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Research Paper

Development of Test Cycle for Centrifugal Clutch of CVT Driven Scooters Intended for Urban Traffic Conditions

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Abstract

The present study deals with the development of a test cycle for the centrifugal clutch of continuously variable transmission (CVT) driven scooters. Centrifugal clutch experiences different duty cycles during the usage of scooters in city traffic and highways during its lifetime. Since the friction characteristics of the centrifugal clutch are controlled by acquired factors, it is difficult to predict ideal friction characteristics under all conditions. The wear of friction lining increases due to heat generated in the clutch assembly because of the repeated "stick-slip" phenomena. Therefore, an attempt has been made for developing à new test cycle by keeping the engagement frequency as a reference. Road load data for three different riding conditions have been collected and analyzed. The developed test cycle has been automated on the centrifugal clutch test bench and a new set of clutch liners was tested thoroughly. The surface roughness, thickness, and wear of clutch liners have been observed and evaluated against the field vehicles. For the surface roughness, maximum deviations of 3.74%, 3.36%, and 2.16% have been observed for trailing, middle and leading sections of clutch liners respectively. For the thickness, maximum deviations of 3.06%, 2.59%, and 3.14% have been observed for trailing, middle and leading sections of clutch liners respectively. The developed test cycle demonstrates a good correlation with field use.

Keywords: Centrifugal clutch; Road load data; Test cycle; Test bench

1. Introduction

The two-wheeler segment is one of the most important components of automobile sector. It consists of three different types viz. mopeds, scooters, and motorcycles. All modern scooters are equipped with V-belt Continuous Variable Transmission (CVT) and dry centrifugal clutch to provide ease in riding and handling [1]. This could be the main reason why scooters account one third of the total sales of two-wheelers and still growing. Centrifugal clutch is a special kind in which engagement occurs due to centrifugal force while disengagement happens through spring force. **Figure 1** shows a schematic of centrifugal clutch generally used for automotive drive train. In the centrifugal type clutch, the centrifugal force is used to generate the required force for keeping the clutch in engaged position. Additionally, friction plates, clutch plates, and pressure plates are eliminated, simplifying the clutch assembly. The advantage of the centrifugal clutch is that it does not require a clutch lever. The clutch works automatically according to the engine speed. Adequate ventilation in the clutch housing keeps the clutch at a moderate temperature. Dry running of centrifugal clutch can damage the linings as well as the clutch housing [1], [2].

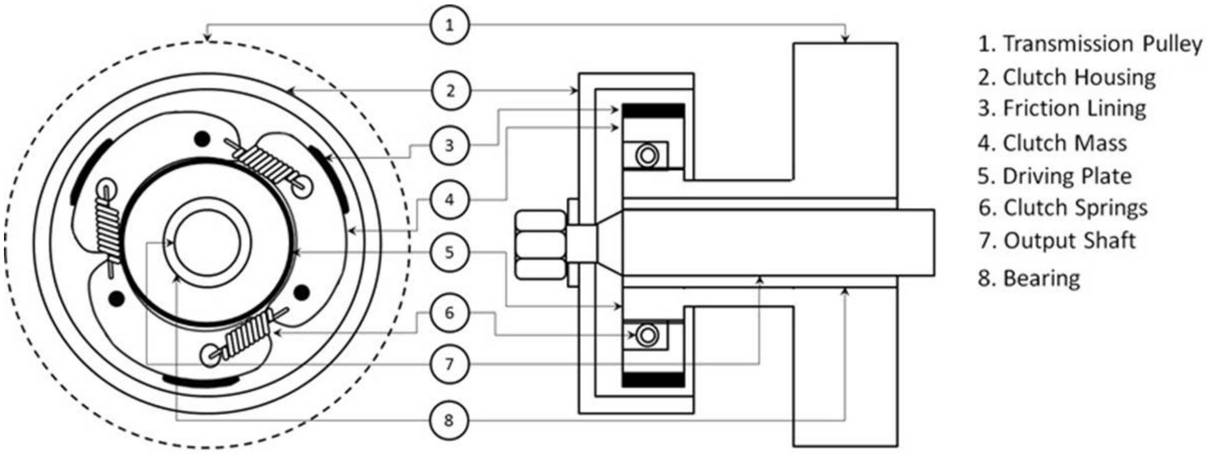


Figure 1. Centrifugal clutch

Even after extensive design considerations, centrifugal clutches play vital role in dropping transmission efficiency and so as the fuel economy of scooters [3]. Deteriorated condition of clutch liners affects the fuel economy drop with a contribution of 62.40% [4]. The interest in centrifugal clutches has been there since many decades and various researchers have presented different researches. Study related to the judder characteristics of centrifugal clutch on 125 cc scooter was carried out. A centrifugal clutch testing equipment was used to find out contributing factors to clutch judder. This study found that the deteriorated friction characteristics of clutch plays vital role in producing judder and suggested that the ideal friction characteristics of clutch are difficult to conclude as it depends on condition of use and surrounding environment [5]. Hybrid simulation model for CVT driven scooter has been developed [6]. The developed model was compared with the experimental data generated on 125 cc scooter using ECE-40 drive cycles. An empirical equation relating maximum torque transmitted by clutch and rotational speed of CVT driven pulley was obtained experimentally. The operation of clutch can be distinguished in three different modes: complete disengaged mode, slipping mode and complete engaged mode. Duration of these modes depends on wheel load and rotational speed of CVT driven pulley. Improvement in temperature rise of CVT and centrifugal clutch was obtained through design iterations for cooling system [7]. The city riding pattern was considered during the experimental validation in which maximum vehicle speed was 48 KMPH and duration was 1900 seconds. Acceleration and deceleration at regular time interval was adopted to generate clutch slippage. Improvement of 40% was achieved in air flow through CVT casing and 8% temperature drop in clutch surface after improving design of cooling system. Investigation of the performance of 18 kinds of friction liners with different material composition was carried out [8]. The relationships between ratio of static torque to dynamic torque and ratio of friction coefficient to relative velocity were established for all friction liners of centrifugal clutch. All specimens were tested for 100 clutch cycles using CVT dynamometer. Later on, same Authors elaborated the experimental procedure adopted on CVT dynamometer [9]. The clutch was accelerated from idle speed to drive speed of 500 RPM. The sliding velocity between friction liners and clutch drum was taken as decisive factor for complete engagement i.e. zero sliding velocity represented as complete engagement. One cycle was defined as rotation of clutch from complete disengagement to complete engagement. The judder test for each friction liner specimen was performed with 100 cycles. A numerical based model was developed to analyze the dynamics of centrifugal clutch [10]. During the experimental comparison, fixed clutch housing was used which rigidly connected with torque sensor. The clutch was rotated from 0 to 2000 RPM and then left free to stop due to friction with stationary housing. Simulated and experimental results were compared and analyzed. The compliant mechanism design techniques were used to analyze compliant centrifugal clutches and to develop effective new centrifugal clutch concepts [11]. The pseudorigid- body model, rigid-body replacement synthesis, force-deflection analysis, compliance potential evaluation, and compliant concept evaluation were employed in this work. The clutch designs were prototyped and tested to measure their torque-speed relationships. The setup used for testing of centrifugal clutches was on lathe machine. The clutch was rotated inside the stationary clutch drum. The output torque was measured by the reaction torque gauge mounted between the tailstock of the lathe and the clutch drum. The clutch drum was mounted on the driving shaft with bearings. Each clutch was cycled through the speed range of the lathe two to three times. Design and testing of high-torque-capacity floating opposing arm clutches was presented in which two Multi-layered Floating Opposite Arm clutches were fabricated and tested for torque-speed characteristics [12]. The test setup layout consisted of an engine as main power source with a large torque output. The engine output was attached to a jack shaft in order to provide the necessary input. The jack shaft had the clutch assembly on one end. Output of the clutch was given to the final shaft with a water dynamometer brake to load the clutch and measures the transferred torque with a torque transducer. A performance test for motorcycle centrifugal clutch was carried out using friction lining made up of composite of wood powder, coconut fiber and green clam shell powder with epoxy resin as a constituent matrix [13]. Chassis dynamometer was used to determine the power and torque on the driving wheel of CVT motorcycle. The full throttle position was applied to make the engine run with maximum power and torque output. Values of torque, power, RPM and time were recorded and used further to conclude that the clutch pads with natural composite materials have higher power and torque values. However, the power and torque loss through CVT and gearbox have not been considered. The performance verification testing for coaxial centrifugal clutch was performed using chassis dynamometer [14]. Values of torque, power and RPM were recorded and used to conclude that the performance of vehicle with coaxial centrifugal clutch was effectively improved.

Rigorous testing at design level helps to predict the particular characteristics of any component. Researches related to centrifugal clutch show well developed theoretical models for design. However, when placed on the field application, the results show drastic drop in friction characteristics of clutch liner. The gradient of the curve of friction coefficient (μ) to sliding velocity (v) drops down to negative [5][15]. The deteriorated condition of clutch friction liners encourages unwanted judder. The most fundamental way to reduce the clutch judder is to improve the friction characteristics of the clutch, thereby preventing the μ-v curve from obtaining a negative gradient. Since the friction characteristics of the clutch are controlled by acquired factors such as the riding habits and the environment, it is difficult to predict ideal friction characteristics under all conditions. During the engagement process, the centrifugal clutch liners have to work counter to the speed of engine and redirected inertia of the vehicle. This generates momentary torque peaks in the clutch as the engagement progresses and makes it to slip temporarily until full engagement of clutch. This continues until both the engine speed and output speed of clutch matches. Heat is generated in the clutch assembly because of these repeated "stick - slip" phenomena, during clutch engagement and disengagement process. This heat generated at the clutch liners and clutch housing interface, leads to wear of friction lining of the clutch. More the number of engagement, more the heat is been produced. Therefore, more effective test cycle can be developed to predict the clutch life, using number of engagement as basis. Number of engagement can be derived using Road load data (RLD). Different types of testing apparatus and test cycles for centrifugal clutch exist today which may be classified in accordance to their layout, type of test employed and level of functionality achieved. With the increasing demand of CVT driven scooters, it can be said that the development of testing methods and apparatus for centrifugal clutches has to reach a point where the different driving conditions of vehicle could be incorporated and tested. This study deals with developing a test cycle for centrifugal clutches integrating three different riding conditions. RLD has been collected and used to develop the test cycle which, later on, has been validated with experimental and field test comparisons.

2. Data Collection Method

RLD acquisition and analysis plays vital role in automobile product and component design as these data represent the parameters the vehicle experiences while traversing a road surface. A procedure was developed to process the road loads for vehicle durability design and optimization of automotive products. The theory was based on the statistical characteristics and fatigue damage equivalency techniques. Similarities and applications of the proposed road load processing procedure were illustrated with field examples [16]. Fatigue life of parabolic leaf spring was examined using past data and endurance rig test. RLD was used to collect the details about loads on the leaf spring [17]. Extensive use and analysis of RLD was implemented to compute the statistical trend in spindle loads for different variants of vehicle. The concluded data were used for deriving better spindle, suspension and frame-to-body configuration [18]. Beam element model for leaf spring was developed and simulated using Altair/Motion View software. Results from the simulated beam element leaf spring model were compared with RLD for further rectifications and improvement [19]. A statistical analysis of RLD was presented to study the effect of vehicle parameters on behavior of drive line components and reactions to road inputs [20]. The multi body dynamic modelling of vehicle was validated and simulations with parametric variants were compared with RLD. A systematic approach to derive the meaningful compressed load cycle for a vehicle from the measured RLD for fatigue simulation was presented [21]. Different instrumentation required to collect the RLD was also illustrated. Optimization and fatigue strength evaluation of parabolic leaf spring was carried out using RLD and endurance rig test [22].

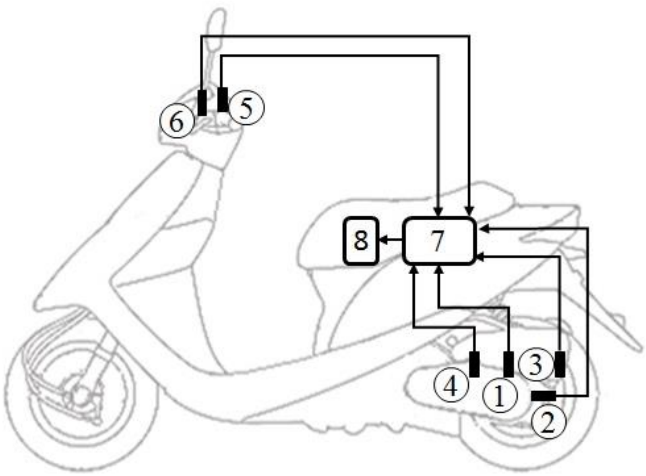
The data collection has been performed by driving CVT driven scooter in different riding conditions. A top selling vehicle from 110 cc segment has been selected as Pilot Vehicle (PV) to generate the data. A survey was performed about the day-to-day riding habits of users which concluded with following three different riding conditions. These three riding conditions have been included during the 500 kms scheduled run of PV [23].

- City riding condition with dense traffic – 60% of 500 kms
- City riding condition with open roads – 30% of 500 kms
- Highway riding condition with light traffic – 10% of 500 kms

Moreover, survey also indicated that 30% riders use their vehicle in abusive riding conditions like sudden accelerations, sudden braking, transporting heavy goods etc. These abusive riding conditions have been considered during test cycle development [24]. Effect of terrain conditions also has been taken into account during route design. As shown in **Figure 2**, PV has been instrumented with various sensors to collect following parameters in dynamic condition.

- CVT driven pulley speed (rpm)
- Clutch speed (rpm)
- Rear wheel speed (rpm)
- Air temperature inside CVT casing (°C)
- Throttle position
- Brake position

Data has been collected through DAQ system and stored in data storage device. Vehicle was thoroughly inspected to avoid errors due to component level defects before putting on to the dynamic condition. Spontaneous spikes or drastic



1. Sensor - CVT Driven Pulley Speed
2. Sensor – Clutch Speed
3. Sensor – Rear Wheel Speed
4. Sensor – Air Temperature inside CVT casing
5. Sensor – Throttle Position
6. Sensor – Brake Position
7. DAQ System
8. Data Storage Device

Figure 2. Representation of pilot vehicle with instrumentation layout

deviations in readings have been identified and rectified to analyze the collected data. The number of clutch engagement, number of throttle ON-OFF and number of braking changed depending upon the riding conditions and have been recorded accordingly. **Figure 3** shows the sample screen which has been used to count the number of stops the PV has undergone either at crossroad signals or because of traffic condition. Sample data set for 1 km run of PV is shown in **Table 1**, similar procedure has been followed for 500 kms runs.

Figure 3. Sample screen for PV1 showing vehicle speed vs time

3. Development of Test Cycle for Centrifugal Clutch

During 500 kms run for PV, data has been collected combining three different riding conditions and different terrain conditions. The first column in **Table 1** contains the point at which data has been captured and stored during ride. Second column indicates time duration for a particular instant. The frequency of throttle closing and the frequency of brake actuated have been derived through last two columns. The sensor position "1" shows open position for both throttle and brake whereas "0" position shows closed condition of throttle and functional condition for brake. The test cycle has been developed considering vehicle speed, CVT driven pulley speed, clutch speed, frequency of clutch disengagement, frequency of throttle application, frequency of braking, frequency of stopping the vehicle and time required to cover 1 km. Therefore, data set of each 1 km has been analyzed successively to develop a test cycle which must represent 1 km run of vehicle on road. Average time consumed to complete 1 km has been derived using duration of each 1 km data set.

As given in **Table 2** , analyzed results of 500 kms run for all three riding conditions have been combined to develop the test cycle for testing of centrifugal clutch. One sub-cycle represents 1 km run of vehicle. One test cycle has been derived as a set of 10 sub-cycles which is combination of 6 sub-cycles of city dense traffic, 3 sub-cycles of city open road and 1 sub-cycle of highway riding conditions. Therefore, the clutch completes the run equivalent to 100 kms when operated through 10 test cycles on centrifugal clutch test bench.

Based on the RLD data, the working of the test bench is programmed as follows: Clutch test bench has been used where the developed test cycle has been automated. The schematic of test bench is shown in **Figure 4**. The engine has been replaced with an electric motor capable of generating 8 N-m torque. Motor speed has been programmed in such a way that it can imitate the engine speeds in accordance with parameters of developed test cycle. Drive to the centrifugal clutch has been provided through CVT. The inertia equivalent to wheel load has been attached at the output shaft of clutch and could be varied to represent different load conditions. The solenoid operated disc brake has been used to stop the inertia. Operation of solenoid has been programmed to imitate the frequency of braking as per the developed test cycle. Each sub-cycle on bench has been completed in definite time period and with derived parameters which is actual representation for 1 km run of vehicle on road.

When running with city dense traffic cycle, motor starts and reaches the speed where clutch rotates with 2100 RPM. During the time period of 190 seconds, motor stops for 10 times, accordingly the clutch also engages and disengages for 10 times. The disc brake applies braking force for 6 times and stops the inertia completely for 2 times. This sub-cycle has been repeated 6 times. When running with city open road cycle, motor starts and reaches the speed where clutch rotates with 3600 RPM. During the time period of 120 seconds, motor stops for 6 times, accordingly the clutch also engages and disengages for 6 times. The disc brake applies braking force for 4 times and stops the inertia completely for 0 times. This sub-cycle has been repeated three times. When running with

Table 1. Sample data set of 1 km

Point	Duration (Sec)	Speed (km/h)	Wheel RPM	Clutch Housing RPM	Distance (km)	Throttle Position	Brake Position
1	0	0	0.00	0.00	0	1	1
2	12	12.59	94.85	853.66	0.017	1	1
3	16	16.88	127.10	1143.90	0.034	1	1
4	20	18.39	138.48	1246.34	0.051	1	1
5	23	19.98	150.41	1353.66	0.069	1	1
6	26	19.25	144.99	1304.88	0.089	1	1
7	29	17.60	132.52	1192.68	0.107	0	1
8	32	21.09	158.81	1429.27	0.123	1	1
9	34	21.41	161.25	1451.22	0.142	1	1
10	37	22.06	166.12	1495.12	0.163	1	1
11	40	19.69	148.24	1334.15	0.181	1	1
12	43	19.58	147.43	1326.83	0.197	1	1
13	47	15.33	115.45	1039.02	0.216	0	0
14	52	5.86	44.17	397.56	0.232	0	0
15	80	16.56	124.66	1121.95	0.248	1	1
16	83	20.44	153.93	1385.37	0.265	1	1
17	86	21.38	160.98	1448.78	0.281	1	1
18	89	22.03	165.85	1492.68	0.301	1	1
19	92	20.16	151.76	1365.85	0.318	1	1
20	95	22.82	171.82	1546.34	0.338	1	1
21	98	23.43	176.42	1587.80	0.357	1	1
22	101	22.28	167.75	1509.76	0.376	1	1
23	104	22.57	169.92	1529.27	0.395	1	1
24	107	23.07	173.71	1563.41	0.413	1	1
25	110	23.65	178.05	1602.44	0.433	1	1
26	114	22.85	172.09	1548.78	0.453	1	1
27	117	25.59	192.68	1734.15	0.473	1	1
28	120	24.84	186.99	1682.93	0.495	1	1
29	123	24.51	184.55	1660.98	0.515	1	1
30	126	25.48	191.87	1726.83	0.536	1	1
31	128	25.09	188.89	1700.00	0.557	1	1
32	131	23.75	178.86	1609.76	0.578	1	1
33	134	24.37	183.47	1651.22	0.598	1	1
34	137	24.91	187.53	1687.80	0.619	1	1
35	140	24.37	183.47	1651.22	0.640	1	1
36	143	24.87	187.26	1685.37	0.661	1	1
37	145	25.56	192.41	1731.71	0.676	1	1
38	148	25.63	192.95	1736.59	0.697	1	1
39	151	26.60	200.27	1802.44	0.719	1	1
40	153	27.03	203.52	1831.71	0.734	1	1
41	156	25.66	193.22	1739.02	0.756	1	1
42	159	26.53	199.73	1797.56	0.771	1	1
43	162	25.48	191.87	1726.83	0.792	1	1
44	165	26.35	198.37	1785.37	0.814	1	1
45	168	26.60	200.27	1802.44	0.836	1	1
46	171	26.67	200.81	1807.32	0.865	1	1
47	174	25.74	193.77	1743.90	0.888	1	1
48	177	22.75	171.27	1541.46	0.908	0	1
49	180	15.69	118.16	1063.41	0.923	0	0
50	222	8.45	63.69	573.17	0.942	0	0
51	226	19.29	145.26	1307.32	0.962	1	1
52	229	23.18	174.53	1570.73	0.978	1	1
53	232	23.50	176.96	1592.68	0.998	1	1
54	235	24.73	186.18	1675.61	1.018	1	1
Average	**21.6**	**162**	**1461**			**6**	**4**

Table 2. Sub-cycles for centrifugal clutch using three riding conditions

Parameter	Sub Cycle – City Dense Traffic	Sub Cycle – City Open Road	Sub Cycle – Highway
Average vehicle speed (KMPH)	20	35	52
Average time for acceleration (sec)	30	20	20
Average Clutch Speed (RPM)	2100	3600	5400
Average Wheel Speed (RPM)	230	409	610
Average No. of Braking	6	4	1
Average No. of Clutch Disengagement	10	6	1
Average No. of Stops	2	0	0
Average time for 1 km (Sec)	190	120	90
Average vehicle speed (KMPH)	20	35	52

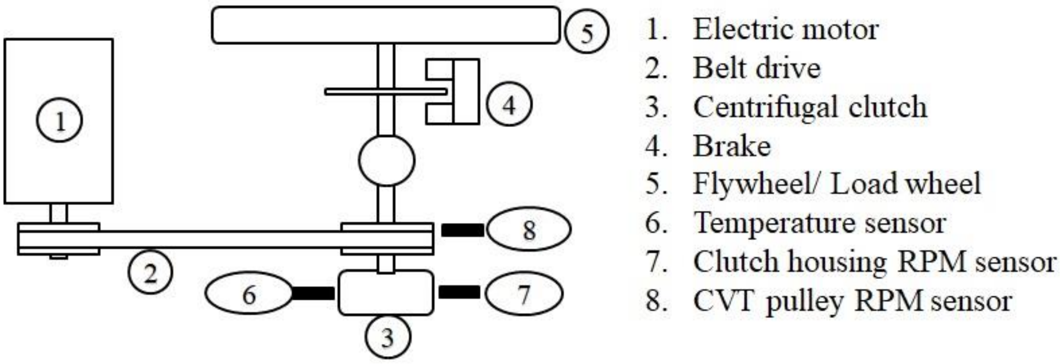


Figure 4. Schematic of centrifugal clutch test bench

highway cycle, motor starts and reaches the speed where clutch rotates with 5400 RPM. During the time period of 90 seconds, motor stops for 1 time, accordingly the clutch also engages and disengages for 1 time. The disc brake applies braking force for 1 time and stops the inertia completely for 0 times. This sub-cycle has been repeated 1 time. Actual experimental setup is shown in **Figure 5**. Test bench has been mounted with appropriate speed and temperature sensors to monitor the parameters as per the developed test cycle.

The surface roughness and thickness of centrifugal clutch liners have been measured and compared with field vehicle with the help of portable surface roughness tester and digital depth gauge respectively. The surface roughness and thickness measurement has been performed after span of 100 test cycles on test bench. The measurement of surface roughness and thickness values have been completed on field vehicles which have been run through distance close to definite service intervals of 3000 km, 6000 km,

Figure 5. Experiment Setup for centrifugal clutch test

9000 km and 12000 km. Figure 6 shows three different predefined test sections of the clutch liner at which the surface roughness and thickness have been measured for further analysis. The clutch liner wear has been derived using the difference of clutch liner thickness at 0 km and at definite intervals of 3000 km, 6000 km, 9000 km, 12000 km and 14000km.

Figure 6. Test-sections of the clutch liner

The experiment run has been started with new pair of clutch liners on centrifugal clutch test bench. The surface roughness and thickness of new clutch liners have been measured and recorded. As per the workshop manual provided by the manufacturer, clutch has to be serviced after every 3000 km. The last permissible thickness of clutch liner friction face is 2 mm after which the liners have to be replaced. Therefore, the new clutch liners have been tested up to last permissible thickness of 2 mm without any service operations. The clutch went through 1400 test cycles on clutch test bench which is equivalent to 14000 km run of vehicle on road. Clutch liner surface roughness and thickness measurements for three test sections have been recorded after each 100th test cycle.

4. Results and Discussion

Surface roughness is an important parameter to characterize the clutch [25]. Figure 7 shows the surface roughness behavior of clutch liner which went through 1400 test cycles. Minor reduction in surface roughness of the trailing section of clutch liner has been observed up to 300th test cycle. Then after, linear behavior in reduction of surface roughness has been witnessed up to 1400th test cycle for trailing section. The middle section and leading section show similar trend for decrease in surface roughness. Surface roughness for middle and leading sections shows drastic reduction up to 1000th test cycle after which surface roughness of both test sections show low variations.

Figure 8 shows the variation in clutch liner thickness during the course of 1400 test cycles. Liner behavior in thickness reduction has been observed for all three test sections of clutch liners as correlation coefficient for all three test sections is 0.99. Equal reduction in thickness reduction has been observed for middle and leading section after 600th test cycle. Equal reduction in thickness has been witnessed for all three test sections after 900th test cycle.

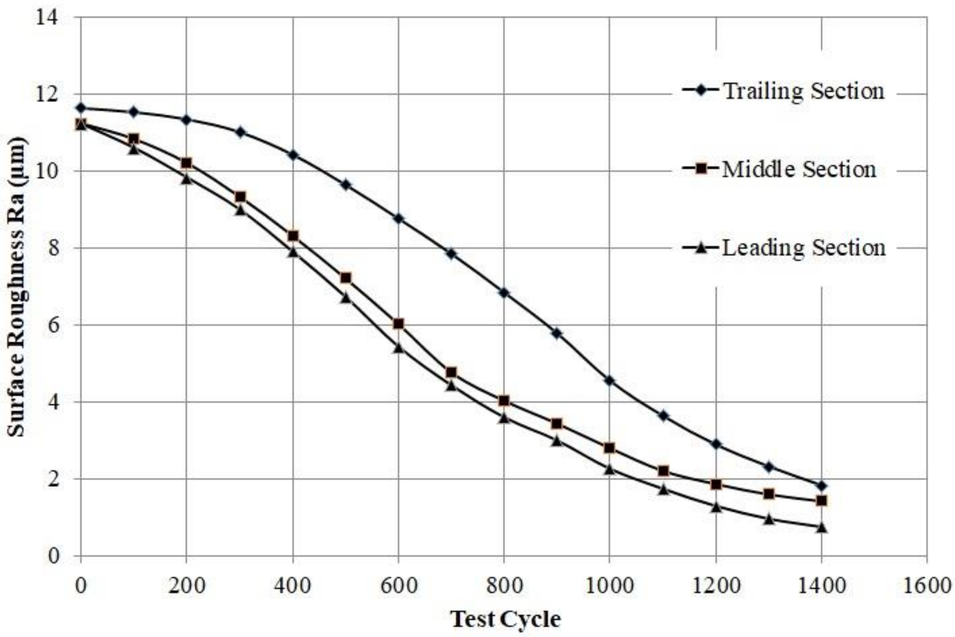


Figure 7. Variation in average surface roughness for three test sections of clutch liner

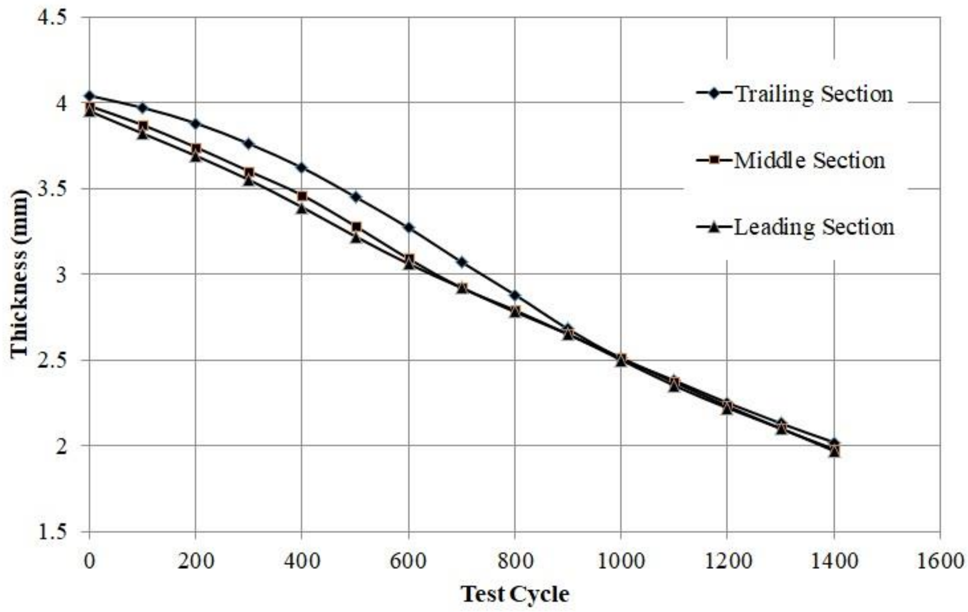


Figure 8. Variation in thickness for three test sections of clutch liner

Two batches of field vehicles have been identified for clutch liner surface roughness and thickness measurements. Vehicles have been selected from authorized service station. Vehicles driven through 3042 km, 5974 km, 9063 km, 12152 km and 14430 km have been included in Vehicle batch - 1 (VB-1). Vehicles driven through 3136 km, 6113 km, 9121 km, 11971 km and 14088 km have been included in Vehicle batch - 2 (VB-2). Due care has been taken to assure that none of these 10 vehicles went through a replacement of any transmission component and they are still running with original components. The surface roughness and thickness measurements have been performed on field vehicles before definite service intervals. Average of these measurements has been used for further comparisons. **Table 3** indicates values of surface roughness and **Table 4** indicates thickness measured on field vehicle clutch liners.

Table 3. Surface roughness for field vehicle clutch liners

Batch-1				Batch-2			
Vehicle Distance (km)	Surface Roughness Ra (μm) for Batch-1 Clutch Liners			Vehicle Distance (km)	Surface Roughness Ra (μm) for Batch-2 Clutch Liners		
	Trailing Section	Middle Section	Leading Section		Trailing Section	Middle Section	Leading Section
3042	10.92	8.834	8.1	3136	10.712	8.35	8.044
5974	9.23	6.325	5.647	6113	8.858	6.097	5.375
9063	6.145	3.814	3.108	9121	5.777	3.234	2.962
12152	2.88	1.78	1.214	11971	3.024	2.025	1.402
14430	1.709	1.395	0.696	14088	1.981	1.487	0.814

Table 4. Thickness for field vehicle clutch liners

Batch-1				Batch-2			
Vehicle Distance (km)	Thickness (mm) for Batch-1 Clutch Liners			Vehicle Distance (km)	Thickness (mm) for Batch-2 Clutch Liners		
	Trailing Section	Middle Section	Leading Section		Trailing Section	Middle Section	Leading Section
3042	3.80	3.60	3.52	3136	3.64	3.48	3.42
5974	3.35	3.22	3.18	6113	3.05	2.82	2.78
9063	2.75	2.72	2.72	9121	2.47	2.44	2.44
12152	2.25	2.25	2.23	11971	2.13	2.09	2.09
14430	2.00	1.95	1.92	14088	1.92	1.91	1.90

Comparison of surface roughness measurement has been shown in **Table 5**. Positive deviations have been witnessed between field vehicle clutch liners and test bench clutch liners. Maximum deviation of 3.74% has been observed for trailing section at 3000 km which gradually decreases with distance and reaches to minimum of 2.27% at 12000 km. Maximum deviation of 3.36% has been observed in roughness of middle section at 3000 km which decreases and reaches to minimum of 2.26% at 12000 km. Maximum deviation for leading section is 2.16% at 3000 km and minimum deviation is 1.45% at 12000 km. At 14000 km, the deviation of 1.03%, 2.15% and 1.19% have been observed for trailing, middle and leading sections respectively.

Comparison of thickness measurement has been shown in **Table 6**. Negative deviations have been witnessed between field vehicle clutch liners and test bench clutch liners. Minimum deviation of 1.08% has been observed for trailing section at 3000 km which gradually increases with distance and reaches to maximum of 2.74% at 12000 km. Minimum deviation of 1.69% has been observed in thickness of middle section at 3000 km which increases and reaches to maximum of 2.76% at 12000 km. Like trailing and middle sections, similar trend for thickness deviation has been observed for leading section. The minimum deviation for leading section is 2.31% at 3000 km and maximum deviation is 2.78% at 12000 km. At 14000 km, the deviation of 3.06%, 2.59% and 3.14% have been observed for trailing, middle and leading sections respectively.

Trend of clutch liner wear can be used to compare the test bench results with field results [26]. Wear data for test bench clutch liners have been compared with wear data of field vehicle clutch liners as shown in **Figure 9**. Higher wear for VB-2 clutch liners than the test bench clutch liners has been witnessed throughout 1400 test cycles. This variance is under acceptable limit and could be the result of acquired factors such as the riding habits, terrain conditions, traffic conditions and the environment. However, similar trend under wear has been observed for VB-1 clutch liners. This provides a convincing test cycle which has been developed to imitate the clutch life, using number of engagement as basis. Condition of clutch friction liners after 1400 test cycles on test bench and equivalent field driven vehicle clutch liners have been shown in **Figure 10**.

Table 5. Surface roughness for test bench clutch liners and vehicle clutch liners

No. of Test Cycle	Equivalent Distance (km)	Surface Roughness Ra (μm) for Test Bench Clutch Liners			Surface Roughness Ra (μm) for Vehicle Clutch Liners (Average for VB-1 and VB-2)		
		Trailing Section	Middle Section	Leading Section	Trailing Section	Trailing Section	Middle Section
300	3000	10.998	9.319	8.991	10.816	8.592	8.072
600	6000	8.753	6.009	5.427	9.044	6.211	5.511
900	9000	5.774	3.43	2.991	5.961	3.524	3.035
1200	12000	2.885	1.857	1.289	2.952	1.900	1.308
1400	14000	1.826	1.410	0.746	1.845	1.441	0.755

Table 6. Thickness for test bench clutch liners and vehicle clutch liners

No. of Test Cycle	Equivalent Distance (km)	Thickness (mm) for Test Bench Clutch Liners			Thickness (mm) for Vehicle Clutch Liners (Average for VB-1 and VB-2)		
		Trailing Section	Middle Section	Leading Section	Trailing Section	Trailing Section	Middle Section
300	3000	3.76	3.6	3.55	3.72	3.54	3.47
600	6000	3.27	3.09	3.06	3.20	3.02	2.98
900	9000	2.68	2.65	2.65	2.61	2.58	2.58
1200	12000	2.25	2.23	2.22	2.19	2.17	2.16
1400	14000	2.02	1.98	1.97	1.96	1.93	1.91

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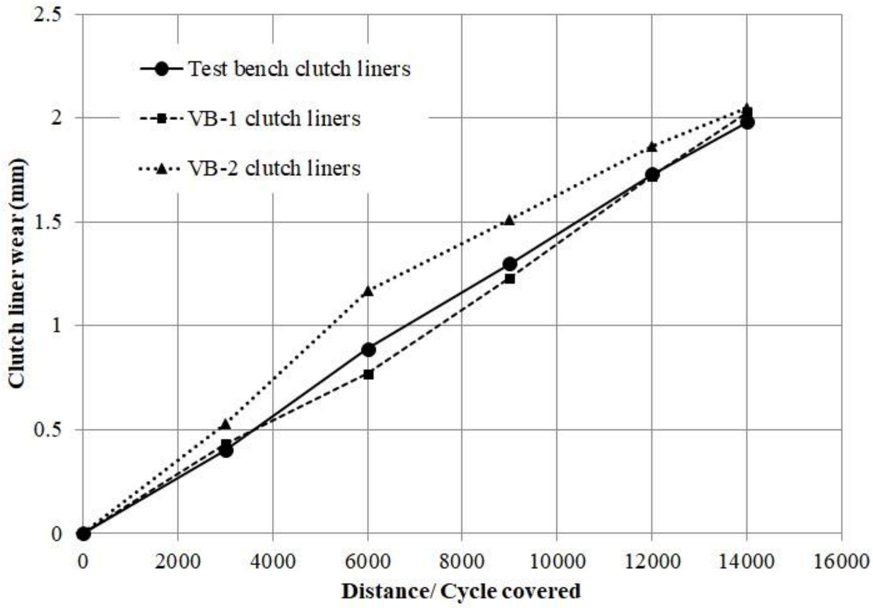

Figure 9. Wear of vehicle clutch liners and test bench clutch liners

Test bench clutch liner VB-1 clutch liner VB-2 clutch liner

Figure 10. Clutch liners after 1400 test cycles on test bench and equivalent distance on vehicle

5. Conclusion

In this study, test cycle for centrifugal clutch has been developed using number of engagement as basis. The road load data collection was performed by driving 110 cc CVT driven scooter in different riding conditions. Analyzed road load data of 500 kms run for all three riding conditions was combined to develop the test cycle for centrifugal clutch. Clutch test bench was used where the developed test cycle was automated. The surface roughness and thickness of centrifugal clutch liners were measured and compared with field vehicle driven through equivalent distance. When compared for the surface roughness, maximum deviations of 3.74%, 3.36% and 2.16% have been observed for trailing, middle and leading sections of clutch liners. When compared for the thickness, maximum deviations of 3.06%, 2.59% and 3.14% have been observed for trailing, middle and leading sections of clutch liners. Similar trend has been observed for wear data of test bench clutch liners and wear data of vehicle clutch liners. The developed test cycle demonstrates good correlation with field use, in terms of surface roughness values, clutch liner thickness, wear and condition of clutch friction lining. Modification in centrifugal clutch design or any specific field issue related to clutch can be assessed using proposed test cycle on the clutch test bench. However, compatibility of developed test cycle can be studied further by selecting more numbers of field vehicles from different geographical regions.

Author's Declaration

Authors' contributions and responsibilities

The authors made substantial contributions to the conception and design of the study. The authors took responsibility for data analysis, interpretation and discussion of results. The authors read and approved the final manuscript.

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