, there could be 3-4% efficiency gain at the top overdrive ratio.

To validate our theory, multi-body dynamic simulation is done using VL-Motion. At first, study is conducted for a friction drive CVT. We try to maintain the secondary clamp force close to F_{clamp}^* and gradually reduce F_p until a major slip (50 rpm) occurs at secondary pulley. As shown in Figure 3, when slip speed is at 50 rpm, the ratio between F_p and F_s , i.e. $KpKs = 1.9$, which fall into the reasonable range of overdrive condition force ratio. Secondly, the positive engagement design of Figure 1 is simulated in a similar manner. The difference from friction CVT is that as we reduce the primary clamp force, we will see a major slip at primary side. As seen in Figure 4, taking 50 rpm as a threshold, the required F_p' is 12599 N while the F_s' is as low as 1000 N. Compared to friction CVT, $F_p / F_p' = 1.75$, which is close to $KpKs$.

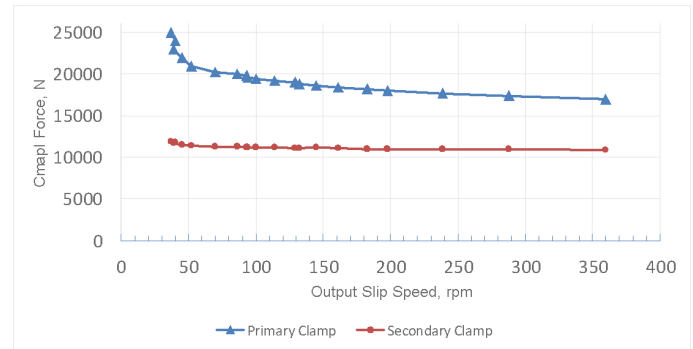


Figure 3. Simulation Study for Friction Drive CVT with Input Torque 150 Nm and Output Speed 2250 rpm

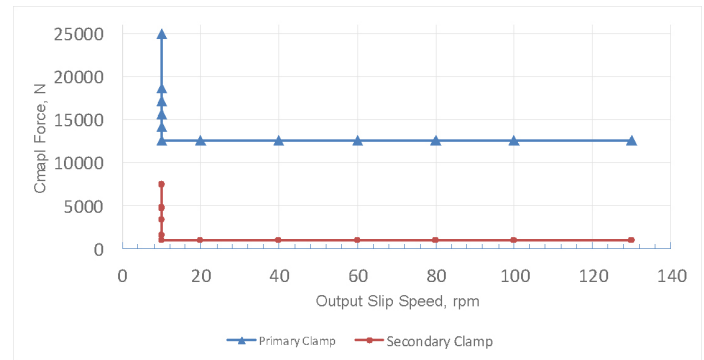


Figure 4. Simulation Study for Positive Engaged CVT with Input Torque 150 Nm and Output Speed 2250 rpm

As discussed earlier, TC-CVT shows efficiency advantage only at max OD ratio or top gear. When evaluating its benefit within the context of driving cycles, for example FTP, the vehicle-level fuel economy will be highly dependent on the amount of time spent at max OD ratio, and is very sensitive to the CVT ratio map calibration. A straightforward approach is employed by modifying the CVT ratio map to allow around 50% of time spent on top gear ratio for the evaluation. TC-CVT is simulated under FTP cycle and compared to the baseline assuming 4% efficiency improvement at max OD ratio due to the positive engagement.

Table 1. Fuel Economy Benefit Evaluation in FTP Cycle

Fuel economy improvement	City	Highway	Composite
4% at max OD	0.2%	1.6%	0.7%

As shown in Table 1, the fuel economy benefit mainly manifests in highway condition when vehicle is cruising at top gear. The FTP composite cycle fuel economy is 0.7% if 4% efficiency improvement at max OD ratio is assumed.