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ORIGINAL ARTICLE

Experimental Investigation on Performance Characteristics of Dry Centrifugal Clutch with Grooved Friction Liners

Dhruv Panchal[1,*], Bhavesh Patel[1] and Hardik Gohil[2]

[1]Mechanical Engineering Department, L. D. College of Engineering, Ahmedabad 380015 Gujarat, India
[2]International Automotive Engineering, Technische Hochschule Ingolstadt, Esplanade 10, 85049 Ingolstadt, Germany

ABSTRACT – The deteriorated condition of friction liners after prolonged use is one of the primary causes of judder in centrifugal clutches. The friction characteristics can be retained by generating specific textures or grooves on the friction liner. An attempt has been made to study the characteristics of centrifugal clutch using grooved friction liners. A test cycle for centrifugal clutch has been developed using a number of engagements as the basis. A vehicle test bench was used for the experiment where the developed test cycle was automated. The performance characteristics of the centrifugal clutch have been recorded and analyzed with normal friction liners and grooved friction liners for 100 test cycles. For this study, the groove area ratio was retained at 0.15, and the grooves were cut at 90°. After completing 100 test cycles, the clutch with a grooved friction liner exhibited better characteristics. After completing 100 test cycles, the surface roughness reduction at the leading section of the grooved friction liner and normal friction liner has been found to be 6.44% and 8.11%, respectively. The thickness reduction at the leading section of the grooved friction liner and normal friction liner has been reported to be 3.73% and 4.98%, respectively. Throughout the run of 100 test cycles, the higher clutch housing temperature has been witnessed in the case of a clutch with a grooved friction liner. At the 100th test cycle, the clutch torque with a grooved friction liner was 15.22% more than the clutch torque with a normal friction liner. Even after prolonged use, the clutch with grooved friction liner exhibited better judder characteristics and also provided higher fuel economy for vehicles.

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INTRODUCTION

In Continuous Variable Transmission (CVT) driven scooters, the power and torque developed by the engine crankshaft are transmitted to the driving wheel through three units; CVT, centrifugal clutch and reduction gearbox [1]. CVT and centrifugal clutch transmit the power and torque using friction, while the reduction gearbox makes use of spur gear pairs. The centrifugal clutch comprises a clutch carrier, friction liners (shoe), retaining springs and clutch housing [1]–[3]. Working of the centrifugal clutch can be divided into three regions: disengagement, sliding and engagement. During the engagement process, the friction liners of the centrifugal clutch 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 slip temporarily until full engagement of the clutch. This process continues until the clutch housing speed matches the CVT-driven pulley speed [4]–[6]. Heat is generated in the clutch assembly because of these repeated "stick-slip" phenomena during the clutch engagement and disengagement process. This heat generated at the clutch friction liners and clutch housing interface leads to the wear of friction lining of the clutch.

Detailed theoretical models for centrifugal clutch have been presented to estimate torque transmission capacity [7], [8]. Present studies are well proven to predict and optimize the material selection for important components of the transmission system, i.e. centrifugal clutch [9]–[11]. Rigorous testing at the design level helps to predict the particular characteristics of any component. However, when placed on the field application, the results show a drastic drop in friction characteristics of the friction liner [12]. During engagement, degraded characteristics of friction liners tend to increase judder [7], [13]. The judder propensity of friction liner increases when the gradient of the curve of friction coefficient (μ) to sliding velocity (V_r) drops down to negative [3], [14]. The most fundamental way to reduce the clutch judder is to improve the friction characteristics of the clutch, thereby preventing the μ-V_r curve from obtaining a negative gradient. Since the friction characteristics of the clutch are controlled by acquired factors such as riding habits and the environment, it is difficult to predict ideal friction characteristics under all conditions. Further, centrifugal clutches play a vital role in dropping transmission efficiency and so as the fuel economy of CVT scooters [15]. The deteriorated condition of clutch friction liners affects the fuel economy drop with a contribution of 62.40% [12].

The practical analysis reveals that the majority of clutch friction lining early failures are caused by high surface temperatures generated during slipping [16]–[21]. To avoid premature failures, it's important to know the temperature distribution of the contact surfaces and the quantity of energy dissipated during sliding. The friction characteristics of liners are affected by the load condition, physical qualities, liner dimensions, and degree of air cooling [22]–[28]. The

*CORRESPONDING AUTHOR | D. Panchal | dhruvpanchal@ldce.ac.in

application of specific textures or grooves to a sliding surface, which consists of flat sections interrupted by cavities, is a useful way to retain the friction characteristics of sliding contacts [29], [30]. The grooves provided on the friction area help to increase the heat transfer through convection. This increase in heat transfer through convection lowers the internal energy of the friction surface, resulting in a longer life of friction clutch [31], [32]. Moreover, the grooved surface with specified geometry makes it simpler for worn debris to escape from the contact zones into the grooves, reducing irregularities and, so as a result, squeal and vibration tendencies [30], [33], [34].

The design of centrifugal clutch using different friction materials and their effects on clutch judder have been extensively studied [4]–[8], [13]. Comparative studies on judder between new clutch and used clutch are also available [3]. Very little literature is available regarding the retention of friction characteristics of centrifugal clutch using grooves on friction liners. Moreover, limited studies have been observed on the effect of grooved friction liners on vehicle fuel economy and vibrations. Therefore, more focus is required to examine the behavior of centrifugal clutch using grooved friction liners. In the present work, the performance of centrifugal clutch using grooved friction liners has been measured and compared with the normal friction liners. A dedicated vehicle test bench was used to automate the test cycle for the centrifugal clutch. Tests under three different riding conditions were conducted. Measurements for various parameters have been recorded and analyzed. The influence of grooves on the performance characteristics of friction liners has been examined.

EXPERIMENTAL SET-UP

The experiments, which involved automating the designed test cycle, were carried out on a special vehicle test bench. Figure 1 depicts the schematic of the test bench. The specially developed test cycle was automated on this test bench [35]. The throttle was controlled by a stepper motor and programmed to produce engine speeds in accordance with the specifications of the created test cycle. A second stepper motor was also used to automate the rear brake. This stepper motor's operation was set to provide the frequency of braking in accordance with the created test cycle. The computer system was used to programme and control both stepper motors. The load wheel to generate inertia equivalent to road load was attached to the output shaft of the roller and could be varied to represent different load conditions. Each sub-cycle on the bench was completed in a definite time period and with derived parameters which was actual representation for a 1 km run of the vehicle on road.

When running with city dense traffic cycle, the stepper motor operates the throttle and increases the engine speed up to the point where the clutch rotates with 2100 rpm. In the course of 190 seconds, the stepper motor stops ten times, which causes the clutch to engage and disengage ten times as well. The rear brake applies braking force six times and stops the load wheel completely two times. This sub-cycle has been repeated six times. The stepper motor controls the throttle during city open-road operation, increasing engine speed until the clutch turns at 3600 rpm. In the course of 120 seconds, the stepper motor stops six times, which causes the clutch to engage and disengage six times as well. The rear brake applies braking force four times but does not stop the load wheel completely. This sub-cycle has been repeated three times. The stepper motor controls the throttle when operating on the highway cycle, increasing engine speed until the clutch turns at 5400 rpm. The clutch engages and disengages once during the 90-second time period when the stepper motor stops once. While applying braking pressure once, the rear brake does not entirely stop the load wheel. There has been one repetition of this sub-cycle.

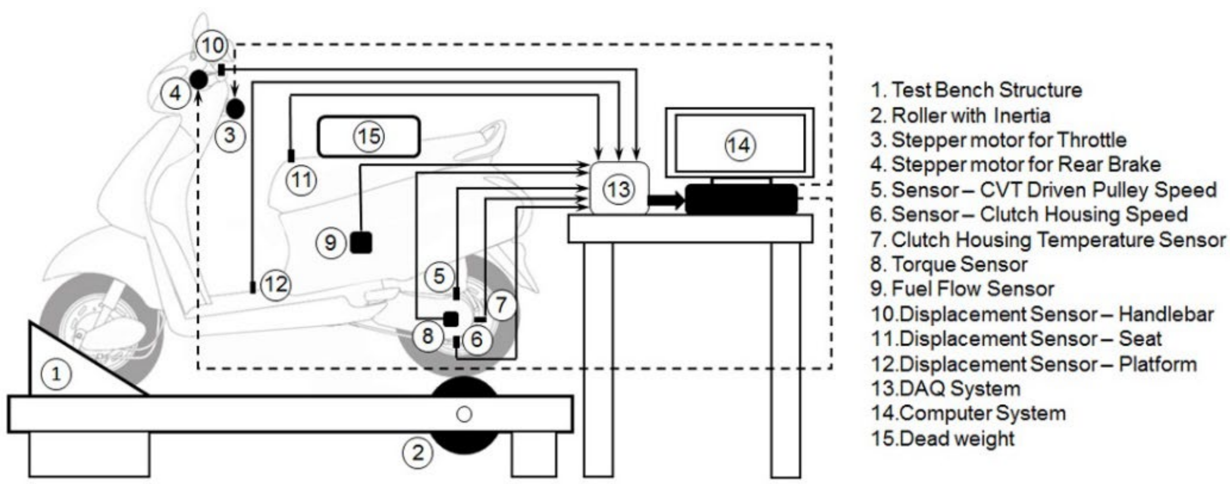


Figure 1. Schematic of the vehicle test bench with instrumentation layout

The test bench was mounted with appropriate sensors to monitor and measure the parameters as per the developed test cycle. Two proximity speed sensors (Make: Autonics, Model: CR18-8DN) were used to measure the rpm of CVT-driven pulley and clutch housing. An infrared temperature sensor (Make: Melexis, Model: MLX90614) was mounted to measure the surface temperature of the clutch housing. Eddy current-type displacement sensors (Make: Micro-Epsilon, Model: eddyNCDT-3300) were mounted at three different positions on the vehicle [3], i.e. handlebar, seat and platform. These sensors were mounted to study the amount of displacement observed at a particular location when the vehicle is subjected to a clutch judder. To measure the output torque, a dynamic torque sensor (Make: Adi Artech, Model: TTD-SS) was assembled with the output shaft of the centrifugal clutch. A positive displacement type flow sensor (Make: Achievers, Model: CE-121) was used to measure the fuel consumption during the test. The fuel economy tests were carried out as per IS 10881:1994 with a gravity-fed fuel system and at atmospheric conditions. It was assumed that the fuel is at atmospheric temperature.

The relative sliding velocity V_r between the friction liner and clutch housing has been evaluated as:

$$V_r = R(\omega_c - \omega_h) \tag{1}$$

where, R denotes the outer radius of friction liner, and ω_c and ω_h represent the angular velocities of the clutch and the clutch housing, respectively [8]. The output torque T_C of the centrifugal clutch can be evaluated by:

$$T_C = nRbF_T \tag{2}$$

where n represents the number of clutch shoes, R represents the outer radius of friction liner, and b denotes the width of the contact strip on the friction liner. The F_T is the tangential force on the friction liner, which is a product of the coefficient of friction μ and normal force on the friction liner F_N [7]. Therefore, Eq. (1) can be written as:

$$T_C = nRb\mu F_N \tag{3}$$

The surface roughness and thickness of friction liners were measured with the help of a portable surface roughness tester and digital depth gauge, respectively. The surface roughness and thickness measurements were performed after a span of 10 test cycles. As shown in Figure 2(a), the measurement for surface roughness and thickness of the friction liner were performed at three different predefined test sections. The frictional characteristics greatly depend on the size and density of the grooves [36]–[39]. Existing research shows that the groove area ratio (GR = groove area / total contact area) at 0.15 is beneficial in terms of contact pressure, heat flux, and wear characteristics [29], [31], [40]. Moreover, the groove at 45° or 90° on the friction surfaces showed great potential for squeal and vibration reduction [30]. Thus, in the present work, the GR has been retained at 0.15, and the grooves have been cut at 90° as shown in Figure 2(b). Overall contact area has been kept equal on both types of friction liners.

A top-selling vehicle from the 110 cc segment was selected for the performance measurement of centrifugal clutch. The experiment run was started with new pair of friction liners. First, the performance characteristics were measured with normal friction liners for 100 test cycles, followed by the measurement of performance characteristics with grooved friction liners. The run of 100 test cycles imitates a 1000 km run of the vehicle on road.

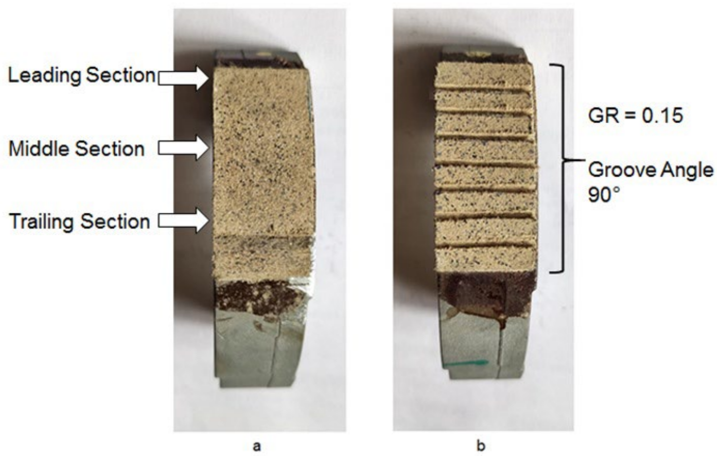


Figure 2. (a) Normal friction liner and (b) grooved friction liner

RESULTS AND DISCUSSION

Engagement Characteristics

The rotational speeds of the CVT-driven pulley and clutch housing, the output torque, clutch housing temperature and fuel consumption were acquired for the comparative study. Rotational speeds of the CVT-driven pulley and clutch housing have been used to evaluate relative sliding velocity V_r. The measured torque has been used to evaluate the coefficient of friction, μ. For the centrifugal clutch under test, Figure 3 illustrates the engagement characteristics with a normal friction liner in which three engagement parameters have been considered for a time period of 10 sec. At the idle speed, the CVT-driven pulley was rotating at about 415 rpm. The rotational speed of CVT driven pulley began to rapidly increase with engine throttling, which in turn generated centrifugal force and moved the clutch shoes outward. The engagement of clutch shoes with clutch housing was initiated at around 1027 rpm and 4.25 sec. At around 8.5 sec, a full engagement was attained when the clutch housing had a rotational speed equal to that of the CVT-driven pulley. In a very short time after the beginning of the engagement, the clutch torque had increased sharply to about 6.5 N-m. During the process of engagement, the clutch torque was found to have fluctuations with a small amplitude initially and gradually became stable after complete engagement.

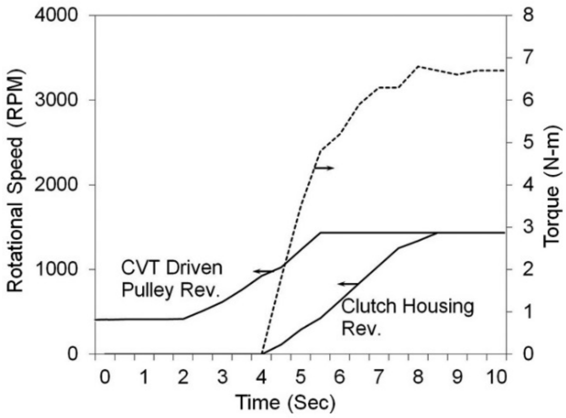


Figure 3. Variation in rotational speed of CVT driven pulley and clutch housing and clutch torque with normal friction liner at 1st test cycle

For the centrifugal clutch under test, Figure 4 illustrates the engagement characteristics with a grooved friction liner in which three engagement parameters have been considered for a time period of 10 sec. At the idle speed, the CVT-driven pulley was rotating at about 415 rpm. The rotational speed of CVT-driven pulley began to rapidly increase with engine throttling, which in turn generated centrifugal force and moved the clutch shoes outward. The engagement of clutch shoes with clutch housing was initiated at around 1027 rpm and 4.25 sec. At around 7.5 sec, the full engagement was attained when the clutch housing had a rotational speed equal to that of the CVT-driven pulley. In a very short time after the beginning of the engagement, the clutch torque had increased sharply to about 7.1 N-m. During the process of engagement, the clutch torque was found to have less fluctuations with a small amplitude than the fluctuations observed with a normal friction liner.

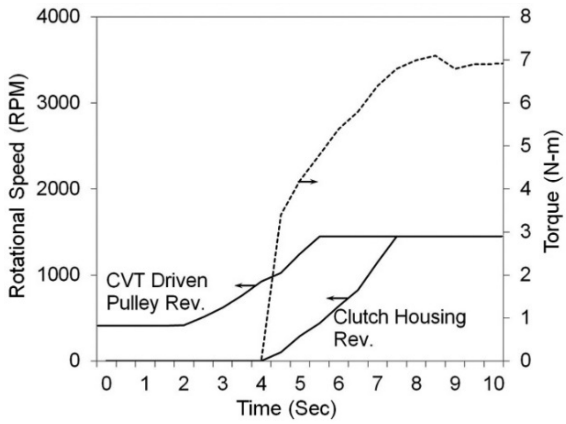


Figure 4. Variation in rotational speed of CVT-driven pulley and clutch housing and clutch torque with grooved friction liner at 1st test cycle

Surface Roughness and Thickness of Friction Liner

Surface roughness is a crucial factor in the clutch's characterization [17], [41]. Figure 5(a) shows a comparison of the surface roughness for the normal friction liner and the grooved friction liner at the leading section. Both friction liners went through 100 test cycles. In the 1st test cycle, the same surface roughness was observed for both friction liners at 11.421 µm. After 100 test cycles, the surface roughness of the normal friction liner was reduced to 10.495 µm, whereas the surface roughness of the grooved friction liner reduced to 10.685 µm. However, surface roughness retention of 1.8% has been observed in grooved friction liners after the 50th and 100th test cycle.

Figure 5(b) shows a comparison of the surface roughness for the normal friction liner and the grooved friction liner in the middle section. In the 1st test cycle, the same surface roughness has been observed for both friction liners at 11.426 µm. After 100 test cycles, the surface roughness of the normal friction liner was reduced to 10.821 µm, whereas the surface roughness of the grooved friction liner reduced to 10.886 µm. However, a retention of 0.8% was observed after the 50th test cycle, and a retention of 0.6% observed after 100th test cycle in the grooved friction liner.

Figure 5(c) shows the comparison for the surface roughness of the normal friction liner and grooved friction liner at the trailing section. In the 1st test cycle, the same surface roughness have been observed for both friction liners, i.e. 11.426 µm. After 100 test cycles, the surface roughness of the normal friction liner reduced to 11.018 µm, whereas surface roughness of the grooved friction liner reduced to 11.056 µm. However, a retention of 0.4% was observed after the 50th test cycle, and a retention of 0.3% was observed after the 100th test cycle in the grooved friction liner.

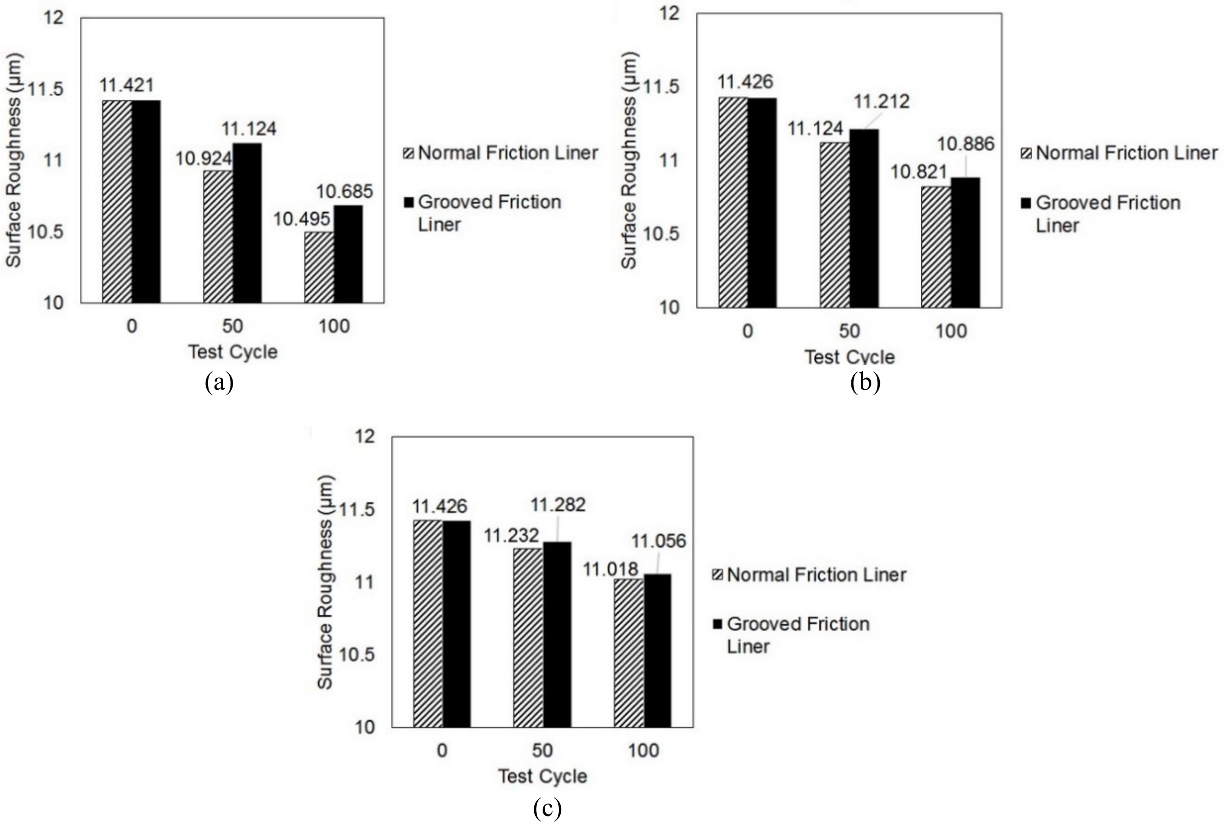


Figure 5. Variation in surface roughness at the (a) leading section, (b) middle section and (c) trailing section

For both types of friction liners, a gradual reduction in surface roughness has been witnessed through the run of 100 test cycles. The surface roughness reduction in normal friction liner has turned out to be 8.11%, 5.29% and 3.57% at leading, middle and trailing sections, respectively. The surface roughness reduction in grooved friction liner has turned out to be 6.44%, 4.73% and 3.24% at the leading, middle and trailing sections, respectively. Therefore, the surface roughness reduction in the grooved friction liner has been found to be less than the normal friction liner, thus retaining the friction characteristics for a longer period of time.

Table 1 compares the thickness of normal friction liner and grooved friction liner. Both the friction liners went through 100 test cycles. The test was started with same thickness for both friction liners at 4.02 mm. After 100 test cycles, the thickness of normal friction liner at the leading section was reduced to 3.82 mm, whereas the thickness of the grooved friction liner at leading section reduced to 3.87 mm. Retention of 1.3% has been observed in the thickness of the grooved friction liner after the 50th and 100th test cycle.

Table 1. Variation in friction liner thickness

Test cycle	Normal liners			Grooved liners		
	Leading section	Middle section	Trailing section	Leading section	Middle section	Trailing section
0	4.02	4.02	4.02	4.02	4.02	4.02
10	4.00	4.01	4.02	4.02	4.02	4.02
20	3.97	4.01	4.02	4.01	4.02	4.02
30	3.97	3.98	4.01	4.01	4.01	4.01
40	3.92	3.98	4.01	3.99	4	4.01
50	3.92	3.95	4.00	3.97	3.99	4.01
60	3.88	3.95	4.00	3.95	3.98	4.01
70	3.88	3.9	4.00	3.93	3.97	4.00
80	3.88	3.9	3.98	3.91	3.96	3.99
90	3.82	3.88	3.98	3.89	3.95	3.98
100	3.82	3.88	3.96	3.87	3.94	3.97

After 100 test cycles, the thickness of normal friction liner at the middle section reduced to 3.88 mm, whereas surface roughness of the grooved friction liner at middle section reduced to 3.94 mm. Retentions of 1.0% and 1.5% have been observed in the thickness of grooved friction liner after 50th and 100th test cycle, respectively. After 100 test cycles, the thickness of normal friction liner at trailing section reduced to 3.96 mm, whereas thickness of grooved friction liner at trailing section was reduced to 3.97 mm. Retention of 0.2% and 0.3% have been observed in the thickness of the grooved friction liner after the 50th and 100th test cycle, respectively.

For both types of friction liners, a gradual reduction in thickness has been witnessed through the run of 100 test cycles. The thickness reduction in normal friction liner has turned out to be 4.98%, 3.48% and 1.49% at leading, middle and trailing sections, respectively. The thickness reduction in grooved friction liner has turned out to be 3.73%, 1.99% and 1.24% at leading, middle and trailing sections, respectively. The thickness reduction in the grooved friction liner has been found to be less than the normal friction liner, thus retaining the friction material for a longer period of time.

Clutch Housing Temperature

The clutch housing temperature has been measured after the span of 10 test cycles for both types of friction liners. The temperature characteristics of both types of friction liners are shown in Figure 6. The higher clutch housing temperature has been witnessed in case of clutch with grooved friction liner than the clutch with normal friction liner. The difference of around 7.7% has been observed up to 40th test cycle. Later, gradual reduction in the difference of housing temperature between grooved friction liner and normal friction liner has been detected and turned out to be 1.56% at 100th test cycle. The higher clutch housing temperature for grooved friction liner might be a result of improved heat transfer by convection through the groove area. This increased heat transfer by convection reduces the internal energy of the friction liner, leading to increased lifetime of clutch [31].

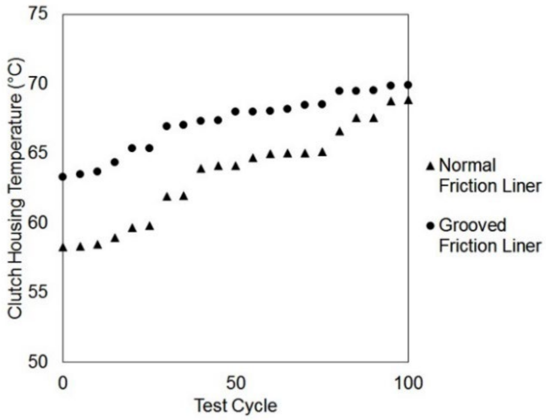
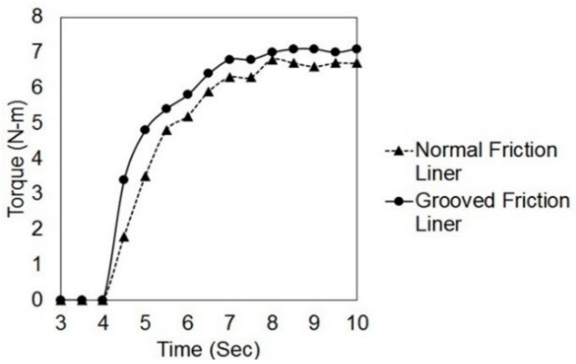


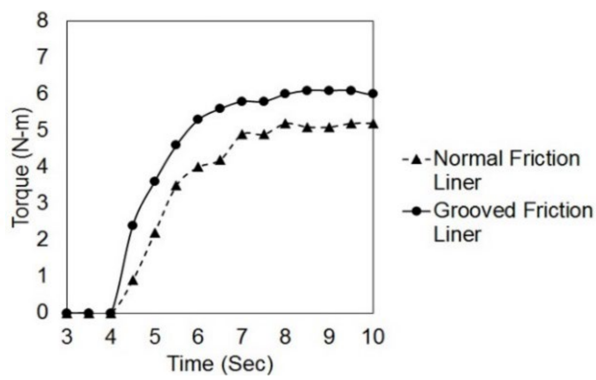
Figure 6. Variation in clutch housing temperature

Clutch Torque

It is important to obtain an accurate value of the clutch torque in order to examine the dynamic performance and the economic performance of a vehicle. Figure 8 shows the comparison of the clutch torque measured using a normal friction liner and a grooved friction liner. Figure 7(a) shows clutch torque variation for the time period of 10 sec during 1st test cycle. A steeper rise in clutch torque for the grooved friction liner has been observed at the beginning of the engagement. The clutch torque with a grooved friction liner was found to have fewer fluctuations than the clutch torque with a normal friction liner. The clutch torque has been found to be stable after full engagement. After achieving a stable value, the clutch torque with a grooved friction liner has 5.97% more than the clutch torque with a normal friction liner. Figure 7(b) and Figure 7(c) show clutch torque variation for a time period of 10 sec during the 50th test cycle and 100th test cycle, respectively. A similar trend has been observed in clutch torque behavior for both types of friction liners. However, gradual reduction has been witnessed in the amount of torque transmitted by the clutch. After achieving stable value in 50th test cycle, the clutch torque with grooved liner was 17.31% more than the clutch torque with a normal clutch liner. After achieving a stable value in the 100th test cycle, the clutch torque with a grooved liner was 15.22% more than the clutch torque with a normal clutch liner. The torque transmitted by the clutch with a grooved friction liner has turned out to be more than the normal friction liner, thus retaining the torque transmission capacity for a longer period of time.



(a)



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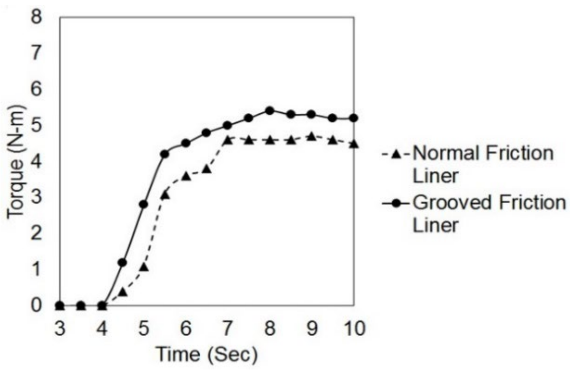


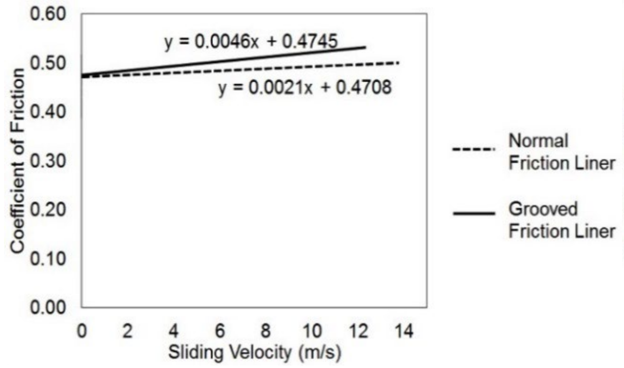
(c)

Figure 7. Variation in clutch torque (a) at 1st test cycle (b) at 50th test cycle (c) at 100th test cycle

A comparison of the rough regression of μ-V_r curve for normal friction liner and grooved friction liner is illustrated in Figure 8. As shown in Figure 8(a) at the 1st test cycle, the positive gradient for the coefficient of friction with sliding velocity have been observed for both types of friction liners. However, the gradient for grooved friction liner is more than the gradient for normal friction liner by 54.34%. This positive gradients indicate less propensity to judder. At 50th test cycle, the positive gradient for the coefficient of friction with sliding velocity has been observed for grooved friction liner while the negative gradient for the coefficient of friction with sliding velocity has been observed for normal friction liner in Figure 8(b). This negative slope tends to increase the propensity to judder. As shown in Figure 8(c), at 100th test cycle, the negative gradient for the coefficient of friction with sliding velocity have been observed for both types of friction liners. The large amplitude fluctuation in friction coefficient have been observed when the engagement was operating at a small sliding velocity. At 100th test cycle, the gradient for grooved friction liner is less than the gradient for normal friction liner by 54.12%. The results for gradient of coefficient of friction with sliding velocity for grooved friction liner may provide a better damping effect and little self-excited vibrations during prolonged use [42]–[44].

The research shows that a positive gradient of coefficient of friction with sliding velocity provide a better damping effect and little or no self-excited vibrations occur whereas a negative slopes increase the propensity to judder [45]–[47]. To measure the effect of judder, displacement sensors have been mounted at three different positions on the vehicle [3] i.e. handlebar, seat and platform. The displacement developed at handlebar for the period of 160 sec have been illustrated in Figure 9. These 160 sec include two parts in which the first 90 sec represents the run with sub cycle-highway followed by 70 sec of run with sub cycle-city dense traffic. The displacement during 1st test cycle for both types of friction liner have been shown in Figure 9(a). For clutch with normal friction liner, the displacement increased once the clutch engagement initiated. The amount of displacement became cyclic and dense after the full engagement. The displacements for the highway run have been found denser than the displacements for the city traffic run. The less density of displacement in city traffic run was a result of lower operating speeds with frequent throttling and braking. The sag near 90 sec was due to the lag period between two successive sub-cycles in test. However, the amount of displacement decreased near 60 sec, 113 sec and 128 sec. For clutch with grooved friction liner, the displacement increased once the clutch engagement initiated. The displacements for the highway run have been found equally dense like the displacements for the city traffic run. However, the amount of displacement decreased near 55 sec, 113 sec and 128 sec. The amount of displacements generated with grooved friction liner were lesser than the normal friction liner. Moreover, the maximum value displaced was 2.17 mm for normal friction liner and 1.62 mm for grooved friction liner.

The displacement during 50th test cycle for both types of friction liner have been shown in Figure 9(b). For clutch with normal friction liner, the displacement increased once the clutch engagement initiated. The amount of displacement was much higher than that observed at 1st test cycle. Moreover, the displacements during 50th test cycle have been found less dense than the displacements for the 1st test cycle. The less density of displacement at 50th test cycle was a result of unapproachable zero position of sensor due to high vibrations. The amount of displacement decreased near 38 sec, 128 sec and 150 sec. For clutch with grooved friction liner, the displacement increased once the clutch engagement initiated. The amount of displacement was higher than that observed at 1st test cycle. The displacements during 50th test cycle have been found equally dense like the displacements for the 1st test cycle. At 50th test cycle, the amount of displacements generated with grooved friction liner were lesser than the normal friction liner. Moreover, the maximum value displaced were 2.64 mm and 2.16 mm with the normal friction liner and with grooved friction liner respectively. The displacement during 100th test cycle for both types of friction liner have been shown in Figure 9(c). For clutch with normal friction liner, the amount of displacement was much higher than that observed at 1st and 50th test cycles.



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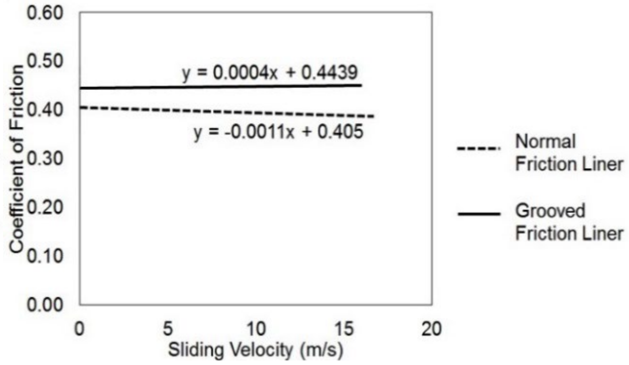

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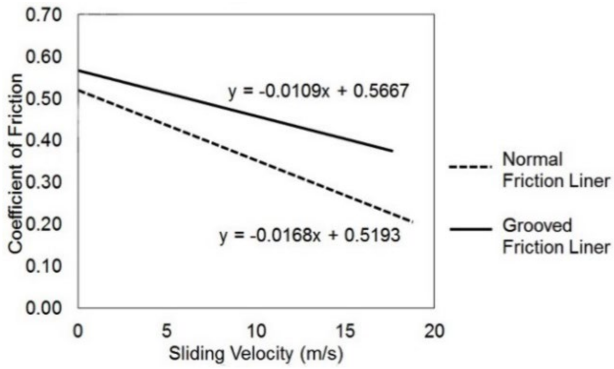
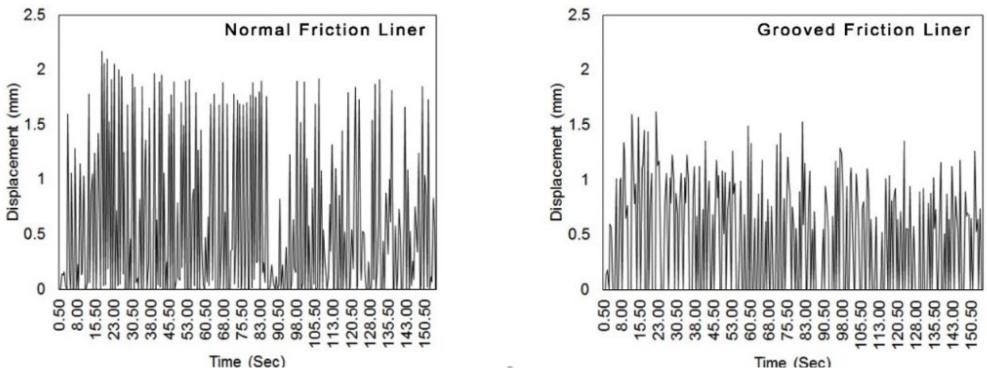


(c)

Figure 8. Rough regression of friction coefficient with relative sliding velocity by a straight line (a) at 1st test cycle (b) at 50th test cycle (c) at 100th test cycle

Moreover, the displacements during 100th test cycle have been found less dense than the displacements for the 1st and 50th test cycles. For clutch with grooved friction liner, the amount of displacement was higher than that observed at 50th test cycle. The displacements during 100th test cycle have been found less dense than the displacements for the 50th test cycle. At 100th test cycle, the amount of displacements generated with grooved friction liner were lesser than the normal friction liner. Moreover, the maximum value displaced were 3.22 mm and 2.8 mm with the normal friction liner and with grooved friction liner respectively. Similarly, displacements at seat and platform have been recorded and illustrated in Figure 10 and Figure 11 respectively.



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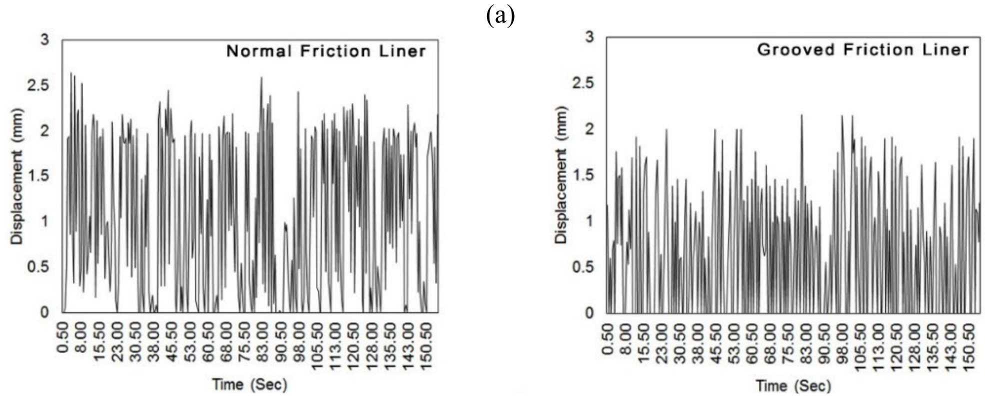

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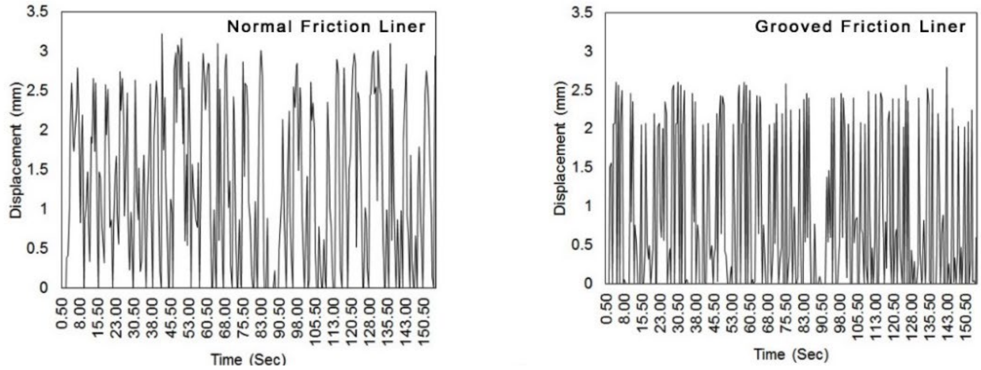
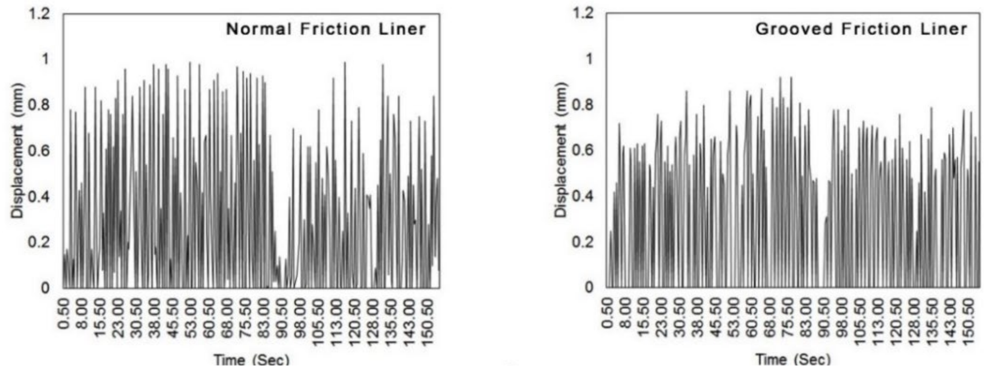
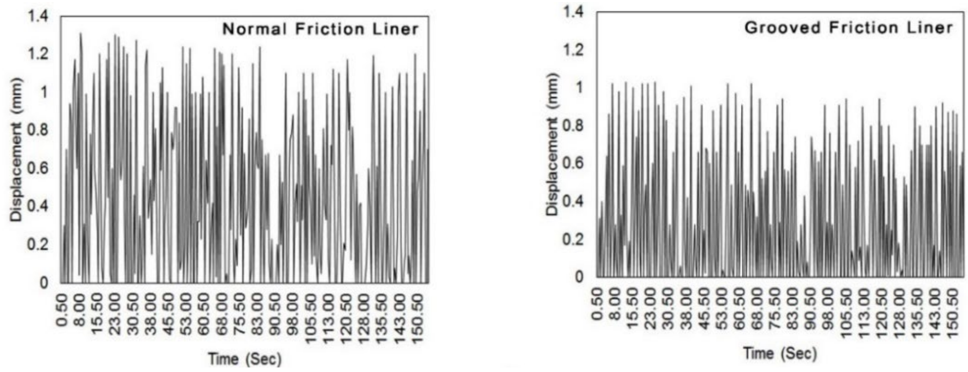


(c)

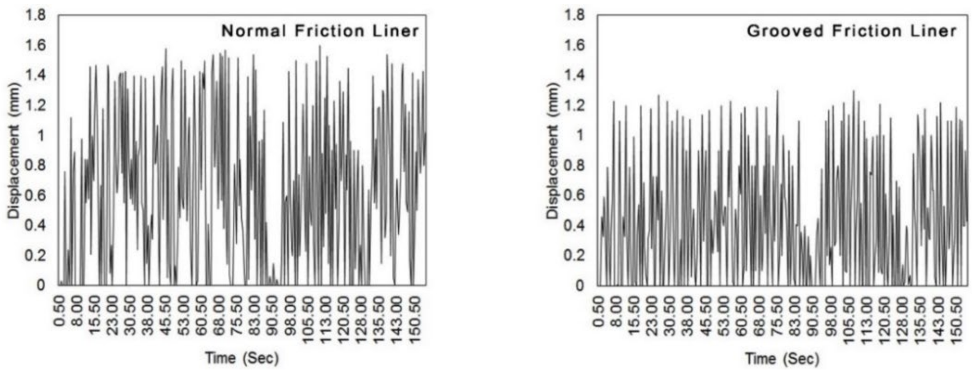
Figure 9. Displacement at handle bar (a) at 1st test cycle (b) at 50th test cycle (c) at 100th test cycle



(a)



(b)



(c)

Figure 10. Displacement at seat (a) at 1st test cycle (b) at 50th test cycle (c) at 100th test cycle

As shown in Figure 10 for the second location i.e. seat, during the 1st test cycle, the maximum value displaced with the normal friction liner was 0.99 mm and with grooved friction liner was 0.92 mm. During 50th test cycle, the maximum value displaced with the normal friction liner was 1.31 mm and with grooved friction liner was 1.03 mm. During 100th test cycle, the maximum value displaced with the normal friction liner was 1.6 mm and with grooved friction liner was 1.3 mm. As shown in Figure 11 for the third location i.e. platform, during the 1st test cycle, the maximum value displaced were 0.99 mm and 0.88 mm with the normal friction liner and with grooved friction liner respectively. During 50th test cycle, the maximum value displaced were 1.2 mm and 1.1 mm with the normal friction liner and with grooved friction liner respectively. During 100th test cycle, the maximum value displaced were 1.59 mm and 1.3 mm with the normal friction liner and with grooved friction liner respectively.

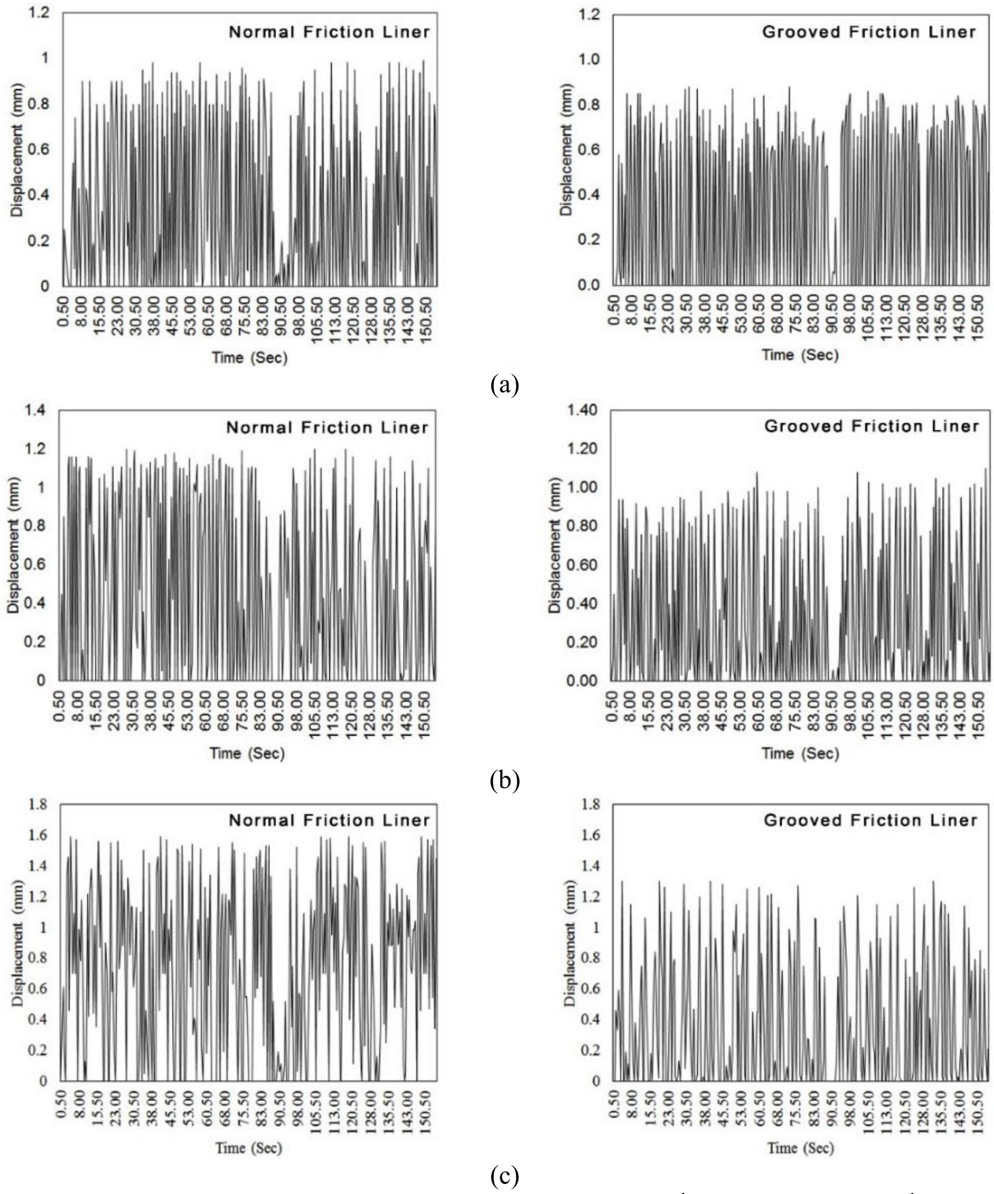


Figure 11. Displacement at platform (a) at 1st test cycle (b) at 50th test cycle (c) at 100th test cycle

The similar displacement trends have been observed for all three locations i.e. handlebar, seat and platform. The larger amount of displacements were results of high amount of judder. The results presented in Figure 8 have been supported with the increasing amount of displacement at all three locations. Moreover, the judder characterization of normal friction liner and grooved friction liner have also been strengthen through the maximum displacement values observed with both type of friction liners. The negative gradients of the coefficient of friction with sliding velocity resulted in to increased judder and so as the displacements at three locations. However, judder generated with grooved friction liner were less than the judder generated with normal friction liner.

FUEL ECONOMY

Frictional loss in the transmission system of CVT driven scooter affects the fuel economy. The centrifugal clutch running with worn out friction liners plays vital role in decreasing fuel economy [12], [15]. The fuel economy tests have been carried out for the vehicle under test and comparative study between clutch with normal friction liner and clutch with grooved friction liner has been presented in Figure 12. The higher fuel economy has been witnessed when vehicle was operated using clutch with grooved friction liner. During 1st test cycle, the fuel economy measured for clutch with normal friction liner was 42.2 KMPL and for clutch with grooved friction liner was 42.88 KMPL. Later during 100th test cycle, the fuel economy measured for clutch with normal friction liner was 39.29 KMPL and for clutch with grooved friction liner was 39.65 KMPL. The improved fuel economy of vehicle using clutch with grooved friction liner after prolonged use indicates the retention of friction characteristics. Figure 13 shows visual comparison of normal friction liner and grooved friction liner after the run of 100 test cycles.

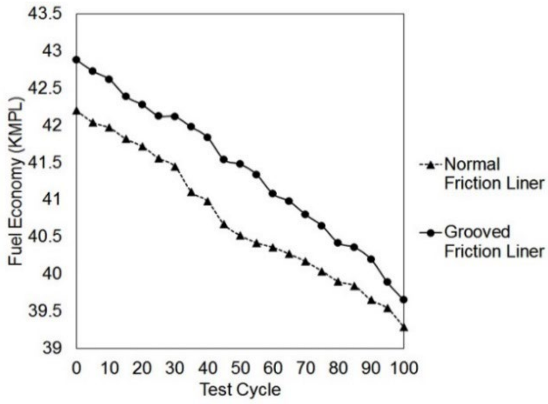


Figure 12. Variation in fuel economy

Figure 13. Variation in fuel economy

CONCLUSIONS

In this study, an attempt was made to study the characteristics of centrifugal clutch using grooved friction liner. A test cycle for centrifugal clutch has been developed using number of engagement as basis. Vehicle test bench was used where the developed test cycle was automated. Test bench was mounted with appropriate sensors to monitor and measure the parameters like fuel economy, surface roughness, thickness of friction surfaces, clutch output torque, clutch housing temperature, the rotational speeds of the CVT driven pulley and clutch housing. The amount of displacements were also recorded at three locations when vehicle was subjected to clutch judder. The judder characteristics of centrifugal clutch under test were also studied. The experiment run was started with new pair of friction liners. First, the performance characteristics were measured with normal friction liners for 100 test cycles followed by measurement of performance characteristics with grooved friction liners. The following points can be concluded from this study.

i. For both types of friction liners, gradual reduction in surface roughness has been witnessed through the run of 100 test cycles. At leading section, the surface roughness reduction in normal friction liner has turned out to be 8.11% and in grooved friction liner the surface roughness reduction has turned out to be 6.44%. At middle section, the surface roughness reduction in normal friction liner has found to be 5.29 % and in grooved friction liner the surface roughness reduction has found to be 4.73%. At trailing section, the surface roughness reduction in normal friction liner has reported to be 3.57% and in grooved friction liner the surface roughness reduction has reported to be 3.24%.Therefore, the surface roughness reduction in grooved friction liner has turned out to be less than the normal friction liner thus retaining the friction characteristics for longer period of time.
ii. For both types of friction liners, gradual reduction in thickness has been witnessed through the run of 100 test cycles. At leading section, the thickness reduction in normal friction liner has turned out to be 4.98% and in grooved friction liner the thickness reduction has turned out to be 3.73%. At middle section, the thickness reduction in normal friction liner has found to be 3.48% and in grooved friction liner the thickness reduction has found to be 1.99%. At trailing section, the thickness reduction in normal friction liner has reported to be 1.49% and in grooved friction liner the thickness reduction has reported to be 1.24%. The thickness reduction in grooved friction liner has turned out to be less than the normal friction liner thus retaining the friction material for longer period of time.

iii. The higher clutch housing temperature has been witnessed when the clutch was operated with grooved friction liner. During the 100th test cycle, the housing temperature for a clutch with grooved friction liner was found 1.56% more than the clutch with normal friction liner.

iv. The clutch torque with grooved friction liner has found 5.97% more than the clutch torque with normal friction liner at 1st test cycle. The clutch torque with grooved friction liner has recorded 17.31% more than the clutch torque with normal friction liner at 50th test cycle. The clutch torque with grooved friction liner has found 15.22% more than the clutch torque with normal friction liner at 100th test cycle.

v. At 50th test cycle, the positive gradient for the coefficient of friction with sliding velocity has been observed for grooved friction liner while the negative gradient for the coefficient of friction with sliding velocity has been observed for normal friction liner. At 100th test cycle, the negative gradient for the coefficient of friction with sliding velocity have been observed for both types of friction liners. The derived judder characteristics were evidently supported through the displacements recorded at handle bar, seat and platform.

vi. The higher fuel economy was witnessed when vehicle was operated using clutch with grooved friction liner. The difference of around 1.59% was observed during 1st test cycle. Later, reduction in the difference of fuel economy was detected and turned out to be 0.91% at 100th test cycle.

Further, the same methodology can be used to study the characteristics of centrifugal clutch using grooved friction liners up to last acceptable life limit. Theoretical models can be developed to study the effect of different types of grooves on the centrifugal clutch performance. Optimization of best groove design can also be carried out.

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