Design of Continuously Variable Transmission (CVT) with Metal Pushing Belt and Variable Pulleys

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Abstract

Nowadays, automakers have invested in new technologies in order to improve the efficiency of their products. Giant automakers have taken an important step toward achieving this objective by designing continuously variable transmission systems (CVT) to continuously adapt the power of the engine with the external load according to the optimum efficiency curve of engine and reducing fuel consumption; beside, making smooth start up and removing the shock caused by changing the gear ratio and making more pleasurable driving. Considering the specifications of one of Iranian automaker products (the Saipa Pride 131), a CVT with a metal pushing belt and variable pulleys have been designed to replace its current manual transmission system. The necessary parts and components for the CVT have been determined and considering the necessary constraints, its mechanism and components have been designed.

Keywords: *CVT, metal belt, pulley, Pride with CVT, design*

1. Introduction

Reviewing the development of Continuously Variable Transmission (CVT) technology, reveals that after the conceptualization of the first CVT by Leonardo da Vinci in 1490 [1], There were a numerous effort in different companies such as Benz, Ford, Fiat, etc to be able to equip their products to CVT. In all these years, the crucial limitation in the application of CVTs has been the capacity of torque transmission and the range of its variations. Finally, in the late 1990s, Nissan designed its own CVT that could transmit higher torque and included a torque converter. Nissan is also the only car maker to use Toroidal CVT, named the Extroid. In July 12 2006, Nissan introduced its masterpiece, Xtronic CVT, a new generation of CVTs in which the range of torque variations increased the acceleration of a car in the low ratio and decreased the fuel consumption in the high ratio. In fact Xtronic CVT is third and the most advanced generation of CVT systems with metal pushing belt and variable pulleys. [2] It has been able to allocate a significant contribution of automobile markets because of its high efficiency. However, there are various kinds of continuous variable mechanisms but metal pushing belt and variable pulleys, Toroidal and hydrostatic CVTs are more common in the automotive industry due to the larger torque transmissions and more accurate control facilities. In fact, nowadays, push belt and variable pulleys types are more commonly used in the automobiles due to higher efficiency and a more accurate control. Although CVT has some weaknesses, they are getting better and better over time. In this regard, these years some other automakers have been trying to introduce their new models of CVT. For instance, Toyota with introducing a new control strategy in its new model (Toyota Corolla 2014) has tried to solve one of the most important weaknesses of CVT which is sloping of the metal belt on pulleys [3]. This phenomenon leads to less car acceleration and also more fuel consumption. In this regard, Toyota via using the CVTi-S (CVT with intelligent shift) has succeeded to convert the continuous range of changes of torque ratio into some districts shift points. This method has been applied to increase the efficiency of the car [4]. Nevertheless, CVT has a lot of advantages which make it an ideal transmission system for cars. The main advantage of CVT is its continuous compatibility with the external load which enables the car to use the optimized function of the engine which not only decreases the fuel consumption, but also eliminates the shocks of the traditional transmission systems and improves acceleration and in so doing the driver feels more comfortable. Also, due to the existence of a torque converter in CVT, the car starts smoothly and compared to manual transmission, in similar conditions such as equal slope, speed, etc. less torque is needed to be produced by the engine for the movement of a car.

Although the number of cars produced and assembled in Iran per year has increased, the quality and safety of the automobiles have not improved accordingly. Hence, an emergency legislative plan was enacted in 2010 in order to improve the quality of domestic vehicles, decrease the fuel consumption, increase the safety and encourage competition in the automotive industry [5]. In this regard, a CVT was designed for a specific car to replace its previous manual transmission. Fortunately, by adopting the approach of this study, it is possible to design a CVT for any potential automobiles which work with manual transmission in order to produce more efficient cars.

This study has an industrial approach without considering the complexities of making an experimental one. It tries to design an efficient push belt and pulley type continuous transmission modified according to the target automobile by installing and replacing the manual transmission without changing the other parts of the vehicle considering the newest generation of CVT. It should be noted that the design is not based on reverse engineering. In fact, firstly the criteria, aims and design constraints which are essential for designing and installing a CVT are identified, and then all the required parts are specified which lead to the selection of the best mechanism through the examination of all the relevant factors. This mechanism is designed competitive with the latest generation of CVTs which has not been currently much changed and the mechanism of these new models has the most efficient state. Most current researches concentrate on generating better control strategies, therefore, this study deals with the mechanism and mechanical design of CVT as the most important stage of manufacturing.

After determining design criteria and constraints of all parts, the design details of parts are described and presented. New relations for hollow shafts have been obtained from the general rigid shaft formula during shaft design. The fields in which CVT systems can evolve in the future are introduced in the Future Development of CVT. Finally, the charts related to the functions of designed system from its simulated mechanism in CATIA were shown and the results of replacement of the manual transmission with this system in the target car are discussed in conclusion.

2. Introducing the Criteria and the Primary Design Constraints of CVT for the Target Car

In order to design and install CVT on the target car, the following criteria and constraints are needed to reach the optimized design:

The ability to produce the needed ratio of revolution-torque in order to produce enough torque to overcome the critical load Smooth and soft start-up Prevention of shutdown while the vehicle is stopped or in other words the ability to separate transmission system from the engine The ability to create reverse rotation in order to move the vehicle backward The ability to create the neutral state in the transmission system The ability of the pulleys to turn back to the primary state while reducing the speed restarting after a sudden stop of the vehicle The ability to automatically and precisely control the pulleys to reach their maximum efficiency Elimination of the clearance of different parts especially about the parts with axial movement such as pulleys Generating enough force to change the pulleys’ states from low ratio to overdrive or vice versa The ability to lubricate different parts of the system installing the park gear to prevent movements of the vehicle on a slope Reducing the transmission system weight as much as possible to increase the weight-related efficiency The ability to join with other parts of the transmission system of the target vehicle The ability to assemble different parts of the system in the process of its production and assembly positioning in the target vehicle with the minimum modifications of the other parts

3. Torque Converter

Torque converter is utilized between engine and automatic power transmission. It is capable of increasing the torque ratio and thus the car’s acceleration. Actually, the main function of torque converter is automatic stopping and connecting the engine and power transmission mechanism which prevents the shutdown of the vehicle while it is stopped. It is noteworthy that it is used just in a very low speed to smoothly start the vehicle and improve the car's acceleration when it launches and also increasing the ability of the car to move uphill. The torque convertor of Pride with automatic gearbox is used in our design which is shown in Figure 1.

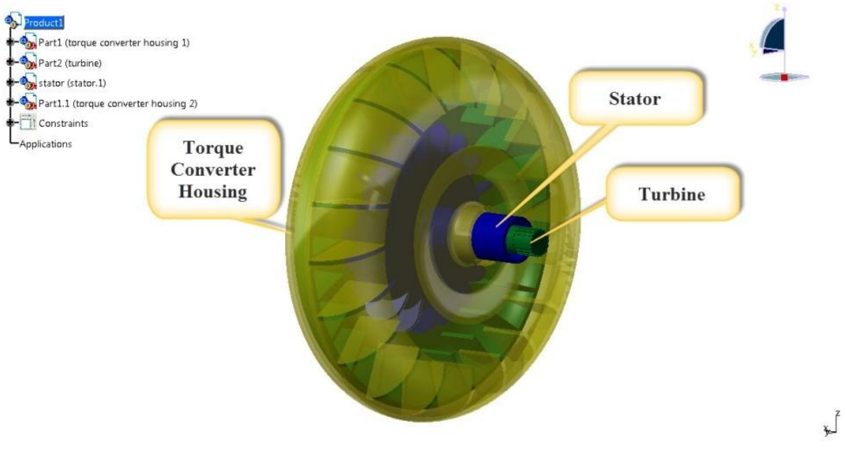


Fig1. Components of Torque Convertor

4. Planetary Gearbox

It is expected from a power transmission system to change the state of the car to a forward, backward or neutral state by utilizing clutches. The characteristics of a planetary gearbox making it an ideal mechanism for this purpose are:

All the parts of the planetary gearbox are located on the main shaft and therefore occupy a lower space comparing to other mechanisms

The planetary gears are always involved with each other. So the possibility of omitting a gear, breaking or noisy functions reduces. Shifting to other gears is also accomplished quickly, automatically and without power downfall.

The capability of power transmission increases in planetary gears because of the transition of the torque among planetary gears and the division of the power among several planetary gears.

Also in the automatic power transmission systems like CVT, it is necessary to use hydraulic clutch or torque converter instead of the frictional clutch. In the planetary gearboxes, which are regarded as automatic gearboxes, since shifting a gear is accomplished under a load it is essential to use torque converter to prevent the resulting shock of the shifting gear or vehicle start-up and therefore it is the perfect option for coupling with the CVT.

5. Using Hydraulics to Control the Pulleys

In the push belt type CVT, belt and pulleys are the main parts of the power transmission. Since the half pulleys are subject to a large amount of force from the belt (whose amount is specified in section 14.5.1) and need high accuracy in creating a smooth and constant movement in the change of the rotation ratio, using the hydraulic system is the best option. It is also possible to use solenoid valves, which have the possibility of being under an accurate control from the electrical section of the vehicle, to control the pulleys and power transmission system automatically.

6. Park gear

To prevent the movement of a car which has been parked on a slope, a part of the power transmission system, connected to the output shafts (axle shafts) must be fixed. Due to the placement considerations of the involvement mechanism and in order to provide a simpler mechanism, park gear is located on the stable half pulley of the secondary pulley which makes the involvement possible by creating some gears on the perimeter of this half pulley. In designing these gears the following points should be considered:

When the car is parked on an uphill, the driver must be able to release it.

With the involvement of the mechanism, the car’s weight should not cause the lever to jump and be released.

For this reason the gears are designed with angled sides in order to be involved when a little force is exerted and when the lever is needed to release they become easily released.

7. Reduction Gears

However, the steel belts used in CVT are more flexible than the normal belts, the flexibility constraints and the diameter restraint of the pulleys require an extra mechanism. A mechanical mechanism such as differential and some idle gears in order to provide the needed ratio to overcome the critical load and change the rotation and torque based on the need.

Consequently, it is possible to use a reduction gear before the primary pulley or after the secondary Pulley to satisfy the ratio needed. In this regard, helical gears are used due to the transferability of a larger amount of the torque, avoiding the extra noise and providing a better involvement.

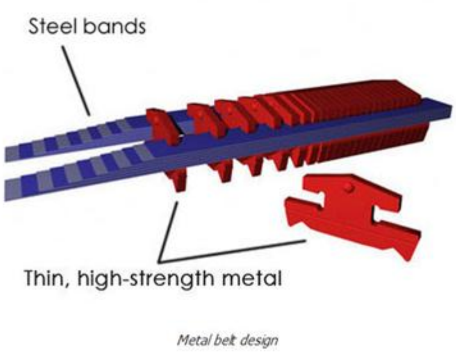


Fig2. steel belt used in pushing belt CVT

8. Belt

In the continuous variable transmissions, it is possible to create an infinite number of torque ratios. But it should be noted that the belt may slip or stretch in a deforming way which reduces the efficiency. Nowadays, with the use of newer materials in making belts this kind of depreciation has been reduced. One of the important advances in this field is utilizing the steel belts (Figure 2). These flexible belts comprise a set of thin steel bands (9 to 12) which have passed through the metal blocks and are placed on one another to increase their strength. The aforementioned steel bands do not slip and have a high resistance and are capable of transmitting larger torque. These belts also produce fewer noises comparing to the rubber belts and work under pressure.

9. The Possibility of Coupling the Designed CVT with the Other Parts of the Target Car

To install the new power transmission system on the target car, it is very important for CVT to couple with the engine in the power transmission input and to couple with the location of the exerted external load on the car, which is differential in this study. Considering a same critical load for automatic Pride with planetary gearbox and a Pride with CVT, the same torque converter of Pride with automatic gearbox is used for CVT as coupling component of one side. Furthermore, the same input gears to the Pride differential are installed in the other side in order to use the same differential and axle shafts of current Prides.

10. Mechanism Design

10.1 First Mechanism

As it was mentioned earlier, torque converter, planetary gearbox, pulleys and the reduction gears are the main components of the CVT which create the possibility of various mechanisms though different positioning. Torque converter is the most important part whose change of its position can create different effects. In one mechanism, it can be located between engine and the primary pulley or it can be located after the secondary pulley and before the reduction gears which makes a different mechanism, all of which have their advantages and disadvantages.

In this mechanism the biggest problem is the delay of the pulleys to return to their beginning position when the car starts or has a sudden stop which reduces the acceleration of the car and on the other hand, when the pulleys return to their original position after a sudden stop, the torque may increase sufficiently for the movement of the car before the completion of the process of returning to the low ratio, but due to the fact that torque ratio has not reached its first state, the acceleration of the car decreases and the slipping time and amount in converter increases to compensate the shortage of the torque which leads to an increase in fuel consumption; this issue casts a shadow on one of the most important advantages of CVT, which is reducing the fuel consumption.

10.2 Second Mechanism

In the second mechanism as it is shown in Figure 3, torque converter is positioned after the engine and before the primary pulley.

Advantages and disadvantage of this mechanism are as follows:

The most important advantage of this mechanism comparing to the previous one is receiving feedback of the car from the speed sensors when the car has a sudden stop, the pulleys are returned to their beginning position. The disadvantage of this mechanism is using of various sensors makes the control of the mechanism complicated.

Since we need high efficiency and accuracy, the Second mechanism is opted for.

11. The Possibility of Positioning in the Target Car

Considering the estimated dimensions of some of the main parts of CVT which has been presented in Table 1, preliminary evaluation reveals that there is a possibility of putting CVT in the target car that is shown in Figure 4.

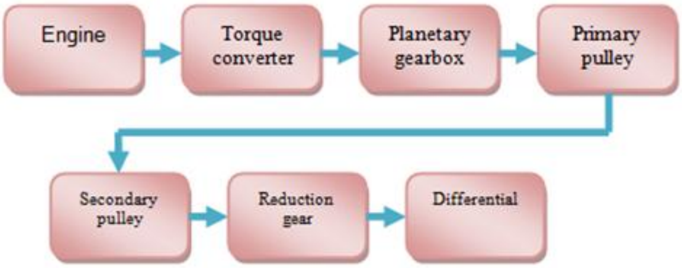


Fig3. ordering the various components of the CVT mechanism number 2

Table 1. Some of the main components dimensions in the CVT

	Pride torque converter	Planetary gearbox	Primary pulley	Secondary pulley
diameter	245	224	145	148.2
width	75	100	44	44

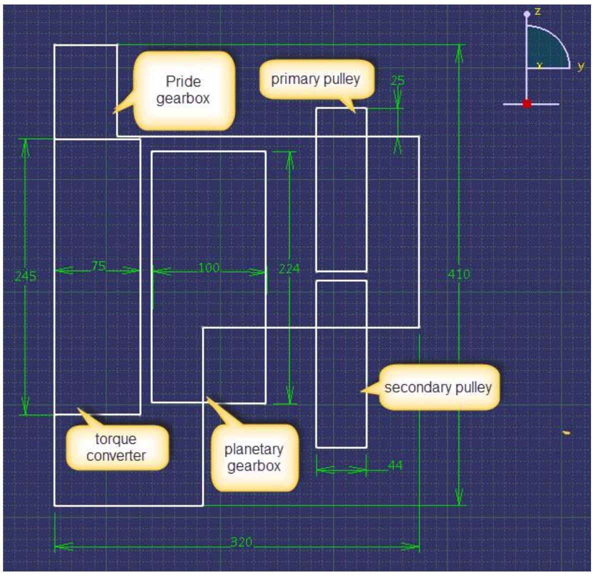


Fig4. The Possibility of Positioning in the Target Car

12. Mechanism Design of the Planetary Gearbox

It is possible to produce six different states when a simple planetary gearbox including a sun gear and three planetary gears that orbit around the sun gear, and a ring gear which has been presented in Table 2:

After examining all the six states and calculating the produced ratio of the gears in each case and also the number of the needed clutches to control the gear functions, the following mechanism, whose components are shown in the exploded map of Figure 5, should be utilized.

In this mechanism, for the reverse status, the first state of the dynamic analysis of the planetary gearbox, which has been presented in table 2, is utilized and in the forward status the direct transition of the engine torque to the pulleys is utilized (planetary gearbox has no effect in increasing torque). To compensate the lack of the needed ratio, the reduction gears are used before differential.

In this mechanism two clutches are needed to create different status including forward, reverse, neutral and park. The forward clutch is used for the forward movement and the reverse brake is used for the backward movement. Different situations of the clutch involvement (released and engaged) in a car are presented in Table 3. [6]

As it is clear in Table 3, the clutches are connected like the following:

Forward clutch is connected to planetary gear and when it is involved, the planetary gear rotates in the direction of the engine.

Reverse brake is connected to planet carrier and when it is involved, the carrier becomes fixed and accordingly to a ring gear rotation, the planetary gear rotates in the reverse direction of the engine. In this state the forward clutch changes its position to the released state.

Table 2: six various states can be created by planetary gearbox

Fixed carrier	1	Reverse	R driving and s driven
	2	Reverse	S driving and R driven
Fixed ring gear	3	Forward	A driving and S driven
	4	Forward	S driving and R driven
Fixed sun gear	5	Forward	A driving and R driven
	6	Forward	R driving and A driven

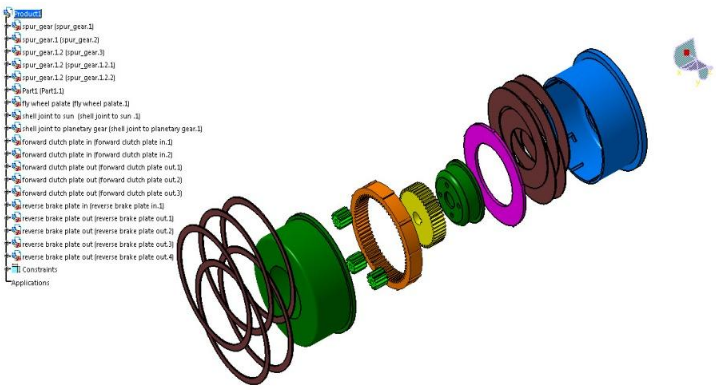
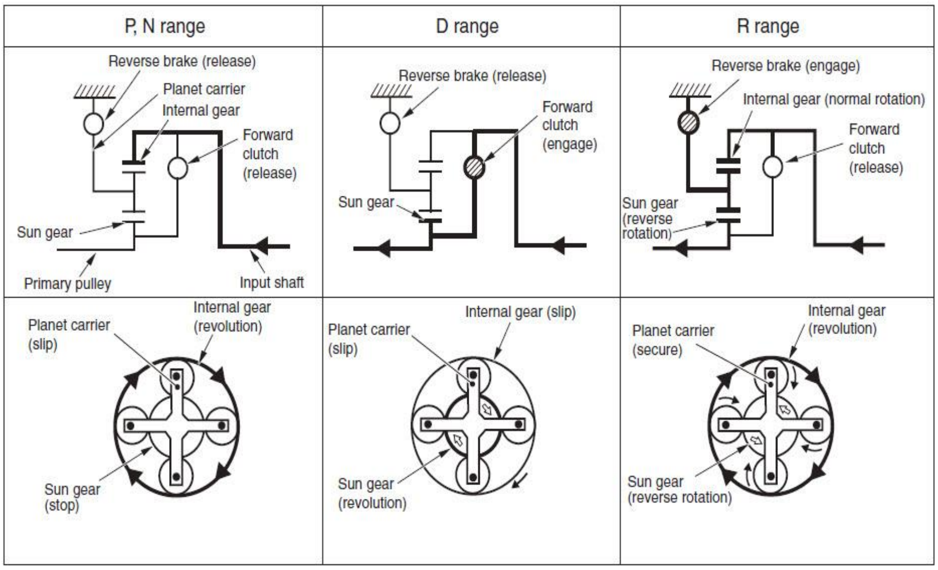


Fig5. planetary gearbox components at second mechanism

Table 3. Various vehicle states created by planetary gearbox



13. Mechanism of Pulleys

In the mechanism design of the pulleys, the followings should be noted:

The shaft-like section connected to the stable half pulleys should be built perforated to satisfy the following needs:

Lubrication of the tank of the movable pulley and lubrication of movable parts of the pulley

Lower weight, creating lower rotational inertia and consuming fewer raw materials in manufacturing

Connecting the related shafts to the pulleys' system Smooth and frictionless movement of the movable half pulleys while the ratio changes Predicting the bearings' location and creating a step and reducing the diameter in different parts of the pulleys' shaft for a better assembly and preventing the movement of the bearings and controlling the range of the movement of different parts. Predicting the necessary mechanism for the pulleys to return to their initial state Predicting the installment location of the output or reduction gears on the secondary pulley's shaft Designing the form of the pulleys' tank's shell Designing some teeth on the secondary pulley for the park gear mechanism To illustrate how pulleys' mechanism works, two situations should be considered: one when the pulleys' ratio changes from low ratio to overdrive and the other when the pulleys' ratio changes from overdrive to low ratio.

The important point in understanding the mechanism is that there is a belt between the pulleys which has a fixed length and is under pressure. In the first situation, when the pressure of the lubricant behind the movable half pulley of the secondary pulley decreases, the movable half pulley of the secondary pulley is pushed back by the exerted force of belt, which is along the shaft, on the half pulleys' surface and thus the rotation diameter of the secondary pulley starts to decrease. Due to the fixed length of the belt and exerted force from the belt's blocks on each other, the belt's diameter in the primary pulley along with the volume of the lubricant behind the movable half pulley of the primary pulley increases and therefore the movable half pulley of the primary pulley starts its movement and creates the necessary pressure to provide the normal force of the surface of the half pulleys to be exerted on the belt in order to satisfy required friction. In the second situation, everything is the same as the first one, except a spring is used for producing the normal force to provide the friction of the secondary pulley. Since each pulley should have the capability to create the necessary force to move forward or backward, in the primary pulley, the forward movement of the movable half pulley is done through increasing the volume of the lubricant in its tank but its backward movement is done through the axial force of the belt on its surface. The spring provides the forward movement of the movable half pulley of the secondary pulley and the backward movement of the secondary pulley is done by increasing the volume of the lubricant in its tank, contrary to primary pulley. The reason of this issue is presented in Figure 6. In fact, the creation of the reverse movement of the secondary pulley in comparison with primary pulley in spite of increasing the lubricant's volume in both the pulleys is due to the fact that there are two shells in the movable half pulley of the secondary pulley in which the inner shell is fixed to the shaft and with the increasing of the lubricant's volume the created force is exerted on the outer moving shell which is fixed to the movable half pulley and makes the backward movement. But in the movable half pulley of the primary pulley, there is only one fixed shell and the forward movement is made through the increase of the lubricant's volume in the tank. The outer and inner shells and other components are named in Figure 7.

With respect to the abovementioned explanations, different pieces of the second mechanism, which are shown in the exploded map of Figure 7, are:

1- Fly wheel
2- Outer shell of the torque converter
3- Turbine
4- Stator
5- Shell of torque converter (pump)
6- Shell of planetary gearbox (connected to turbine)
7- External plate of forward clutch
8- Internal plate of forward clutch
9- Fixed disc connected to the forward clutch
10- Connected shell to the planetary gearbox
11- Sun gear
12- Ring gear
13- Planetary gear
14- Outer shell of the planetary gearbox
15- Internal plate of reverse brake
16- External plate of reverse brake
17- Shaft No.2
18- Shaft No.3 (stable primary half pulley)
19- Movable primary half pulley
20- Fixed tank's shell of the primary movable half pulley
21- Shaft No.4 (stable secondary half pulley)
22- Secondary movable half pulley
23- Spring
24- Fixed shell of the secondary movable half pulley
25- Moving shell of the secondary movable half pulley
26- Output gear
27-shaft No.5
28- Idle gear
29- Differential

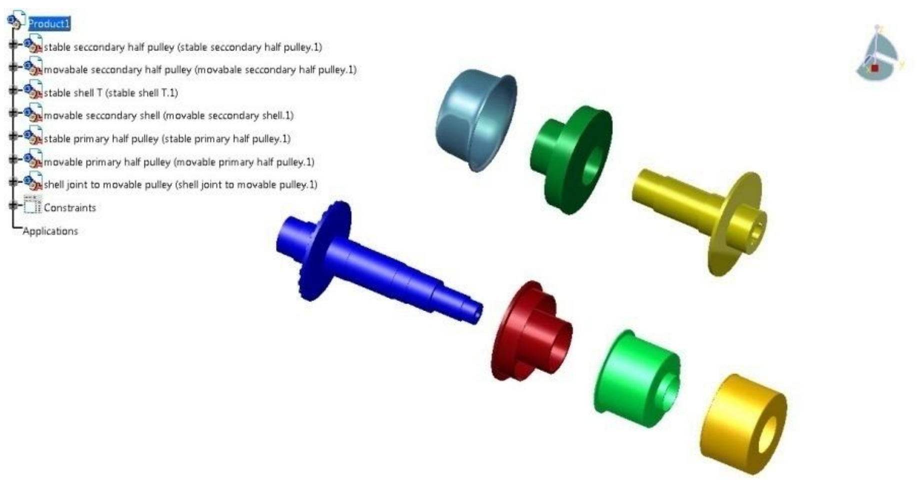


Fig6. View of the primary and secondary pulley components

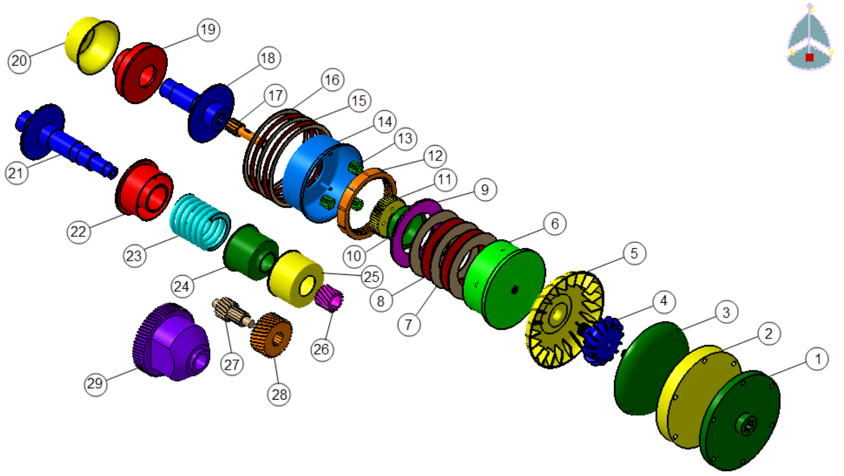


Fig7. view of various components of designed CVT

14. Components Design

14.1 Critical Input Load

In accordance with what is mentioned in Pride's guide book [7], maximum torque happens at 2500 rpm and maximum engine power happens at 5000 rpm which their values are:

$$T_{\max})_{n=2500rpm} = 103N.m \qquad (1)$$
$$P_{\max})_{n=5000rpm} = 63hp$$

Therefore, the maximum torque at maximum power is equal to:

$$T = \frac{P}{\omega} \Rightarrow T_{p_{\max}} = 89.75N.m \qquad (2)$$

So the maximum torque generated by engine occurs at 2500 rpm and its value equal 103 N.m. On the other hand when the car speed is less than 20 km / h, the torque produced by engine is doubled by the torque converter [8]. Thus, the critical input load exerted into the CVT system equals 206 N.m at 2500 rpm that it changes for any of the components relative to the value of its torque ratio.

Overdrive status can create another critical situation when a dynamic load is exerted to the components and rotary components are in the highest rotational speed rate. In this situation, torque and rotation of the secondary pulley are equal to:

$$T_{S.P} = T_{P.P} \times Ratio = 45.8N.m \qquad (3)$$
$$n_{S.P} = {n_{P.P}}/{Ratio} = 5618rpm$$

14.2 Selection of Torque Converter

According to the mentioned reasons in the section 9, the torque converter of Pride with automatic gearbox is used in the designed CVT. Its specifications are shown in Table 4:

14.3 Selection of Belt

Having the necessary capacity for the required torque as well as create a required range of torque variation according to the specification of the target car are the most important factors in the selection of CVT belt. The capacity of transmitted torque by metal belt must be at least:

$$T = T_{\max} \times R_{T.C} \times R_{CVT\ \ at\ \ N=2500rpm} \qquad (4)$$
$$= 366N.m$$

These kinds of belts are produced just in limited classes and for special cars. Thus, after plenty of researches eventually the metal belt No. P811 whose specifications are given in Table 5 [9] is chosen considering the car specification. The only problem is its Low Ratio which is less than required rate. As it is explained in the following sections, for solving this problem, some reduction gears are used before differential.

Table 4: Specifications of torque converter used in the designed CVT

Part name	Diameter(mm)	Width(mm)	
torque converter	245	75	2

Table 5: Specifications of the selected metal belt for the CVT

Variator parameters	
Center distance [mm]	155
Belt length [mm]	649.7
Element width [mm]	24
Number of rings per set	9
Ratio coverage [-]	5.55
Lubrication fluid type	ESSO EZL799
LOW ratio [-]	2.47
primary running radius in LOW [mm]	30.0
secondary running radius in LOW [mm]	74.1
TOP ratio [-]	0.6
primary running radius in TOP [mm]	66.7
secondary running radius in TOP [mm]	40.0
OD ratio [-]	0.445
primary running radius in OD [mm]	72.5
secondary running radius in OD [mm]	32.3

14.4 Planetary Gearbox

14.4.1 Dynamic Analysis

As it has been described in section 12, for the reverse status, a fixed carrier, a driving ring and a driven sun gear is used. For the forward status, the torque is transmitted from engine to the pulleys through the torque converter directly. After performing the relevant calculations, the reverse gear ratio has been obtained as follows:

$$\frac{T_R}{T_S} = \frac{N_R}{N_S} = 1.5, Backward\ \ Ratio = 3.2 \quad (5)$$

As a result, the reduction gear ratio can be expressed as follows:

$$Reduction\ gear\ ratio = \frac{backward\ \ ratio}{CVT\ \ low\ \ ratio\,/\,Planetary\ \ ratio} \quad (6)$$

$$= 1.943$$

In this situation, the torque ratio in CVT when the car starts moving forward is equal to:

$$CVT\ \ low\ \ ratio\ at\ \ forward\ \ range = \frac{Pride\ \ forward\ \ ratio}{reduction\ gear\ \ ratio} \quad (7)$$

$$= 1.777$$

14.4.2 Gears Design

After several calculations, considering the constraints of components location, the pressure angle ($\varphi = 22.5°$), AGMA standards [10], geometric constraints and designing constraints introduced in reference number [11] the gears of planetary gearbox are designed. In this regard, firstly the ring gear and the planetary gears and secondly the sun gear and the planetary gears have been designed, and finally the larger width has been chosen. The main constraints considered are:

- The planetary gearbox should not have clash with the secondary pulley after locating.
- The selection of gear ratio between ring gear and planetary gear and thus the diameter of planetary gears should resist the input force and torque.
- In addition to the above constraints, after the dynamic analysis and selection of appropriate ratio, to ensure full involvement of the teeth together, the

gears must satisfy five criteria listed in reference number [11].

According to the calculations, the width of gears is obtained which equals $b = 24.83mm \cong 25mm$ and also the position angles of the planetary gears around the sun gear equal: $\theta_1 = 123°, \theta_2 = 123°, \theta_3 = 114°$

The number of gears teeth according to the design constraints introduced at reference number [10] and thus the diameter of each of the gears are:

$$N_P = 12, N_S = 48, N_R = 72 \tag{8}$$
$$d_P = m \times N_P = 24mm$$
$$d_S = m \times N_S = 96mm$$
$$d_R = m \times N_R = 144mm$$

14.4.3 Design of Forward Clutch and Reverse Brake

As it is explained in section 12, in the planetary gearbox mechanism, two clutches are needed for changing the status between neutral, forward and backward.

Due to the space limitations and the possibility of using sequential plates to overcome the torque of engine, multiple-disk clutch-brake is used. In this section the diameters and the number of plates which are needed for the clutch have been obtained. As shown in Figure 4, the following geometric constraints are considered as design constraints:

$$d_{out\ forward\ clutch} = d_{out\ ring} = 160mm \tag{9}$$
$$d_{in\ reverse\ brake} > d_{out\ ring} = 160mm$$
$$1.1 < \frac{D}{d} < 1.5$$

Note that the "Uniform Wear theory" has been used for the design of the clutch. The forward clutch specifications are obtained as follows:

$$D = 160mm, d = 106mm, n = 2 \tag{10}$$

And the specifications of the Reverse Brake are obtained as follows:

$$D = 209mm, d = 190mm, n = 2 \tag{11}$$

14.5 Design of Pulleys

14.5.1 Obtaining Exerted Forces

As it has been mentioned in section 13, each pulley consists of two half pulleys, one of which is stable and the other one is movable. The stable half pulley is assumed as a half pulley which is fixed to a separate shaft (Figure 8). Then, the thickness of half pulley and its root which is fixed to an assumptive separate shaft are obtained experimentally by finite element software, considering local stresses concentrations to be a safe region. The other part of stable half pulley is designed as a separate shaft.

In order to calculate the forces applied by the belt to surfaces of pulleys, it should be noted this kind of belt is composed of several steel bands which have passed through the metal block. The bending strength of these bands is almost zero. Therefore, when the blocks are under pressure in order to prevent buckling of the bands the resultant forces at any point of the belt must be positive or zero. In this regard, by considering CVT at low ratio status as its most critical situation which leads to the maximum compressive force in the metal blocks, the minimum force which is necessary to stretch the bands is obtained:

$$T_{S.P} = 366N.m$$
$$\Rightarrow Q_0 = \frac{T_{secondary\ pulley}}{r_{seccondary\ pulley}} = 5382N \tag{12}$$

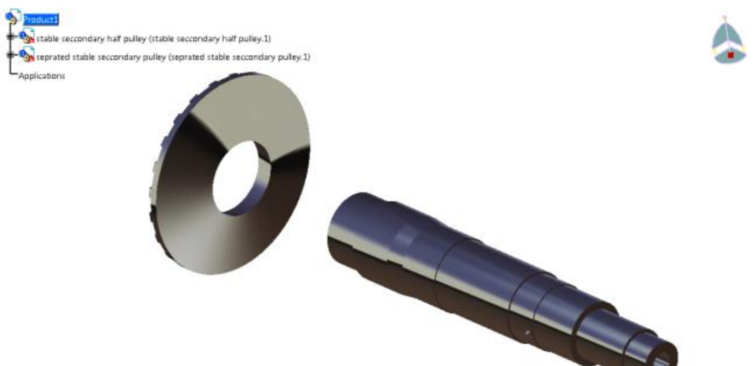


Fig8. two separate hypothetical parts at secondary stable half pulley

The method introduced in reference number [12] has been used to obtain the exerted forces from the metal blocks to the surfaces of the half pulleys. It should be noted that a friction occurs where the belt and pulleys are in contact between the metal blocks and the bands due to the discrepancy in velocity between them but it can be ignored because of a small friction coefficient [13] and therefore the tensile force in bands which was obtained in the previous section can be considered constant in all position of CVT. By considering a small element of bands and in the next stage a small element of blocks as a body diagram, first-order nonhomogeneous linear differential equation in terms of θ which represents the pressure force between the blocks is obtained:

$$\frac{dQ}{d\theta}+\left(\frac{\mu}{\sin\sin\frac{\beta}{2}}\right)Q=\frac{\mu rF_d}{\sin\sin\frac{\beta}{2}} \tag{13}$$

Considering counter-clockwise direction for the primary half pulleys and clockwise direction for the secondary half pulleys as positive direction for θ and also by considering the first point of the belt which contacts with the pulley as starting point with θ=0, the compressive force profile between the blocks can be obtained as follows:

$$Q=A\exp\exp\left(\frac{-\mu\theta}{\sin\sin\frac{\beta}{2}}\right)+rF_d \tag{14}$$

To obtain the most critical half-pulley, boundary conditions are used to evaluate constants A and μ in the above equation for each of the four half-pulleys at low ratio and overdrive status. In the next step, Q is placed at the differential equation derived from the summing forces in direction of r and then integrates the equation to obtain the normal force. The axial force can be obtained by imaging the normal force at axial direction. The results of calculation for primary half pulley at low ratio and overdrive status and for secondary half pulley at low ratio and overdrive status are as follows:

$$\begin{aligned} rF_{n1} &= 20478N \Rightarrow F_{c1}=20102N \\ rF_{n2} &= 20430N \Rightarrow F_{c2}=20055N \\ rF_{n3} &= 15917N \Rightarrow F_{c3}=15625N \\ rF_{n4} &= 15893N \Rightarrow F_{c4}=15601N \end{aligned} \tag{15}$$

14.5.2 Finite Element Analyses with ABAQUS

Stress analysis in Pulleys must be examined in both static and dynamic status at stable and movable half pulleys, in each of the primary and secondary pulleys. It is noteworthy that the stress analysis at half pulleys has been done by the ABAQUS software which in order to simulate the sheer force resulting from friction between the blocks and pulley surface, the analysis has been done in "General Static" mode and bearings are considered as fixed boundary and the loads are applied periodically. By analyzing the pulleys, it was determined that the sufficient thickness had been considered for pulleys. This issue is shown for the stable secondary half pulley in Figure 9 as a sample. The analysis also reveals that an edge fillet should be used for making a uniformed flow stress at the junction of the root of stable pulleys to their shafts.

14.6 Design of a spring for Movable Secondary Half pulley

As it has been described in details in section 13 a compression spring is needed to mount in the movable secondary half pulley.

As it is mentioned in reference number [10], "set removed" springs should not be used in fatigue thus a "before set removed" spring is used for this case. The part of movable pulley with diameter d =72 mm which has to be placed into the spring is a geometric constraint for designing the spring; furthermore the following constraints are considered [10]:

$$n_s \geq 1.2, \xi \geq 0.15, 4 \leq C \leq 12, 3 \leq N_a \leq 15 \tag{16}$$

Another proposed constraint according to reference number [14] is $\delta_2 = 2\delta_1$ which is considered to design the spring. This constraint makes the spring behave in a linear controlled way in all situations.

After examining the spring wires given in reference number [10], it has been concluded that only wire No. 7/0 can satisfy all the above constraints. Then the obtained diameter has been examined in the fatigue analysis and the safety factor has been obtained by using Zimmerli data at Gerber torsion fatigue failure criterion whose value is equal to n=2.44.

14.7 Design of Park Gear

In accordance with the mentioned reasons in section 6, the teeth of park gear are made with angular faces as much as 11° (trapezoidal teeth). The value of angle can be optimized with empirical testing. Teeth have been designed based on contact stress at static analysis. By considering 2 mm as the width of teeth, the height of them is obtained h=6 mm.

Note that on the one hand, the face area of teeth attached to the pulley is larger than another face whose contact stress has been examined and on the other hand, due to be greater shear strength compared to contact strength, shear failure does not occur in the teeth.

14.8 Design of Output Gear and Idle Gear

As it has been described in section 14.3, in order to supply the required torque ratio of the car, in addition to pulleys, the reduction gears are needed. As mentioned in section 10.2, these gears are located after the secondary pulley which makes high torque transmitted through the gears. For this reason and also to produce less noise, helical gears are used. The specifications of these gears have been calculated as follows:

$$N_P = 17 \Rightarrow N_G = m_G \times N_P = 33 \tag{17}$$
$$, b = 43.09mm$$

14.9 Determine the Estimated Location of CVT Components and the Proposed Length and Diameter of Shafts

Considering what is mentioned in the introduction, the purpose of this study is designing and installation of CVT system and replacing the manual transmission system, with minimal changes in the target car, like the location of the engine, chassis and other components connected to the lower car plate, etc. In this section, considering the dimensions of the designed parts and also the geometric constraints, the various components have been located at predicted space as shown in Figure 10. In order to obtain an estimated length of each shaft and the profile of tank shell in movable half pulleys.

Also considering the geometric dimensions of different parts such as standard bearings, gears and other components designed, and the geometric design parameters, the diameter of shafts have been estimated or in some cases, the maximum possible diameter (for example, for the linear ball bearings) has been determined. In the next section, the required diameter of each shaft has been obtained with respect to the exerted forces and then compared with estimated/required diameter.

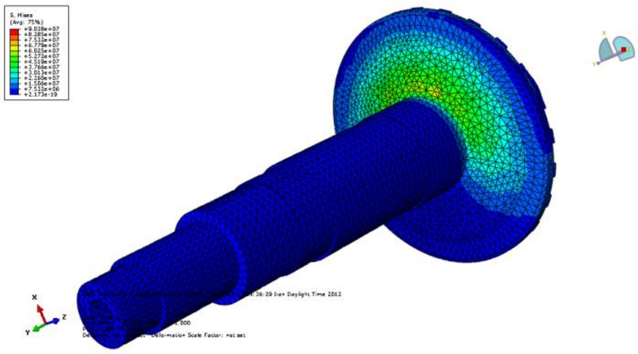


Fig9. Stress analysis of stable secondary half pulley with ABAQUS

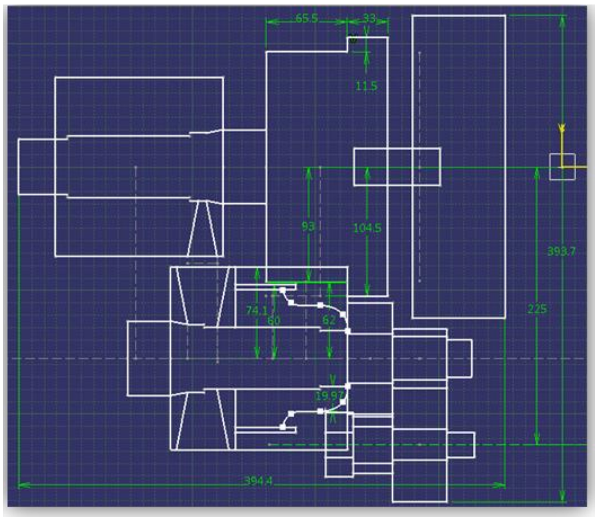


Fig10. locating the main designed components in the CVT system and estimating the shafts dimensions

14.10 Design of Shafts

In the designed CVT, there are five shafts some of which are part of other components; these shafts displayed in Figure 7, are:

1) The input shaft which connects the torque converter to the planetary gearbox.

2) Shaft number 2 on which the sun gear is mounted and which has been connected to the primary pulley by spline.

3) Shaft number 3 which is a part of the stable primary half pulley and on which the movable primary half pulley is mounted.

4) Shaft number 4 which is a part of the stable secondary half pulley and the movable secondary half pulley on which the output gear are mounted.

5) Shaft number 5 on which the reduction gear and the idle gear are mounted.

The design has been done according to Shigley's Mechanical Engineering Design [10]. Because of using the current torque converter of the car with automatic gearbox for designed CVT, the shaft of the torque converter is used as input shaft but its length should be reduced in order to be able to located between the torque converter and the planetary gear box. Their dimensions are shown in table 6:

DE-ASME Elliptic theory is used to design the shafts number 2, 3 and 4 but it should be noted that the relations introduced in reference number [10] have been obtained for a round solid bars without any axial forces. But in this project the round perforated shafts have been used for the shafts number 3 and 4 due to the reasons listed in section 13. Also, axial forces are not small enough to be ignored. Thus, the desired relation is obtained for a perforated shaft by considering the axial stresses (equation 18) in the terms of Won Mises stress. In order to calculate the axial stresses $d_{in} = \alpha * d_{out}$ is considered and by assigning an arbitrary number to α, the value of d_{out} and thereby d_{in} is achieved. Then, d_{out} is compared with the proposed diameter of the shaft that was obtained at pervious section and if it was bigger than proposed diameter, α would be changed.

$$\sigma_{aA} = k_{fA}\frac{F_a}{A} = k_{fA}\frac{F_a}{\frac{\pi}{4}(d_{out}^2 - d_{in}^2)} = k_{fA}\frac{4F_a}{\pi(1-\alpha^2)d_{out}^2} \tag{18}$$

$$\sigma_{mA} = k_{fA}\frac{F_m}{A} = k_{fA}\frac{F_m}{\frac{\pi}{4}(d_{out}^2 - d_{in}^2)} = k_{fA}\frac{4F_m}{\pi(1-\alpha^2)d_{out}^2}$$

Now, Von Mises stresses (equation No. 19) can be calculated. The loading factor (k_c) for σ_B and τ_T and the size factor (k_b) for σ_A equal one. It should be noted that σ_m is a static type and S_y has a constant value which does not depend on the diameter, so $\sigma_m^{'}$ does not change.

$$\sigma_a^{'} = \left(\left(\frac{\sigma_{aB}}{k_b} + \frac{\sigma_{aA}}{k_c}\right)^2 + 3\left(\frac{\tau_{aT}}{k_b}\right)^2\right)^{1/2} \tag{19}$$

$$= \left[\left(\frac{k_f}{k_b}\frac{32M_a}{\pi(1-\alpha^4)d_{out}^3} + \frac{k_{fA}}{k_c}\frac{4F_a}{\pi(1-\alpha^2)d_{out}^2}\right)^2 + 3\left(\frac{k_{fs}}{k_b}\frac{16T_a}{\pi(1-\alpha^4)d_{out}^3}\right)^2\right]^{1/2}$$

$$\sigma_m^{'} = \left(\left(\sigma_{mB} + \sigma_{mA}\right)^2 + 3\left(\tau_{mT}\right)^2\right)^{1/2}$$

$$= \left[\left(k_f\frac{32M_m}{\pi(1-\alpha^4)d_{out}^3} + k_{fA}\frac{4F_m}{\pi(1-\alpha^2)d_{out}^2}\right)^2 + 3\left(k_{fs}\frac{16T_m}{\pi(1-\alpha^4)d_{out}^3}\right)^2\right]^{1/2} \tag{20}$$

By substituting the Von Mises stresses in the equation DE-ASME Elliptic, the main equation for calculating the diameter is obtained:

$$ASME\ formula: \left(\frac{n\sigma_a^{'}}{S_e}\right)^2 + \left(\frac{n\sigma_m^{'}}{S_y}\right)^2 = 1 \tag{21}$$

$$\Rightarrow \frac{1}{n^2} = \left[\begin{array}{l}\left(\frac{k_f}{k_b S_e}\frac{32M_a}{\pi\left(1-\alpha^4\right)d_{out}^3} + \frac{k_{fA}}{k_c S_e}\frac{4F_a}{\pi\left(1-\alpha^2\right)d_{out}^2}\right)^2 \\ +3\left(\frac{k_{fs}}{k_b S_e}\frac{16T_a}{\pi\left(1-\alpha^4\right)d_{out}^3}\right)^2\end{array}\right]$$

$$+\left[\begin{array}{l}\left(\frac{k_f}{S_y}\frac{32M_m}{\pi(1-\alpha^4)d_{out}^3} + \frac{k_{fA}}{S_y}\frac{4F_m}{\pi(1-\alpha^2)d_{out}^2}\right)^2 \\ +3\left(\frac{k_{fs}}{S_y}\frac{16T_m}{\pi(1-\alpha^4)d_{out}^3}\right)^2\end{array}\right] \tag{22}$$

By substituting the values of each parameter in the above equation, the diameter at the detected dangerous point can be obtained for each shaft. Then, the diameters of the other sections of the shafts are calculated according to the considerations of location which was introduced in the previous section. The diameters of the shafts number 2, 3, 4 and 5, in their dangerous points (after standardization for installing bearings and hydraulic considerations) are shown in Table 7:

14.11 Design of Keys and Splines

In order to mount the various components on the shafts or due to coupling components together, some keys and splines are used on the shafts which are designed according to [15] and [16]. The keys dimensions are shown in Table 8 and the splines dimensions are shown in Table 9:

Table 7: diameters of designed shaft at their dangerous points

	External Diameter (mm)	Internal Diameter (mm)
Shaft No.2	22	14.66
Shaft No.3	50	14.66
Shaft No.4	50	14.66
Shaft No.5	32.86	-

Table 8: keys dimensions

	b*h	length
The key has been used on the shaft number two for mounting sun gear	6*6	17
The key has been used on the shaft number four for mounting the output gear	10*8	14

Table 9: splines dimensions

	d	D	B	N
The spline has been used on the left side of the shaft number two in order to join the shaft with the primary pulley	**28**	**32**	**7**	**6**
The spline has been used for mounting the idle gear on the shaft number five	**32**	**36**	**6**	**8**

15. The Future Development of CVT

The ratio coverage and torque capacity were the main problems, but nowadays, Toyota Corolla has the ratio coverage of 6.26 (0.396-2.48) in its new model (CVTi-S) [4] and the new JATCO CVT in Nissan (2.0 L-class, FWD-Vehicles) has the ratio coverage of 7.0. [2] This development in ratio coverage has been achieved by smaller minimum diameter of metal belt and adding an auxiliary transmission. Furthermore, torque capacity could increase from about 100 Nm to 380 Nm with a torque converter and to 400 Nm without it in the third generation, e.g. in Xtronic CVT in Nissan. This improvement was achieved by increasing the belt size [2].

The other challenge is evolution of CVT technology is improving fuel consumption. This issue has already been followed by the capability of adaptation of engine power with the exact required torque for diverse driving situation which is followed by increasing the torque ratio and torque capacity and also improving control methods. In addition to these factors, the focus of researchers is on the enhancement of mechanical efficiency ,too. Accordingly, reducing the mechanical friction (internal belt friction between bands and block and also external friction between block and pulleys) is one of the best subjects for improvement which is obtained by modification of metal belt (Figure 11) and improving the efficiency of oil pump. The pump could be improved in both the body of the oil pump and its hydraulic system. Nissan, as one of the pioneers in this field, has well focused on this issue by reducing the fluid leakage and optimizing required hydraulic pressure (Figure 12) [2].

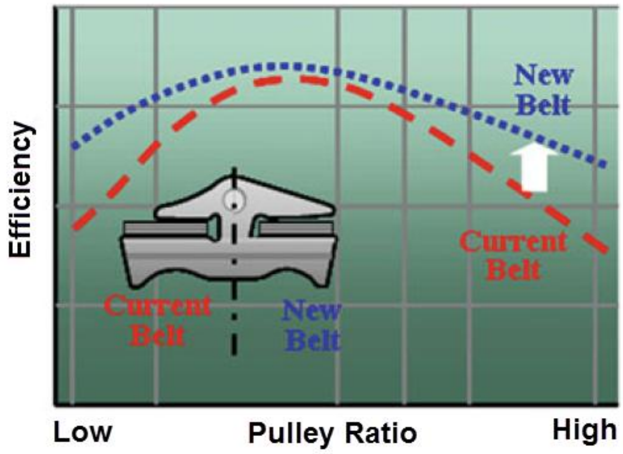


Fig11. comparing current and new metal belt in new model of CVT of Nissan [2]

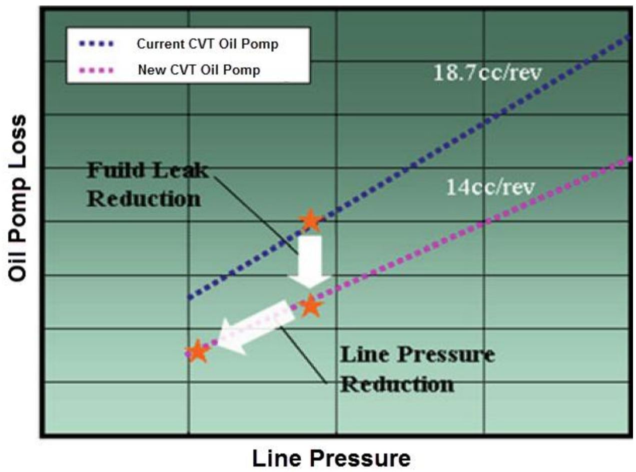


Fig12. comparing current and new CVT oil pump in new model of CVT of Nissan [2]

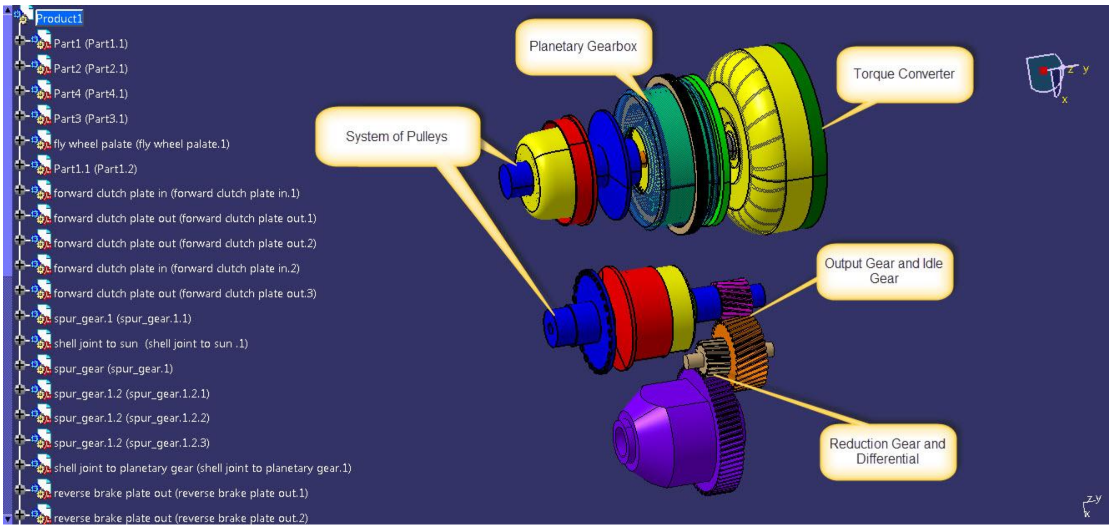


Fig13. : the ultimate mechanism of designed CVT

Table 10: The main subsystems and the source of supplying

No.	Subsystem	Source	Main Function	Explanation
1	Torque Converter	Outsourced	-Automatic connection /disconnection between engine and transmission system -increasing start-up torque	-
2	Planetary Gearbox	Designed	Switching between reverse/neutral/forward status	It must be designed based on geometric constrains and internal torque
3	System of Pulleys	Designed	- Vary the revolution /torque continuously	It must be designed based on required torque and shifting axial displacement
4	Output Gear and Idle Gear	Designed	- Increasing torque to the required torque	-
5	Reduction Gear and Differential	Outsourced	The differential of the target car is preferably used	-

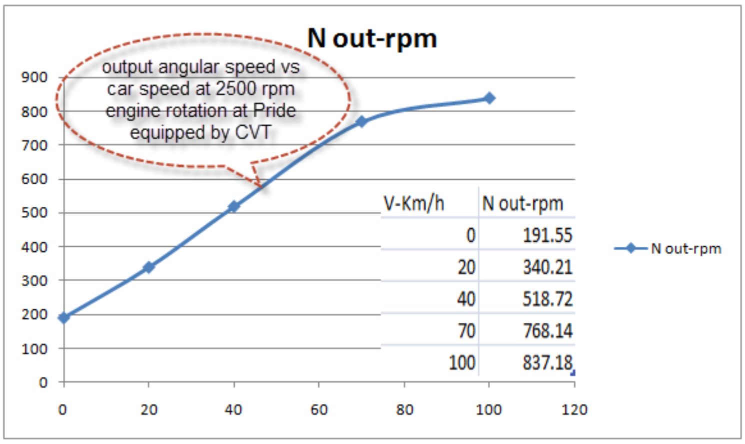


Fig14. CVT output angular speed diagram

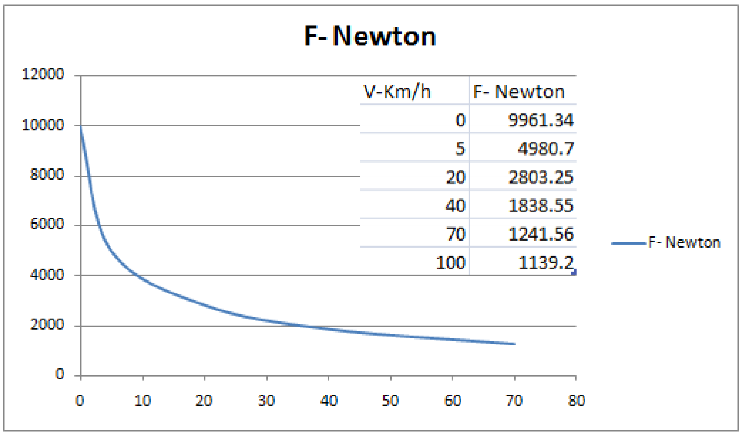


Fig15. vehicle propulsion generated in Pride equipped with CVT

16. Results and Discussion

A continuously variable transmission system was designed for a manual transmission car designed by a Korean company and manufactured in Iran, i.e. Pride 131without any changes in other power transmission components. Comparison of the manual transmission dimensions of the target car, i.e., 320 * 410 * 420 mm and the designed CVT, i.e. 394.4 * 393.7 * 245 mm, shows that the designed CVT, whose final design is shown in Figure 13, can be easily replaced by MT.

As it is shown in the 2 the whole system consists of five subsystems. Each subsystem consists of various parts for doing different functions described below in Table 10. The designed Mechanism can create the same torque ratio as manual transmission continuously. Note that the output angular speed can be calculated for each desired input angular speed by computing the torque ratio of various components or by using the speed sensor on output shaft at DMU kinematics part at CATIA software. An output angular speed versus the car speed diagram is plotted using CATIA software and shown in Figure 14 [17].

CVT also makes continuous propulsion which is transferred from engine to the wheels without falling during changes of torque ratio (Figure 15). This characteristic can eliminate the shift-shock and thus creates a better feeling while driving and also leads to reduction of fuel consumption and acceleration improvement in the target car. [17]

17. Conclusions

In this paper step by step stage of designing a CVT proposed to be used for an Iranian automaker manual transmission (MT) automobile is presented. As it is shown the required torque ratio for each vehicle can be obtained by designing an auxiliary planetary gearbox, selecting the proper belt, designing an output and idle gear and with no need to change any other components of the vehicle. In order to lubricate different parts, the new formula for designing a perforated shaft has been derived from DE-ASME Elliptic theory. The results confirm the require space needed for replacing the old transmission system with CVT with considerable fewer weight. The designed CVT considerably reduces the fuel consumption while causing a pleasure driving.

List of Symbols

b	Gears width
F_d	Force between blocks and bands
n	Number of clutch plates
N_P	Number of planetary gear teeth
N_R	Number of ring gear teeth
N_S	Number of sun gear teeth
Q_0	Initial maximum compressive force in the metal blocks
R_{CVT}	ratio of CVT pulleys
$R_{T.C}$	ratio of Torque converter
T_R	Torque on the ring gear
$T_{S.P}$	torque on the Secondary pulley

Greek symbols

α	Proportion between inner and outer diameters of shafts
β	Angle between walls of pulley
μ	Friction coefficient between blocks and pulleys
σ'	Von Mises stress

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