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PHYSIOLOGICAL CHARACTERISATION OF ELECTRO-PNEUMATIC CLUTCH ACTUATION SYSTEM FOR HEAVY DUTY VEHICLES

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ABSTRACT

Characterisation is a process of describing the distinctive nature or feature of an environment, with the purpose of establishing the existing nature and scientific measurement of that environment's parameters. The electro-pneumatic clutch actuation system of a heavy-duty vehicle was characterised. A Mercedes Benz Actros Truck model MP 2, 2031 was chosen for the investigation. The empirical study cut across relevant features of the actuation chamber including piston and control valve springs, gearing timing parameters and resultant error signals in the actuator. The test instruments including linear tools and specialised ones like Xentry Tab 2, Xentry Connect and Mass Spring Balance were deployed in the characterisation. Data in respect of the Sensor Monitor signals/Clutch travel distance, Gear Shift parameters under static and dynamic conditions, error signals as the Piston rod in the actuator translates from disengagement to engagement modes and vice versa and the resultant forces on the piston spring during clutch engagements and disengagements were all established by direct measurements in most cases.

KEYWORDS: Actuation, Characterisation, Measurement, Motion, Transmission.

1.0: INTRODUCTION

In automotive industries, transmission process is an essential mechanism used in transferring mechanical power of a motor vehicle from the engine to the wheels. Actuation on the other hand is important in any physical process that requires mechanical movement. Actuation process is a system that utilises its outputs to achieve a control action on a machine or device, with the ultimate aim of producing a linear or rotary motion. A device or element deployed for this purpose of actuation is an actuator (Carlos, 2016). An actuator can further be explained as an element or device that transforms a mechanical movement from linear to rotary motions or vice versa. This transformation is usually required in any physical process. Both transmission and actuation processes are independent terms but are often inseparable in a drive line system such as heavy-duty vehicles. The block diagram representation of an actuator is shown in figure 1.

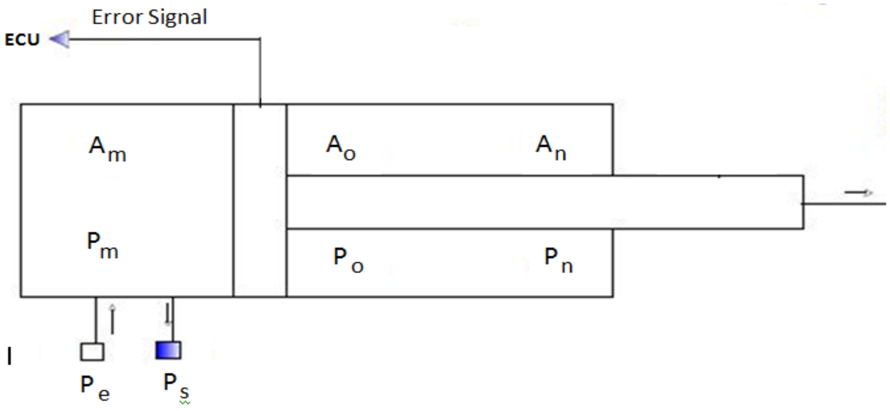

Fig. 1: Block diagram of an Actuator Chamber.

The Actuator chamber is demarcated into chambers A_m and A_n respectively by the piston rod.

A_m is the area of first chamber.

A_n is the area of second chamber.

p_m is the pressure on first chamber.

p_n is pressure on second chamber.

A_o is the area covered by the piston rod or the difference in area of the piston rod in first and second chambers subject to atmospheric pressure, p_o.

$A_o = A_m - A_n$

p_o is the ambient atmospheric pressure of the area A_O.

p_s is the inlet port through which pneumatic pressure is supplied to the chamber.

The pressure (p_s) pushes the piston in the chamber towards the right direction during clutch disengagement. Similarly, pressure (p_e) is let off the chamber through the exhaust port. This pressure is responsible for pushing the piston in the reverse direction during clutch engagement. The sensor error signal is the feedback signal to the electrical control unit (ECU) that monitors the clutch travels or piston rod displacements in the chamber and thus ensure that each cycle of engagement or disengagement are fully realised.

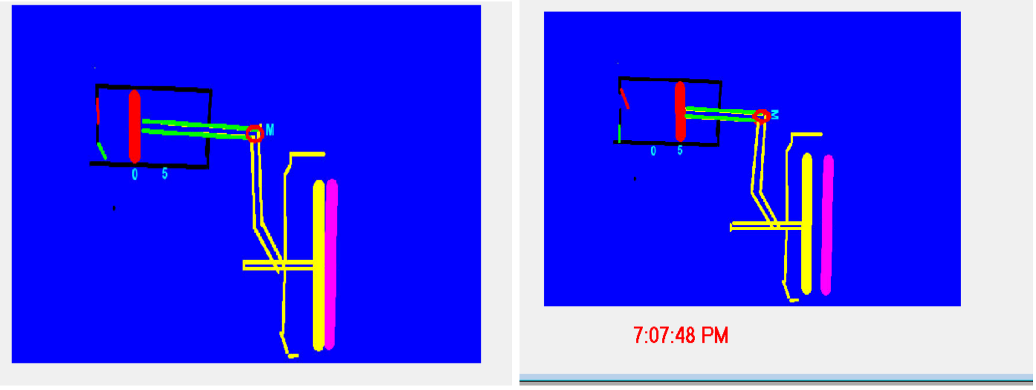

Fig. 2: Engagement and disengagement modes of a clutch system.

In other to understand the actuator, its characterisation becomes imperative. Characterisation is a descriptive disclosure of the important features of a system. It could be established in figures or in images (Lifshin and Cahn 1993). The actuator is a driveline system component. Driveline system component are those elements that transmit and control power and motion. Typical examples are the brakes and the clutches. A Brake system can be explained from the view point of a clutch system in the sense that it can assist in reducing the motion of a moving body like a clutch. But unlike a clutch where both shafts are rotatory, one of the shafts of a brake is fixed or held stationary while the other shaft rotates just like in a clutch. The essence of clutching is to bring about partial or total engagement of two shafts that are rotating on their own independently at different speeds. By so doing, it may result in reduced speed of the shafts when partially engaged. If both shafts are held together, it can result in total stoppage of the rotation of the shaft. This is total engagement of the clutch. These independently rotating shafts can be transmitted or linked together by direct mechanical lockup or by mechanical friction. It can also be linked by electromagnetic, hydraulic or pneumatic forces (Schmid *et al*, 2014). Clutches are applied in vehicles and industrial equipment through either one or a combination of the aforementioned transmission mediums. Each has their merits and demerits and are usually a subject of considerations for each application. A pneumatic process of actuation uses compressed air as a means for the transmission and control of energy. Train door controls, automatic production lines and mechanical clams are some of the many application areas of pneumatic technologies. Basically, a pneumatic actuator is any system which is capable of converting energy from a compressed air supply into a linear or rotary motion to do work.

Essentially, pneumatic actuators generally represent the output action that perform business or work at the end of any pneumatic system process. Pneumatic actuation is therefore a process of energy transfer through which specific tasks or works (ie. clamping, pick and place, filling, ejection and tool changing) are accomplished. Initially, pneumatics technology was limited only to areas where actuations are commonly available in cylinder actuators and where linear movement of a comparatively large piston rod are confined. Economic considerations coupled with improved design requirements in some powerful and efficient devices have led to an increased demand for a wide range of pneumatic actuators (Carlos, 2016). The essential numerical features or images of a clutch actuation system in which electrical control and pneumatic transmissions are deployed for the effective operation of heavy-duty vehicle are the thrust of this presentation.

2.0: METHODS

Empirical Research Design method was adopted. Experiments were done with the following;

1. Meter Rule were used to measure free lengths of Piston and Valve Control coils or sprigs, displacements in the free lengths of the piston and valve control coils, clutch displacements in the actuator and cross-sectional diameter of the actuator respectively.
2. Venire Callipers were used for the measurement of the diameter and thickness of piston and control valve coils.
3. Mass Spring Balance was the tool for measuring spring compression with respect to applied force.
4. The Xentry Tab 2 and Xentry Connect were used inside the test truck. The Xentry Connect formed an interface with the vehicle engine module, while the Xentry Tab 2 was the output display monitor from where data on engagement and disengagement timings of the clutch at different gear positions for both static and dynamic conditions were read and recorded.
5. An a. c. motor or dynamometer in which a driver shaft was attached was made to rotate with the speed of the dynamometer.
6. A Tachometer was deployed for monitoring the speed of the driver shaft via a led sensor.

7. The driven shaft or disc plate or flywheel was loaded via a pulley system. The setup of the experiments is pictured in figure 3 below.

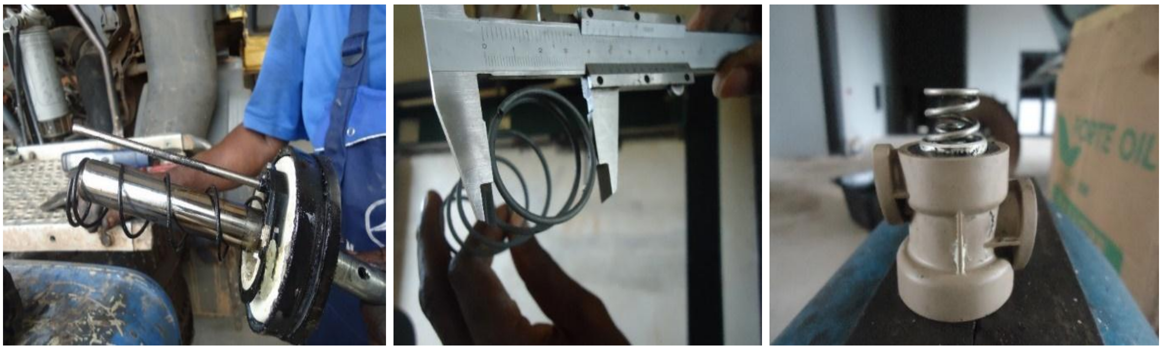

Fig. 3(a): Piston and control valve springs undergoing measurements, (Ndubuisi et el 2022).

Fig. 3 (b): Test truck; Actros Truck model MP 2, 2031 and Xentry Connect module, (Ndubuisi et el 2022).

Fig. 3 (c): Features of the experimental set up of a single loaded clutch disc, (Ndubuisi et el 2022).

3.0: Experimental Design, Results and Discussions

The results of the experiments are presented here under the following headings.

3.1: Piston Spring coils, Supply pressure (Ps) and Exhaust Pressure (Pe)

Table 1 shows the result of mass spring compression of the piston spring. The trend of the compression is graphically illustrated in figure 4 with **force against spring extensions.**

Table 1: The result of mass/spring compression of piston coil experiment, (Ndubuisi et el 2022).

Force/weight (N)	Free Length, L_f (cm)	Length Variations, L_i (cm)	L_f-L_i (cm)
0	17	17	0
5	17	16.5	0.5
10	17	15.9	1.1
15	17	15.5	1.5
20	17	14.9	2.1
25	17	14.5	2.5
30	17	14.0	3.0
35	17	13.5	3.5
40	17	12.9	4.1
45	17	12.5	4.5

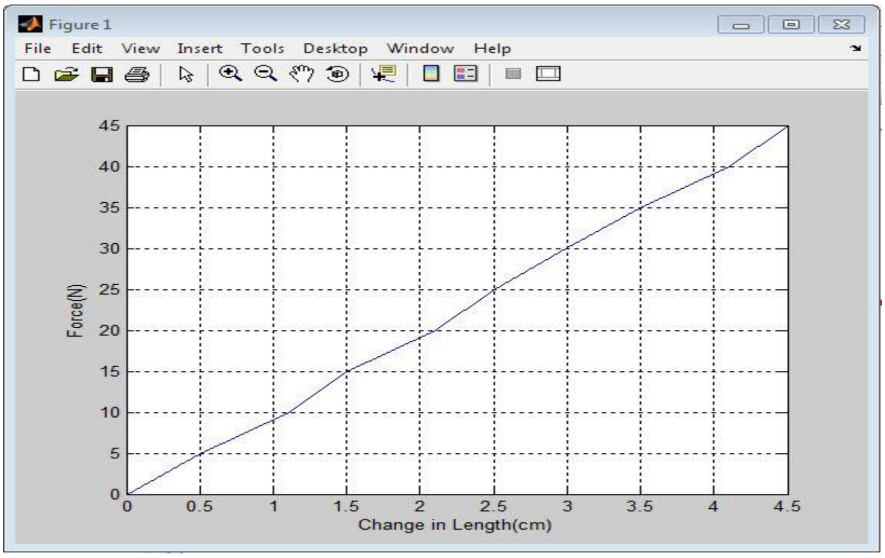

Fig. 4. Graph of Mass Spring Compresion of piston spring, (Ndubuisi et el 2022).

Other measurable data of the piston spring is presented in table 2 while table 3, shows the summary of derived data as culled from '**An Experimental Determination of the Supply and Exhaust Pressures in an Electro-Pneumatic Clutch Actuation System**' (Ndubuisi *et al*, 2022) where the mechanics of spring materials standard analysis for springs under tensile or compressive stresses were applied.

Table 2: Measured Parameters related to piston spring, (Ndubuisi et el 2022).

S/N	Parameters	Readings
1	Piston coil diameter (D)	4.01cm

2	Piston wire diameter (d)	0.235cm
3	Piston coil free length (L_f)	17cm
4	Number of Piston coils (N)	7
5	Compressor capacity	8 bars
6	Diameter of the inlet/supply port or valve (Vs)	1.41cm
7	Diameter of the outlet/exhaust port or valve (Ve)	1.30cm

Table 3: Derived Parameters related to piston spring, (Ndubuisi et el 2022).

S/N	Parameters	Readings
4	Piston coil solid length (Ls)	1.88cm
6	Piston coil maximum deflection P(Δ_L)	15.12cm
7	Supply Pressure (P_S)	9.61 Bars
9	Oil pressure	1.61bars
12	Exhaust Pressure (Pe)	$11.299bars$

3.2: The Control Valve Spring Coil and Electrical Control module

Like the piston spring, the control valve spring was also subjected to mass spring compression experiment. The result is tabulated in table 4 and consequently plotted with force against spring extensions in figure 5.

Table 4: The result of the mass/spring compression of Control Valve coil experiment, (Ndubuisi et el 2022).

Force/weight (N)	Free Length, L_f (cm)	Length Variations, L_i (cm)	L_f-L_i (cm)
0	1.94	1.94	0
20	1.94	1.82	0.12
40	1.94	1.63	0.31
60	1.94	1.46	0.48
80	1.94	1.25	0.69

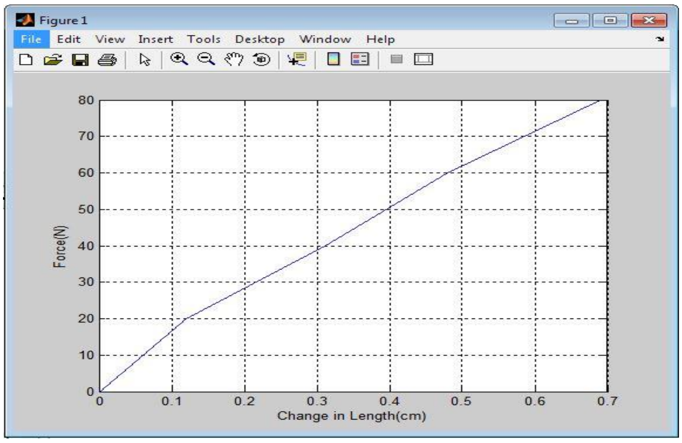

Fig. 5: Graph of Mass Spring Compression of control valve spring, (Ndubuisi etal 2022).

The data associated with the control valve spring measurement is recorded in table 5.

Table 5: Measured parameters related to control valve spring, (Ndubuisi et el 2022).

S/N	Parameters	Readings
1	Valve coil diameter (D)	1.435cm
2	Valve wire diameter or thickness (d)	0.165cm
3	Control valve diameter or port (Dc)	2.865cm
4	Valve coil free length (L_f)	1.94cm
5	Number of coils in the Valve (N)	5
6	Control Valve voltage (V_{CV})	24V

The mass spring system experiment was also used to determine the amount of pressure exerted by ECU in switching the control valve for the opening of the supply valve/port and closing the exhaust valve/port during clutch disengagement. It was also used to determine pressure required to close the supply valve/port and open the exhaust valve/port during clutch engagement. Some of the parameters were however derived. The details are contained in the study titled '**Investigation of the Valve Control Pressure and Current in an Electro-Pneumatic Clutch Actuation System'** (Ndubuisi *et al*, 2022). Control valve data as presented in tables 4 and 5, including the graph of figure 5 are deployed to yield the derived data shown table 6.

Table 6: Derived parameters related to control valve spring, (Ndubuisi et el 2022).

S/N	Parameters	Readings
1	Valve coil solid length (Ls)	0.99cm
2	Valve coil maximum deflection (Δ_L)	0.95cm
3	Control Valve pressure (V_{CP})	191.529 bars
4	Control Valve current (V_{CI})	82 milliamps
5	Control Valve power (V_{CP})	$1.96 watts$

3.3: Sensor Monitor (Clutch travel measurement)

The sensor rod was used to monitor the signals of the clutch travel in units called 'counts.' The signal is sent to the engine module through the electrical control unit which processes the sensor signal and establishes if the clutch travel is within a safe region or otherwise. The clutch travel is also related to clutch wear. The result of the measurement of clutch travel is illustrated in table 7, while the graphical representation is displayed in figure 5 respectively.

Table 7: Clutch travel measurement and piston displacement in the actuator.

S/N	Parameter	Reading(mm)/ Status
1	Clutch travel (mm)	60min/Engage to 65max/ Disengage
2	Sensor signal (counts)	0.450 / Engage to 0.950/ Disengage

3	Clutch travel or displacement (mm)	5
4	Diameter of the actuator chamber housing the piston (mm)	108
5	Length of the actuator chamber housing the piston (mm)	90

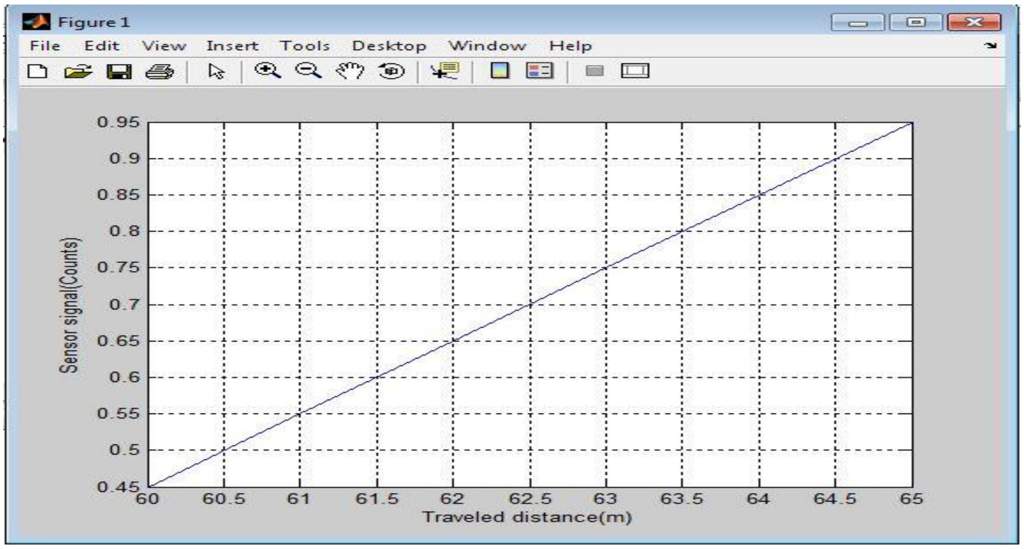

Fig. 6: Graph of clutch travel in the actuator.

The distance the clutch can travel as it disengages or engages is a measure of the permissible displacement of the piston rod in the actuator chamber. The sensor rod which monitor’s the position of the piston in the actuator is calibrated in counts while the displacement of the piston rod in the actuator is calibrated in millimetres. At clutch engagement mode, the sensor signal reads 0.45 counts while the clutch travel determined by the piston position in the actuator reads 60mm. This is the datum or rest position of the actuation. At clutch disengagement mode, signal monitor is at 0.95 counts corresponding to clutch travel displacement of the piston of 5 millimetres and at a calibrated value of 65millimeter on the actuator chamber. At any point in between these established points is a slippery mode of the clutch and is not desirable in cutch actuation.

3.4: Gear Shift Characteristics and Error Signal at different Gear Positions

The Xentry Tab 2, and Xentry connect test instruments were deployed in the engine model terminal provided in the vehicle for gear shift characteristics measurements. Table 8 is the data obtained when the test truck is steaming while stationary. Gears were changed from position one to position four, one at a time while the duration for engagements and disengagements were recorded. The fourth gear is the maximum that could be accessed in a stationary mode. Table 9 was read as the truck took off usually from gear two. The rest of the reading were obtained while the truck was in motion. Under this dynamic mode, gear transitions are up to eight. The neuro-fuzzy disengagement time is deduced from the visual basic design for demonstration of clutch actuation timing and gearing position.

Table 8: Gear Shift (static) Time Characteristics in milliseconds (ms).

Clutch Status	**Gear 1**	**Gear 2**	**Gear 3**	**Gear 4**
Engage (ms)	175	220	255	285
Disengage (ms)	85	85	90	90

Table 9: Gear Shift (Dynamic) Time Characteristics in milliseconds (ms).

Clutch Status timing (ms)	**Gear 1**	**Gear 2**	**Gear 3**	**Gear 4**	**Gear 5**	**Gear 6**	**Gear 7**	**Gear 8**
Engagement	175	640	225	260	410	335	210	205
Disengagement	85	85	90	90	90	90	95	90
Neuro-Fuzzy Disengagement	80	80	85	85	85	85	85	85

The error signal experienced in the gears can be defined as;

$$\text{Error signal } (ESG_x) = \frac{\text{Travel distance of the piston} - \text{conventional distance of piston}}{\text{Conventional disengagement time} - \text{neurofuzzy disengagement time}}$$

Where,

$\text{Travel distance of the piston}$ are calibrated distances in the actuator.

$\text{Conventional distance of piston}$ are fixed points from which the piston initiates movement in either direction. $\text{Conventional disengagement time}$ and $\text{neurofuzzy disengagement time}$ are as tabulated in table 3 above.

Error signal for gear 1 $(ESG_1) = \frac{1.25-0}{85-80} = \frac{1.25}{5} = 0.25mm/s$

Error signal for gear 2 $(ESG_2) = \frac{2.5-0}{85-80} = \frac{2.5}{5} = 0.5mm/s$

Error signal for gear 3 $(ESG_3) = \frac{3.75-0}{90-85} = \frac{3.75}{5} = 0.75mm/s$

Error signal for gear 4 $(ESG_4) = \frac{5-0}{90-85} = \frac{5}{5} = 1.00mm/s$

Error signal for gear 5 $(ESG_5) = \frac{5.5-0}{90-85} = \frac{5.5}{5} = 1.1mm/s$

Error signal for gear 6 $(ESG_6) = \frac{6-0}{90-85} = \frac{6}{5} = 1.2mm/s$

Error signal for gear 7 $(ESG_7) = \frac{6.5-0}{90-85} = \frac{6.5}{5} = 1.3mm/s$

Error signal for gear 8 $(ESG_8) = \frac{7-0}{90-85} = \frac{7}{5} = \frac{1.4mm}{s}$

The results obtained from these calculations are summarised in table 10.

Table 10: Summary of error signal at different gear positions.

Gear position	**Error signals (ESG_x) mm/s**
Gear 1	0.25
Gear 2	0.5
Gear 3	0.75
Gear 4	1.0
Gear 5	1.1
Gear 6	1.2
Gear 7	1.3
Gear 8	1.4

3.5: A Single Loaded Clutch Disc Experiment

To realise this experiment, the driver shaft was attached to rotate with the speed of an a. c. motor or dynamometer. The driven shaft was a clutch disc or flywheel of radius 30.325mm. It was loaded via a pulley system. Loads or weight of 200 grams up to 1000grams were added gradually at 200 grams' increment on the pulley base pan and loading speed observed. Conversely, the loads were decremented at 200gram intervals from 1000gram to 200 grams and unloading speed were also recorded. The tachometer was used for measuring the speed of the driver shaft via a led sensor monitor positioned on top of the plate. The resultant speeds were read from the monitor accordingly and presented in table 11. The analytical relationships between speed, torque and power are subjects of another presentation in Ndubuisi et al (2022) and titled **'Empirical Study and Analysis of a Single Loaded Clutch Disc'.** However, the summary of the result of the analysis is presented in table 12. The setup of the experiment was shown pictorially above in figure 3 (c).

Table 11: Load (Weight), Loading speed and Unloading speed of a clutch system, (Ndubuisi et el 2022).

S/n	**Load (Weight) (gm)**	**Loading Speed (RPM)**	**Unloading Speed (RPM)**
1	0	∞	∞
2	200	1300	1310
3	400	1140	1150
4	600	990	1020
5	800	860	960
6	1000	735	800

Table 12: Loading or output Torque and output power of a clutch system, (Ndubuisi et el 2022).

S/n	Load (Weight) (gm)	Loading or output Torque (Nm)	Output Power (kw)
1	200	0.059498	8.100
2	400	0.1189952	14.207
3	600	0.179239	18.583
4	800	0.237991	21.434
5	1000	0.297488	22.899

4.0: CONCLUSION

The need to characterise engineering system is to establish the existing parameters. The data obtained can be analysed and often used to pave way for improvement in performance index of the system which at this instance is a heavy-duty vehicle. Piston error signal parameter that arises with increasing clutch load due to ageing requires to be contained at minimal level for optimum performance of the vehicle. Improved design to address this error ideally affect the engine module with respect to torque, speed and power of the heavy-duty vehicles. The conventional control parameters for error, torque, speed and power are obtained in this characterisation and will serve as sprig boards for adaptive intelligent agent control of an electro-pneumatic clutch actuation system. The proposed technology for which this characterisation will be deployed is aimed at improving the performance of heavy-duty vehicles.

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