

Performance Characterization of a Triple Input Clutch, Layshaft Automatic Transmission Using Energy Analysis

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ABSTRACT

This paper details the design and operating attributes of a triple input clutch, layshaft automatic transmission (TCT) with a torque converter in a rear wheel drive passenger vehicle. The objectives of the TCT design are to reduce fuel consumption while increasing acceleration performance through the design of the gearing arrangement, shift actuation system and selection of gear ratios and progression. A systematic comparison of an 8-speed TCT design is made against a hypothetical 8-speed planetary automatic transmission (AT) with torque converter using an energy analysis model based upon empirical data and first principles of vehicle-powertrain systems. It was found that the 8-speed TCT design has the potential to provide an approximate 3% reduction in fuel consumption, a 3% decrease in 0-100 kph time and 30% reduction in energy loss relative to a comparable 8-speed planetary AT with an idealized logarithmic ratio progression.

CITATION: Robinette, D., "Performance Characterization of a Triple Input Clutch, Layshaft Automatic Transmission Using Energy Analysis," *SAE Int. J. Engines* 6(4):2013, doi:10.4271/2013-01-9042.

INTRODUCTION

Automatic transmission technology for passenger, light duty and heavy duty vehicles continues to evolve to meet increasingly stringent greenhouse gas (GHG) emission regulations, see [1] for example. As with any transmission system, a balance between size, mass, cost, number of gear states, number of shifting elements, type of launch device and the requirements of the application for acceleration, fuel economy and shift quality must be made. A report prepared for the EPA, [2], indicated the incremental cost and performance improvements for AT technology to transition from 6-speeds to 8-speeds as a key enabler to reducing GHG emissions. An approximate 2 to 6% reduction in CO₂ emissions was the reported benefit in [3] by implementing leading edge transmission technologies. Over the past decade, the number of fixed gears states for step gear planetary ATs have steadily increased, with 6-speeds common place and 8-speeds gaining more market volume, [4 and 5], demonstrating the reported incremental improvements in performance. Six and seven speed dual clutch transmissions (DCT's) have emerged in the past few years as an alternative type of AT due to their favorable energy usage and ratio tuning potential over planetary type AT, [2, 6, 7 and 8]. A search for relevant intellectual property reveals a multitude of unique planetary and multi-input shaft parallel axis gear transmission

powerflows with 8 or more gears states, utilizing 5 or more clutching elements, see [9, 10, 11, 12, 13, 14, 15, 16] as representative examples. The aim of these designs are to leverage the kinematic design of the gearbox to optimize gear ratios and progression while minimize parasitic losses to reduce GHG emissions without sacrificing acceleration performance or drive quality. The TCT design presented in this paper is based upon these motivating principles.

The goal of this investigation is to introduce the TCT design concept primarily as an 8-speed and compare its performance relative to an 8-speed planetary AT with torque converter and leading edge technologies. The attributes of acceleration, both launch and passing, parasitic torque loss, energy usage and fuel consumption will be explored in detail using energy analysis based upon the vehicle-powertrain system. For the analysis conducted in this investigation, an inline four cylinder, boosted gasoline engine of 2.0L displacement and a peak torque output of 350 Nm is utilized in a 1530 kg rear wheel drive (RWD) vehicle.

TRIPLE CLUTCH TRANSMISSION

8-Speed Powerflow

The basis of the transmission kinematic design is derived from a two stage DCT design, with an added input shaft and a two speed output on each countershaft. There are three

concentric input shafts and two counter shafts, and an output shaft that is coaxial with the input shafts in a RWD configuration as shown in [Figure 1](#). Each counter shaft has fixed gears meshing with the input shafts and a 2-speed output, selectable via a full synchronizer and two planes of gears to transfer torque to the output shaft. This general arrangement will be referred to as a triple clutch transmission, abbreviated TCT, with a prefix number indicating the number of forward gear states, i.e. 8TCT.

As an 8TCT, the inner and outer input shafts each have a fixed gear while the middle input shaft has two free gears, selectable by a full synchronizer to determine coupling to the countershafts. Each countershaft has two fixed gears meshed with the input shaft gears and a 2-speed output. Reverse is achieved with an idler gear meshed to the outer input shaft and a selectable gear on one of the countershafts with a half synchronizer. The 8TCT cross-section is shown [Figure 1](#), containing 8 forward and 2 reverse gear states from 6 planes of gears. The gear ratios resulting from equal center distance countershafts with the same module for all gear pairs are contained in [Figure 1](#). The gear ratios follow nearly a geometric decay producing a flat ratio progression.

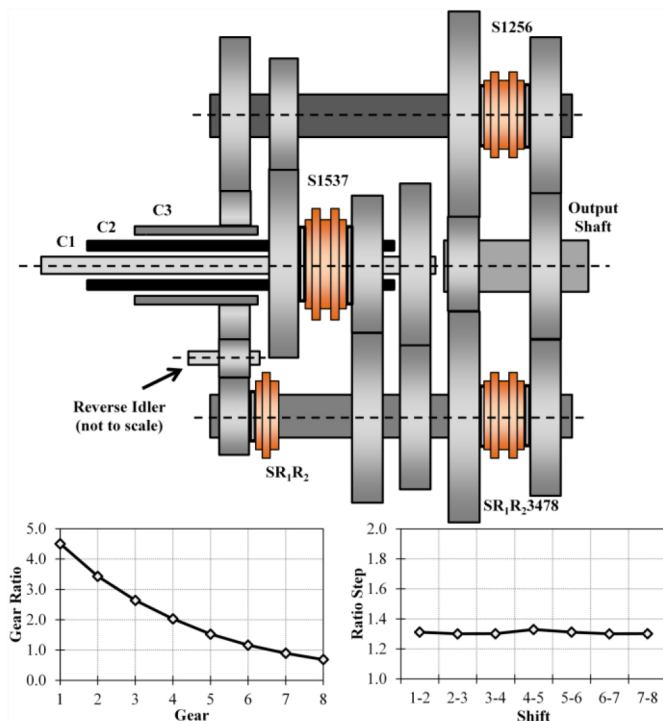


Figure 1. 8TCT cross-section and example gear ratios and progression.

Wet clutches could serve as both the launch device and shifting elements, coupling the prime mover to the transmission input shafts via a spring mass isolator such as a dual mass flywheel. Alternatively, a torque converter with a lockup clutch and damper could be used to drive a common triple clutch housing to couple the input shafts with wet clutches. For this investigation, a torque converter and wet

shifting clutches will be considered to facilitate a more direct performance comparison with a conventional planetary AT with torque converter.

Although a RWD configuration is presented in [Figure 1](#) with coaxial input and output shafts, a front wheel drive (FWD) configuration is equally as applicable to the TCT concept. The principal design difference is that the output shaft is no longer coaxial with input shafts. An additional transfer axis is required to transfer torque to the final drive and differential.

Shift Operation

[Table 1](#) contains the clutching sequence for the 8TCT arrangement shown in [Figure 1](#). Shifting of gear states occurs by switching input shafts, countershafts and the selectable output gears on the countershafts meshed with the output. [Table 2](#) contains all the possible clutch-to-clutch (C2C) shifts possible with the 8TCT that allow uninterrupted power transfer. All sequential upshifts and downshifts are nominally C2C. A list of possible C2C skip shifts is also provided in [Table 2](#). All other desired skip shifts than those listed in [Table 2](#) will need to be sequenced or executed through neutral.

Table 1. Clutching table for 8TCT, X=torque carrying element, O=engaged, non-torque carrying element.

Gear	C1	C2	C3	SR	S1537	S1256	SR3478
R1			X	X	O		X
R2			X	X	O		X
N					X	X	O
1		X			X	X	O
2			X		O	X	O
3		X			X	O	X
4	X				O	O	X
5		X			X	X	O
6			X		O	X	O
7		X			X	O	X
8	X				O	O	X

Table 2. Available clutch-to-clutch shifts with 8TCT.

Sequential Shifts		Skip Shifts	
1-2	2-1	1-4	4-1
2-3	3-2	2-4	4-2
3-4	4-3	3-6	6-3
4-5	5-4	4-6	6-4
5-6	6-5	2-7	7-2
6-7	7-6	4-7	7-4
7-8	8-7	5-8	8-5
		6-8	8-6

9 and 10 Speed Variants

TCT variants with more than 8 forward gear states can be created with additional synchronizers and planes of gears. A 9 speed, 9TCT, variant can be made by adding a synchronizer to directly couple the inner input shaft and output shaft. This configuration provides nearly the same gear ratio's and overall ratio spread as the 8TCT, but with direct drive at 7th gear. The gear shifts up to the 5-6 are nearly equal step sizes at 1.3, while the 6-7 and 7-8 shifts are smaller at 1.15, before increasing to 1.3 for the 8-9 shift. Adding a seventh plane of gears and a full synchronizer to the inner input shaft of the 8TCT of Figure 1 will form a 10 speed, 10TCT. The 10TCT gear ratios follow a near geometric progression with a 4.6:1 1st and 0.63 10th gear. One downside of the 10TCT arrangement is that not all the sequential shifts are C2C. The 5-6 upshift and 6-5 downshift requires a through neutral shift with S1245610 synchronizer needing to change position in addition to switching the C3 and C2 input clutches. The 5-6/6-5 shift will typically occur at vehicle speeds between 20 to 35 kph, well away from initial launch. With a small ratio step of 1.22 and at higher vehicle speeds it is possible that the 5-6/6-5 shift can be executed without being perceived as poor shift quality if calibrated appropriately. Figure 2 provides a comparison of the gear ratios and ratios progressions of the 8, 9 and 10 triple input clutch transmission variants.

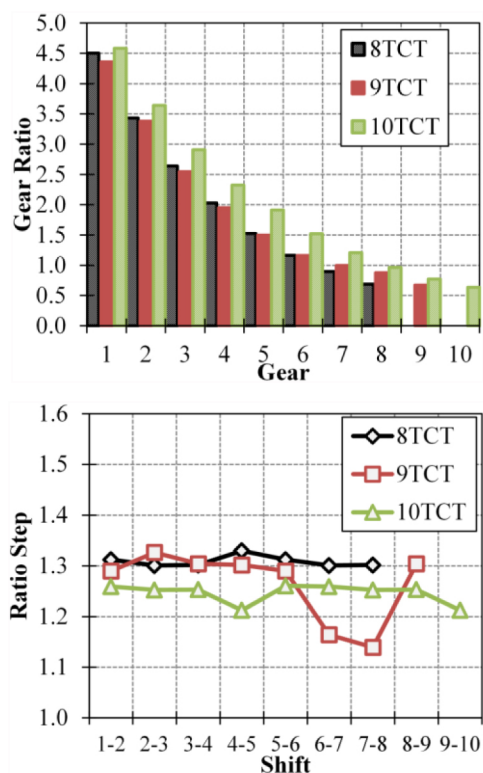


Figure 2. Example 9TCT and 10 TCT gear ratios and ratio progressions.

8-Speed Benchmarking

There are currently a number of 8-speed planetary ATs available in passenger vehicles that facilitate performance comparison with the 8TCT. Figures 3 and 4 contain the forward gear ratios and progressions of the 8-speed RWD transmission currently available in the market. These transmissions are labeled 8 Spd RWD 1 through 3; see [4 and 5] for more details. For the 8-speed transmissions in Figure 3, 1st gear ratios range from 4.5 to 4.7 while 8th gears are all between 0.64 and 0.69. The distinguishing feature of the 8TCT noted in Figure 4 is its flat, geometric ratio progression, providing taller gear ratios for 2nd through 8th, when compared to 8 Spd RWD's 1 through 3. The other 8-speed options have large ratio steps initially that decrease as top gear is approached; the 8TCT ratio steps are all roughly 1.3. The result is closely spaced launch gears and larger spaced cruising gears. The smaller ratio steps are a potential enabler to faster shift times due to reduced component accelerations negating the need for large amounts of torque management during upshift events for improved shift feel. The performance benefits realized from these design features will be highlighted later in the paper.

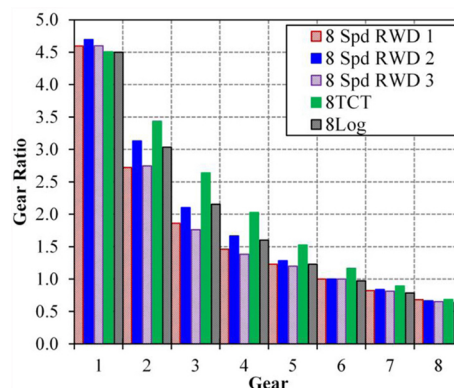


Figure 3. Competitive assessment of gear ratios in 8-speed AT currently available in the market.

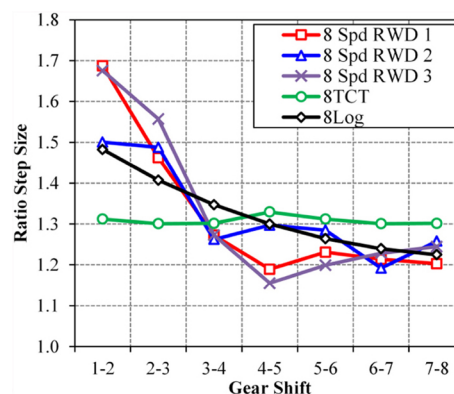


Figure 4. Competitive assessment of ratio progressions in 8-speed AT currently available in the market.

A hypothetical 8-speed planetary AT, labeled 8Log was created for this investigation to perform a more in depth performance comparison with the 8TCT. The 8Log is based upon leading edge 8-speed AT technologies, namely those designs based on 5 shifting clutches and 4 simple planetary gear sets, [4 and 5]. The transmission launches with a 1st gear ratio of 4.5 and decays based upon a logarithmic progression down to an 8th gear ratio of 0.64. The intent of this transmission is to represent an idealistic 8-speed planetary AT from a parasitic loss and ratio progression perspective. The logarithmic decay in gear ratio is a typical sought after progression in passenger vehicle design for shift quality and acceleration performance according to [18]. The remainder of this paper will focus on comparing the design attributes and performance characteristics of the 8TCT relative to the hypothetical 8Log planetary AT, both with torque converters.

PERFORMANCE

Analysis Technique and Assumptions

A first principles, energy based analysis technique was utilized in this investigation to compare the performance characteristics of the 8TCT and 8Log transmissions. The modeling technique and key assumptions around the vehicle-powertrain system follows closely the approach described by [17]. The MATLAB®/Simulink® based model includes all the relevant dynamics and powertrain controls features common to conventional vehicle-powertrain systems that contribute to the determination of performance and fuel consumption as depicted in Figure 5. All sources of energy input and output are detailed by measured or empirically derived data in lookup tables as a function of speed, load, temperature, operating pressure, and/or gear state. The model solver is based upon a tunable observer following a predefined vehicle velocity schedule. In general, this type of modeling approach shows good correlation with physical test results; see [17 and 18] for details. Since both the 8TCT and 8Log transmissions are hypothetical designs, there is no hardware test data available for correlation with the modeling results.

It can be difficult to make a direct comparison between evolving transmission designs, e.g. 6-speed AT vs. 8-speed AT, because too many of the boundary conditions that govern performance and fuel consumption have drastically changed. The goal of this investigation is to keep the comparison as relevant as possible. The same 1st gear ratio exists for both transmissions. Parasitic losses are representative of the torque transfer technology present within each transmission. A hydrodynamic torque converter with a stall K-factor rating of 200K was used for each transmission type, with the same stall torque ratio of 1.90. The torque converter clutch, TCC, apply and release criteria were maintained for both transmissions as well as the slip maps that are typically required for noise and vibration refinement. The same methodology and constraints used to develop the shift

patterns across a variety of final drive ratios (FDRs) was the applied to each type of transmission. All the relative modeling assumptions for the 8TCT and 8Log transmissions used to complete the performance analysis while maintaining the most relevant comparison possible are contained in Table 3.

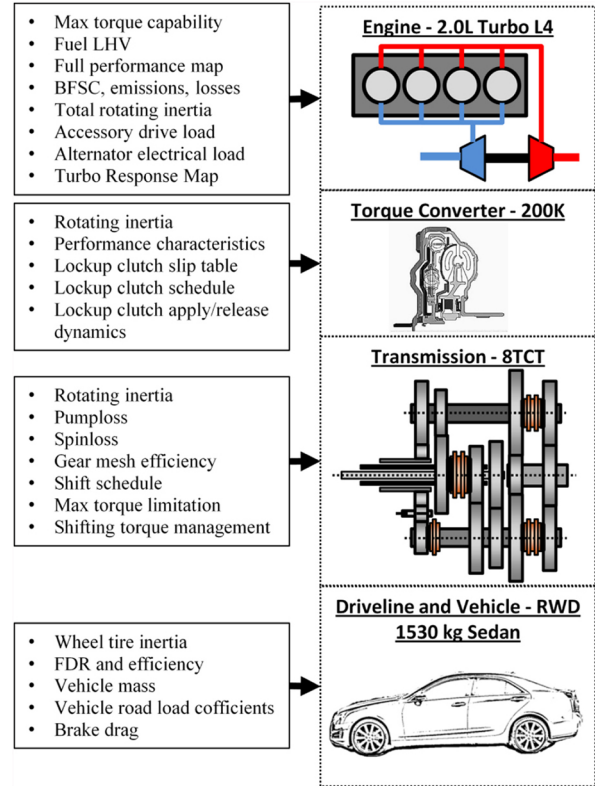


Figure 5. Parameter inputs for energy based model.

Table 3. Assumptions and inputs for 8TCT vs. 8Log performance analysis.

	8Log	8TCT
Launch Device	~ 240 mm, 200K Torque Converter	
Initial TCC Apply	1st Gear, minimum 1200 rpm	
TCC Slip Maps	Identical, load and speed dependent	
Pump	Off axis, variable displacement	
Spinloss	5 shifting clutch planetary AT	3 input clutch layshaft
Gear Efficiency	96 to 100%	98%
Shift Time	0.6 seconds	0.45 seconds
Reflected Inertia at Input (kg-m ²)	1 st - 1.39 8 th - 0.06	1 st - 0.93 8 th - 0.04
Pattern Constraints		
Min Turbine Speed in Gear, rpm	1 st 2 nd 3 rd 4 th 5 th 6 th 7 th 8 th 0 800 850 900 900 950 1050 1100	
Min DFCO Gear	5th, ~ 1.5:1 ratio	
Min Shift Hysteresis	3 to 4 kph	
Min TCC Hysteresis	3 to 4 kph	
Min PPS Hysteresis	Identical, between 8 and 24%	
FDR Options	2.56, 2.85, 3.15, 3.27, 3.45 and 3.73	