

Research on Designing the Continuously Variable Transmission Test Bench

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ABSTRACT

The continuously variable transmission (CVT) is designed to enhance the performance of the powertrain system. It helps the vehicle have better acceleration as well as improves the fuel economy. Fuel consumption in a car is directly linked to transmission efficiency, primarily influenced by the vehicle's drivetrain system. Factors affecting the transmission efficiency of transmission including belt structure, hydraulic pressure, and control technique are factors that need to be studied and improved. This study was carried out to calculate and design the mechanical, hydraulic, and electrical control systems of a CVT test bench. With the initially designed test bench simulated, it is possible to evaluate the characteristic quantities controlling the transmission ratio of the gearbox such as the pressure acting on the pulleys, the force acting on the belt, the torque at axes corresponding to different load modes vary from 2kW to 6 kW. The results of this study will be the basis for manufacturing an experimental system for in-depth studies and evaluation of the CVT such as belt slip, as well as the factors affecting transmission efficiency of CVT.

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1. Introduction

The Continuously Variable Transmission (CVT) is widely used in automobiles. The transmission ratio is controlled continuously to help the vehicle move to meet all road traffic load more effectively. One of the biggest factors affecting the transmission performance is the structure of the belt. A recent research about Van Doorne's CVT belt, which is a single metal belt that applies thrust to change the pulley diameter. This V-belt is characterized by high reliability and durability [1]. However, the transmission performance also needs to be studied and improved. The main cause of the reduced transmission efficiency of Van Doorne's CVT is due to excessive pressure resulting in the metal belt slipping [2]. It is the slip that leads to frictional losses in the system due to the load acting on all parts of the gearbox. Excessive pressure also reduces the lifetime of the belt element because thrust in large elements increases wear. In addition, the power and durability limitations of CVT gearboxes are limited by the strength of the steel belt and the resistance to frictional wear between the transmission elements [3]. This study will conduct calculate and design a CVT automatic transmission test bench based on input parameters including an electric motor instead of an internal combustion engine; variable transmission with steel belt (L-box). The load generator and electro-hydraulic control system are shown in Figure 1.

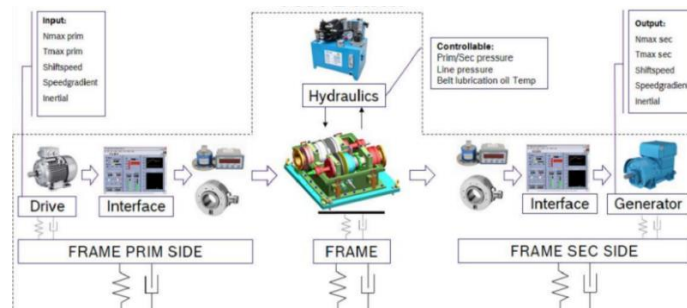


Figure 1. Test bench function layout

Description of system's operation: The drive motor works as an internal combustion engine and is controlled in operating modes through an inverter to drive the primary pulley of the CVT gearbox. The output of the gearbox (secondary pulley) is connected to the load resistance creator by a generator. The gear ratio of the CVT will be controlled through the pressure applied to the primary and secondary pulleys. In addition, the electronic control system is designed to control the operating modes at different gear ratios as well as display the power and torque values of the driving motor and the load generator. This study will calculate and design an automatic transmission test bench with different modes of changing gear ratios corresponding to the operating modes of the CVT gearbox on a specific vehicle.

2. Theoretical basis for calculating and designing the parameters of the test bench

2.1. Requirements for calculating and designing Test Bench

The test bench is designed to test the components of the CVT gearbox. The gearbox is mounted on the test bench by two connection points. The primary shaft is connected to the drive motor and the secondary shaft is connected to the load generator. The design parameters with operational requirements for the test modes are as follows:

Table 1. Design parameter table with operation requirements for test modes.

Test Condition	LOW	ELOW	EOD
Geometrical ratio	1.8	2.27	0.39
Primary torque [N.m]	135	125	125
Secondary torque [N.m]	243	283.75	48.75
Primary speed [rpm]	1750	1500	1250
Secondary speed [rpm]	972	661	3205
Engine power (Motor) [kW]	24.74	19.63	16.36

To ensure dynamic stability for the drive line, the equipment should be designed and arranged on a transmission line that is as short as possible. For dynamic testing mode, the drive motor and load generator need to have low inertia to avoid oscillation. Therefore, the drive motor and generator should be permanent magnet type, and the specifications need to be calculated, such as power, maximum torque, maximum speed, and moment of inertia [4]

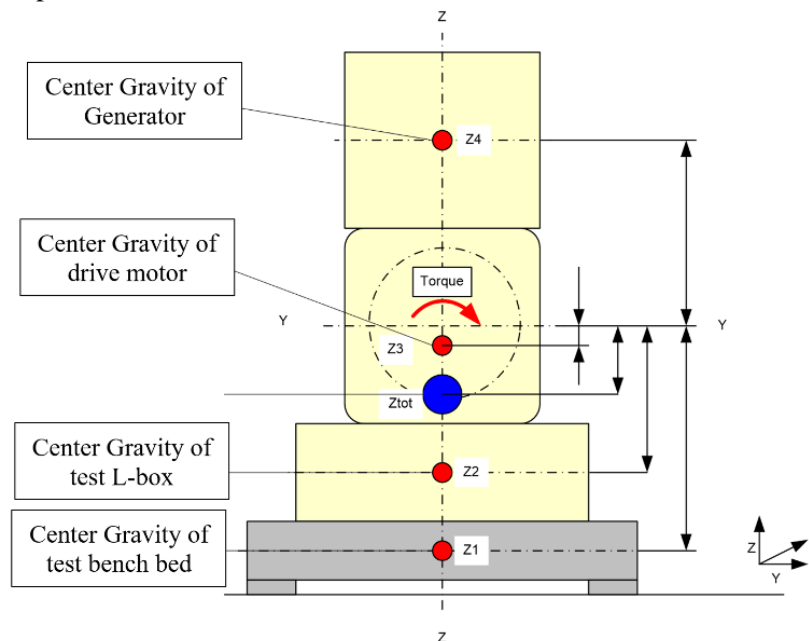


Figure 2. Center of gravity components

Calculation requirements for the center of gravity of the test rig: The center of gravity of the drivetrain (x-axis) should be as close as possible to the center of gravity of the machine. This indicates that the mass above the x-axis creates a relatively high center of gravity. The distance between the center of gravity of the test rig and the main power transmission axis should be as short as possible [5].

Design requirements for measuring devices and communication software: To measure the torque and speed of the primary and secondary shafts, the testing equipment needs to have flanges for attaching torque/speed measuring devices. Measuring the torque of the primary and secondary shafts is more accurate if they are mounted closer to the input and output shafts [5-6]

2.2. Calculation of parameters of the test bench

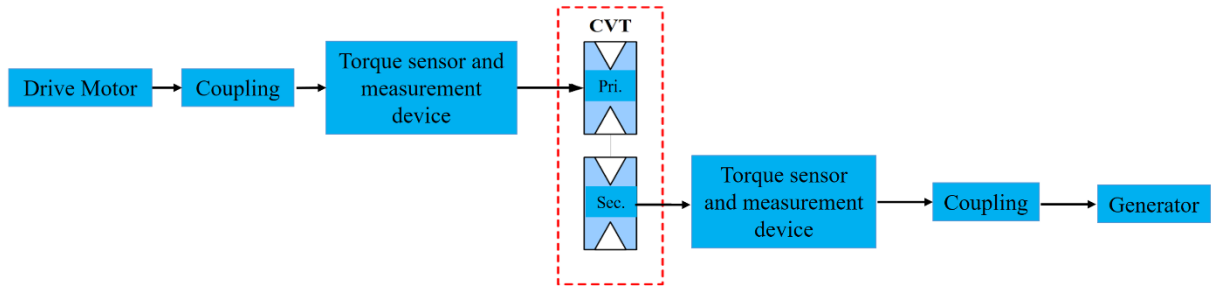


Figure 3. Components layout diagram of test bench

$$\text{Primary drivetrain influences: } T_p = J_p \cdot \alpha_p \quad \text{with} \quad \alpha_p = \frac{2 \cdot \pi \cdot a}{60} \quad (1)$$

$$\text{Secondary drivetrain influences: } T_{s(i)} = \frac{J_{sec} \cdot \frac{\alpha_p}{i} + T_{s_set} + T_{spinloss}}{i} \quad (2)$$

$$\text{Load on drive motor: } T_{drive2(i)} = T_p + T_{s(i)}; P_{drive2(i)} = \frac{T_{drive2(i)} \cdot 2 \cdot \pi \cdot \frac{np}{60}}{1000} \quad (3)$$

$$\text{Load on generator: } T_{generator2(i)} = T_{s_set}; P_{generator2(i)} = \frac{T_{generator2(i)} \cdot 2 \cdot \pi \cdot \frac{(\frac{np}{60})}{i}}{1000} \quad (4)$$

$$\text{Calculation shift speed: } Pri_speedk(i) = \left[\left[\frac{k \cdot 10^3}{2 \cdot \pi \cdot \frac{np}{60}} - (T_{s_set} + T_{spinloss}) \right] \cdot \frac{1}{J_{tot(i)}} \cdot \frac{60}{2 \cdot \pi} \right] \quad (5)$$

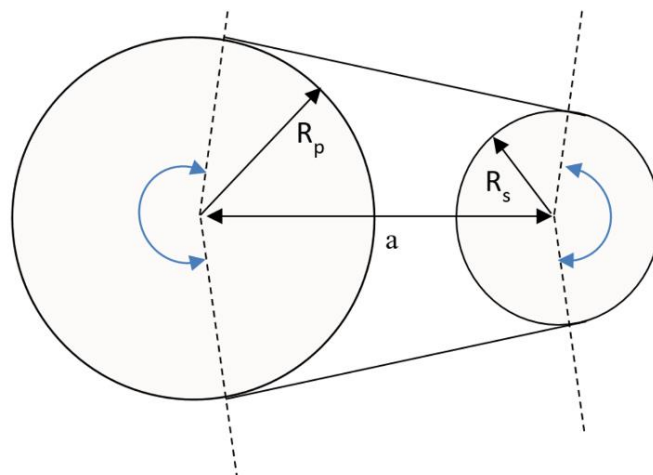


Figure 4. Belt length calculation diagram

The belt length is calculated by the equation below [7-9]:

$$l = 2a + \frac{(d_{pri} + d_{sec})\pi}{2} + \frac{(d_{sec} - d_{pri})^2}{4a}; d_{pri} = \frac{\sqrt{\left[\frac{(i+1)\pi}{2}\right]^2 + \frac{4(i-1)^2 \cdot (l-2a)}{4a}} - \frac{\pi(i+1)}{2}}{2 \cdot \frac{(i-1)^2}{4a}} \quad (6)$$

Where i is the transmission ratio; l is the belt length; a is the distance between the centers of two pulleys; d_{pri} , d_{sec} are the diameters of two pulleys. The inertia moment of drive component in the transmission system are calculated by below values.

$$J_{Drive\ motor} = 0,142; J_{Coupling} = 0,02; J_{Torque\ sensor} = 0,0065; J_{Pri.pulley} = 0,0224$$

The influence of the inertia moment of drive elements [9]:

$$J_{pri} = J_{Drive\ motor} + J_{Coupling} + J_{Torque\ sensor} + J_{Pri.pulley}; T_{pri} = (J_p \cdot \alpha_p) \cdot i \quad (7)$$

The inertia moments of driven components [9]:

$$J_{Sec.Pulley} = 0,0239; J_{orque\ sensor} = 0,0065; J_{Generator} = 0,57; T_{p_des} = 100; T_{spinloss} = 10;$$

$$T_{s_static(i)} = T_{p_des} \cdot i + T_{spinloss}$$

Impact of inertial moments of elements.

$$J_{sec} = J_{Sec.pulley} + J_{Torque\ sensor} + J_{Coupling} + J_{Generator}; T_{sec} = J_{sec} \cdot \frac{\alpha_p}{i} + T_{s_static(i)} \quad (8)$$

Torque at the shafts:

$$P = T \cdot \omega \Rightarrow T = P/\omega \quad (9)$$

Where: P (W) is the engine power; T (N.m) is the transmitted torque; ω (rad/s) is the speed of the transmission shaft. At the point where P = 10 kW and $\omega_{sec} = 1000$ rpm, the torque can be calculated:

$$T = \frac{P}{\omega_{sec}} = \frac{10 \cdot 1000}{1000 \cdot \pi/30} = 95.54 [N.m]$$

The contact force at the pulleys are calculated by following formulas [9]:

$$Fax_{Sec} = \frac{M_{Sec} \cdot \cos(11) \cdot Sf}{2 \cdot \mu \cdot \left(\frac{R_{Sec}}{1000}\right)}; Fax_{Pri} = Fax_{Sec} \cdot KpKs \quad (10)$$

Where: Fax_{pri} and Fax_{sec} are respectively the force on each pulley [N]; Sf is the safety factor, when $Sf < 1$ it will cause belt slip, The $Sf = 1.35$; μ is the coefficient of friction.

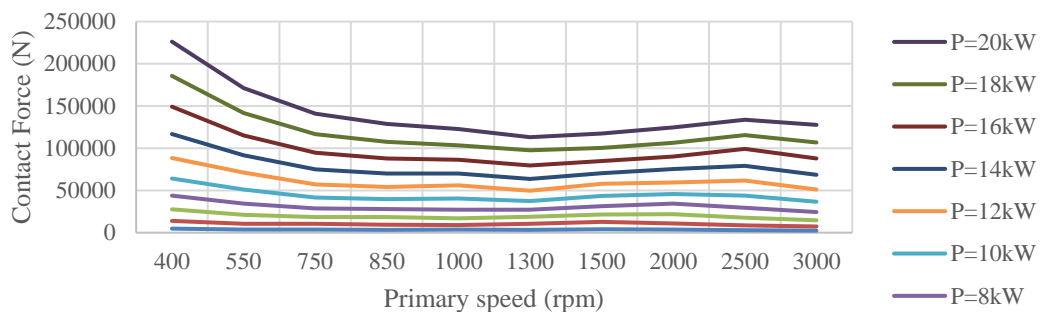


Figure 5. The graph of contact force at the primary pulley

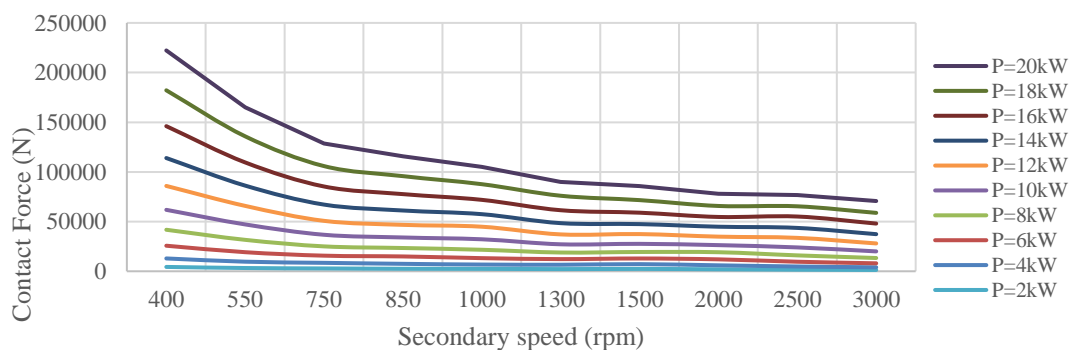


Figure 6. The graph of contact force at the secondary pulley

Hydraulic pressure at the pulleys.

The hydraulic pressure at pulleys are calculated by following equations [9].

$$P_{Pri} = \left[\frac{Fax_{Pri} - Cf_{Pri} \cdot \left(\frac{\omega_{Pri}}{1000}\right)^2}{\frac{A_{Pri}}{10000}} \right] / 100000; P_{Sec} = \left[\frac{Fax_{Sec} - Cf_{Sec} \cdot \left(\frac{\omega_{Sec}}{1000}\right)^2}{\frac{A_{Sec}}{10000}} \right] / 100000 \quad (11)$$

Where: P_{Pri} and P_{Sec} are the hydraulic pressures at the primary and secondary pulleys [bar].

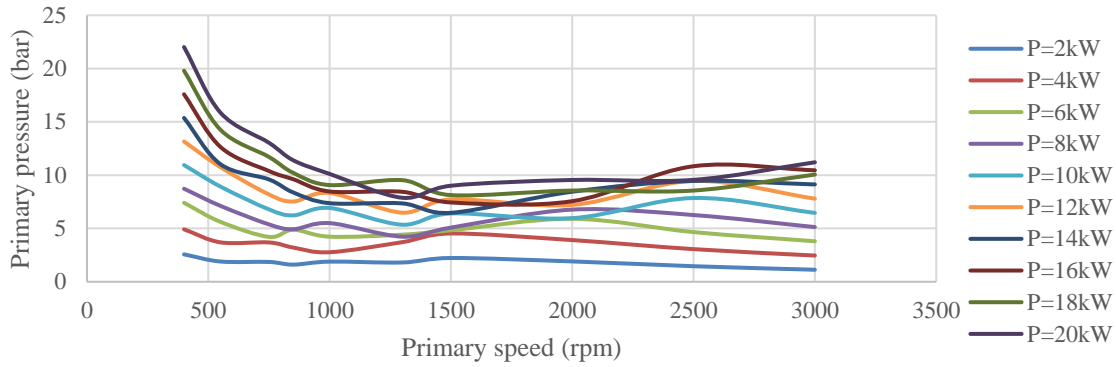


Figure 7. The graph hydraulic pressure at primary pulley

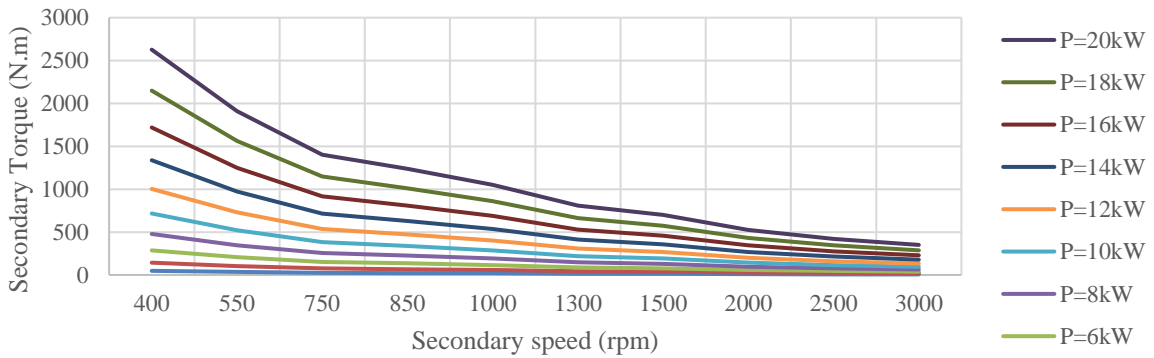


Figure 8. The graph hydraulic pressure at secondary pulley

3. Test bench design

3.1. Mechanical system

The CVT Testbench mechanical system is designed with main components including the machine bed, machine frame, drive motor supports, L-box support stand, generator load, 2 torque sensors along with bearings and associated couplings. Other additional components on the machine include sliding components, stopper bearings, rubber feet cushions, oil collection trough, display screen control panel...

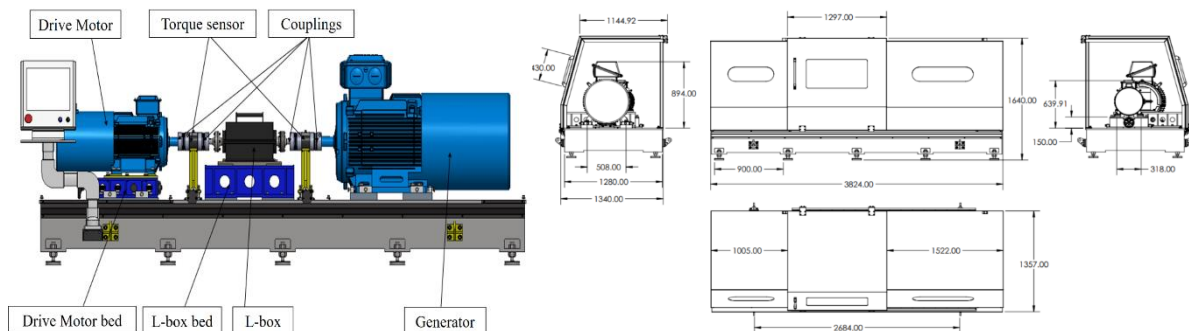


Figure 9. The Mechanical design of CVT test bench

3.2. Hydraulic system

The requirements for controlling the hydraulic system are precisely control the primary pressure line pressure within the range of 0-20 bar with a tolerance of 0.5 bar, and ensure the acceleration response and settling time within the allowable limit. The oil temperature in the system for pressure generation and lubrication at a temperature of 70°C with a tolerance of 5°C, and ensure the response time is under 60 minutes. The solenoid valves are controlled for lubrication and for opening and closing the primary and secondary pressure lines. The analog proportional valves generate accurate pressure for the primary and secondary pressure lines to prevent belt slip.

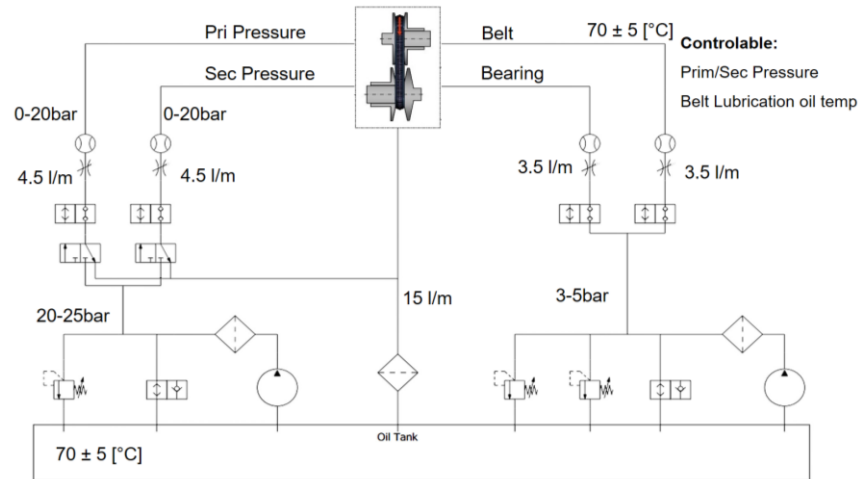


Figure 10. Hydraulic system layout

The pressure in the system is designed to include the line pressure to supply to the press chambers at the primary and secondary pulleys pressure and lubricate parts. All these pressures are adjustable. Hydraulic oil flow is designed with a stable value for each position. The fluid flow to pulleys is designed at 4.5 l/min. The oil flow to the L-Box is designed at 15 l/min. For the system to operate at high efficiency, the fluid in the system must be properly heated and cooled.

3.3. Electronic control system and data acquisition

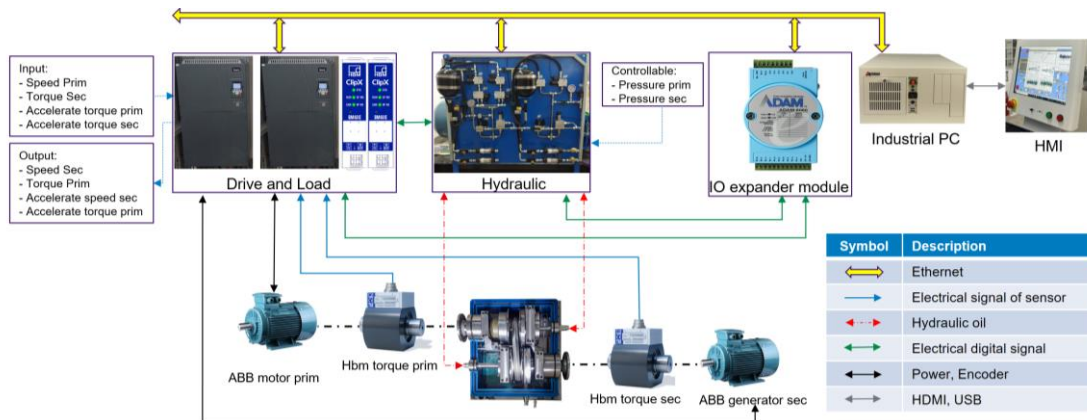


Figure 11. Electronic control system layout

The GD350A frequency converter controls the closed-loop speed and torque of the primary and secondary shafts. The Master-Slave Share Load function helps to recover energy between the two motors during testing. The PLC system is connected to the server via profinet communication, allowing for intuitive control and monitoring of the system. Changes to the torque, speed, and control mode can be easily made. The Siemens motor device with large inertia and integrated encoder is suitable for monitoring force and controlling speed. The operating process is user-friendly: the operator inputs the

required test conditions such as torque and speed from the server to the system. The server has software to send data to the system and retrieve data from the system. With the drive motor: The maximum acceleration for the drive motor to increase its speed is 500 rpm per second; Acceleration of the maximum torque increase of the drive motor is 600 N.m per second. The maximum speed is 1500 RPM. The allowable speed deviation is ± 1 [rpm]. The maximum torque $T_{max} = 200$ [N.m]. With the generator: The maximum acceleration for the generator motor to increase its speed is 500 RPM per second. Acceleration of the maximum torque increase of the Generator Motor is 700 N.m per second. The maximum speed is 3000 rpm. The allowable speed deviation is ± 1 [rpm]. The maximum torque of $T_{max} = 400$ [N.m].

3.4. Experiment and result analysis

Based on the 5 fixed ratios that were calculated and tested following the Bosch standard, which are 0.39, 0.45, 0.62, 1.8, and 2.27, and with the reference from the literature and testing methods, the test bench layout for these 5 step gear ratios has designed, which correspond to gears 10, 9, 8, 2, and 1, respectively. Combined with the remaining 5 ratios based on the average division method, the completed the design of the reference gearbox layout for the selected 10 gears were carried out. The gear shift layout design as the figure below:

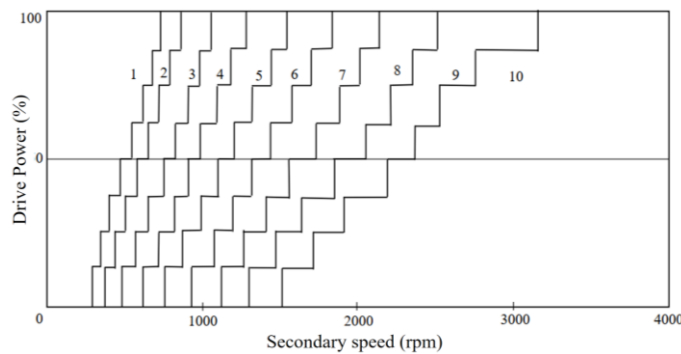


Figure 12. Transmission layout design

Based on the transmission diagram, a table of gear ratios according to motor power and secondary shaft speed can be created. Assuming at time t , $P = 8$ [kW] and $\omega_{sec} = 1500$ (rpm). When plotting this coordinate on the transmission diagram in Figure 12. t falls within the range of gear ratio 7. Similarly, for all other cases, a table of gear ratios according to the engine power and secondary shaft speed based on the transmission diagram created. After inputting the initial parameters to conduct the experiment, a comprehensive graph displaying the parameters of both the primary and secondary axis is constructed. Within each graph, 4 main parameters will be represented, including the transmission ratio, the rotational speed of the axes, the torque at the axes, and the hydraulic pressure acting on the pulleys.

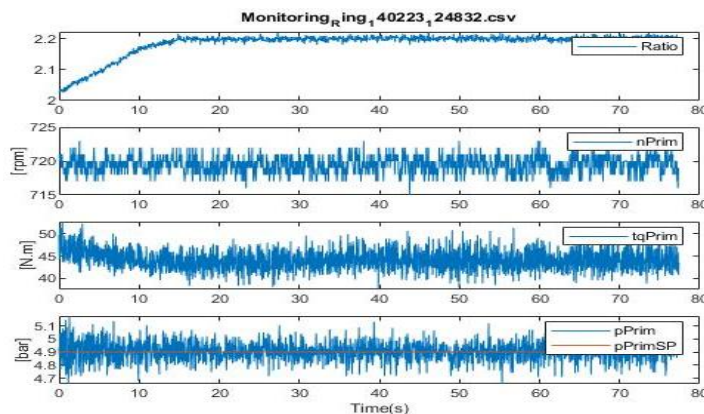


Figure 13. Graph of torque and hydraulic pressure at primary pulley with $P = 4$ kW.

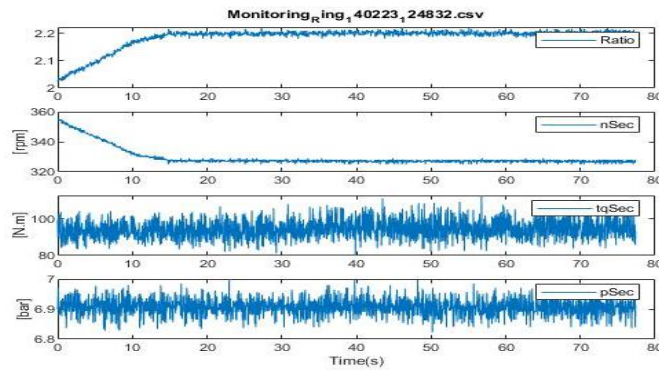


Figure 14. Graph of torque and hydraulic pressure at secondary pulley with $P = 4 \text{ kW}$.

During the time interval from 0 to 20 second when the torque value is set $T_{Sec} = 95.54 \text{ [N.m]}$, the ratio will increase from 2.04 to 2.2, while the speed of the second shaft will decrease from 360 rpm to 325 rpm. With the input parameters set to $P_{Pri} = 4.92 \text{ [bar]}$, $P_{Sec} = 6.93 \text{ [bar]}$, $\omega_{Pri} = 720 \text{ [rpm]}$, $T_{Sec} = 95.54 \text{ [N.m]}$, the actual transmission ratio of the transmission is $i = 2.2$, while the theoretical calculated value is $i = 1.8$, and the actual speed of the second shaft is $\omega_{Sec} = 325 \text{ [rpm]}$, while the calculated value is $\omega_{Sec} = 400 \text{ [rpm]}$. There is a significant deviation between the theoretical calculated value and the experimental results in the real model.

Similarly, with the rotation speed $\omega_{Sec} = 750 \text{ [rpm]}$, with the input parameters set to $P_{Pri} = 3.96 \text{ [bar]}$, $P_{Sec} = 4.51 \text{ [bar]}$, $\omega_{Pri} = 787.5 \text{ [rpm]}$, $T_{Sec} = 50.96 \text{ [N.m]}$, the actual transmission ratio of CVT is $i = 1.54$, while the theoretical calculated value is $i = 1.05$, and the actual speed of the second shaft is $\omega_{Sec} = 507 \text{ [rpm]}$, while the calculated value is $\omega_{Sec} = 750 \text{ [rpm]}$. There is a large deviation between the theoretical calculated value and the experimental results in the real model.

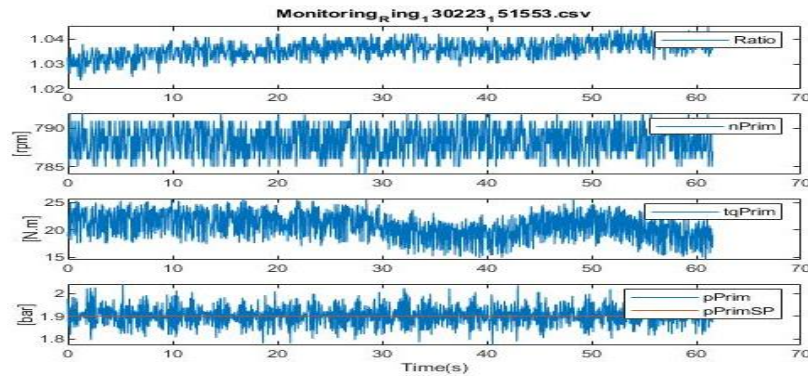


Figure 15. Graph of torque and hydraulic pressure at primary pulley with $P = 2 \text{ kW}$.

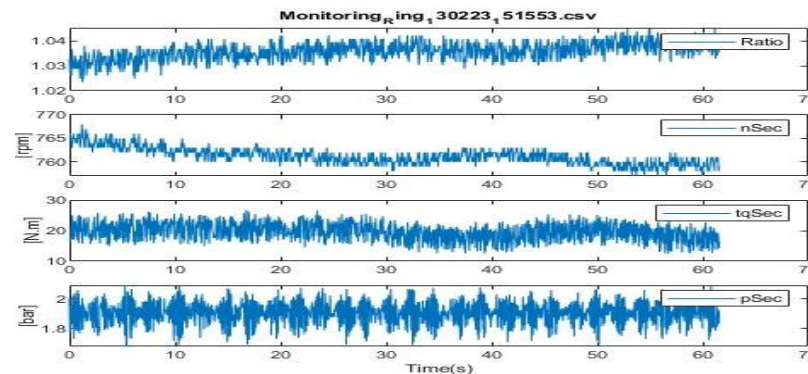


Figure 16. Graph of torque and hydraulic pressure at secondary pulley with $P = 2 \text{ kW}$.

According to the graphs from figures 13 to 16 with the input parameters set as $P_{Pri} = 1.89 [bar]$, $P_{Sec} = 1.89 [bar]$, $\omega_{Pri} = 789 [rpm]$, $M_{Sec} = 19.11 [N.m]$, the actual transmission ratio of the CVT is $i = 1.03$ while the theoretical value is only $i = 0.789$, the actual rotational speed of the secondary shaft is $\omega_{Sec} = 760 [rpm]$, while the calculated value is $\omega_{Sec} = 1000 [rpm]$. There continues to be a large discrepancy between the theoretical values and the experimental results at the actual model.

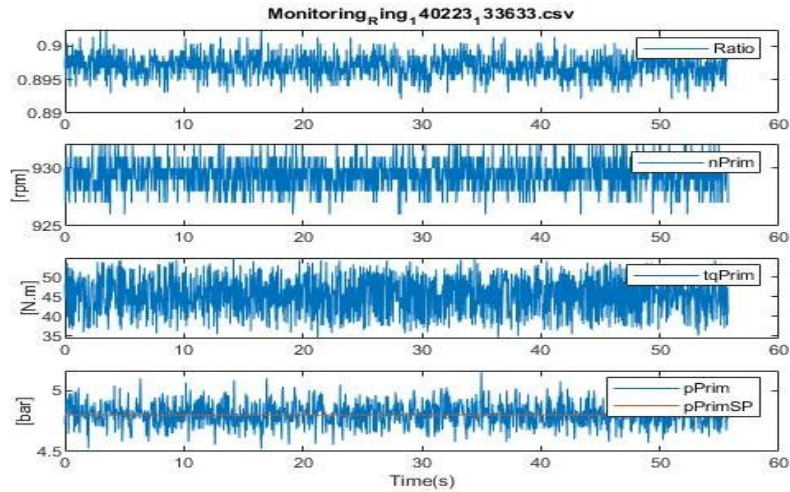


Figure 17. Graph of torque and hydraulic pressure at primary pulley with $P = 6 kW$.

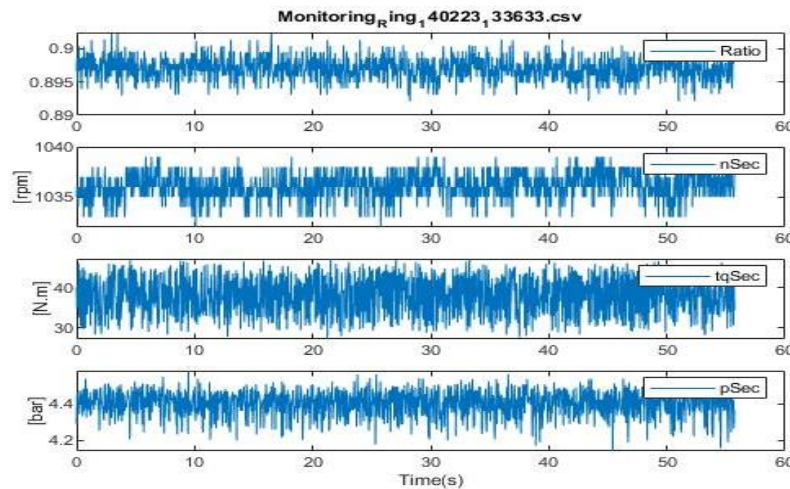


Figure 18. Graph of torque and hydraulic pressure at secondary pulley with $P = 6 kW$.

Based on the graphs from figures 17 and 18 show that for the input parameters set as $P_{Pri} = 4.83 [bar]$, $P_{Sec} = 4.39 [bar]$, $\omega_{Pri} = 930 [rpm]$, $M_{Sec} = 38.22 [N.m]$, the actual transmission ratio of the CVT is $i = 0.895$, while the theoretical value calculated is only $i = 0.62$. The actual speed of the secondary shaft $\omega_{Sec} = 1035 [rpm]$ while the calculated value is $\omega_{Sec} = 1500 [rpm]$. There is a significant discrepancy between the theoretical calculation results and the experimental results obtained from the real model.

4. Conclusions

This study has calculated, designed, and tested the CVT test bench with variable load modes. With the test bench, the parameters affecting the transmission performance of the CVT gearbox will be evaluated including the pulling torque, the resistance torque, the pressure at the pulleys, and the slip ratio of the belt. In addition, belt strength is also evaluated through the test modes of the CVT test bench.

These results will be the fundamentals for more profound research to improve the transmission efficiency of CVT and fuel consumption in automobiles.

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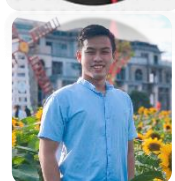
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