International Journal of Recent Technology and Engineering (IJRTE)
ISSN: 2277-3878 (Online), Volume-11 Issue-6, March 2023

Effect of Fabricated Twin Roller Arrangement of Chassis Dynamometer on Measurement of Performance Parameters of Electric Two Wheelers

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Abstract*: The majority of vehicle dynamometer test rigs available in the market are constructed rigidly, which can test vehicles with a very narrow range of specifications, so there is a need for provision for load compensation to simulate the vehicle to test different vehicles & enhance the quality of measured results. The fabricated twin-roller arrangement enhances tire-roller contact, and the provision for balancing system inertia with the vehicle's inertia improves the accuracy of results during testing. This makes the chassis dynamometer system a versatile test rig for testing various vehicles and parameters, including speed, acceleration, brake force, and brake distance. A lightweight and low-base-inertia fabricated twin-roller arrangement is designed in the present study. An inertia flywheel arrangement is connected to it for load compensation during acceleration. An experiment is performed by loading an electric two-wheeler onto a two-roller setup to evaluate its performance. After testing an electric vehicle, the maximum speed and acceleration were determined to be 95 km/h and 3.65 m/s², respectively, and the brake force was 279.8 N with a braking distance of 7.48 m when applied from a speed of 20 km/h to zero. The obtained results were precise due to the vehicle's stability, provision for load compensation in the vehicle simulation, and the use of a mechanical inertia simulation with a flywheel in this test rig. This test rig can be used to test different vehicles with a wide range of test specifications.

Keywords*: Fabricated Roller, Twin Roller Chassis Dynamometer, Vehicle Equivalent Inertia, Load Compensation.

I. INTRODUCTION

In the current market, most chassis dynamometer test rigs are constructed rigidly and also take more time to load and unload vehicles on them before and after testing. To meet end-of-production line requirements, chassis dynamometers should be designed to reduce lead time, which can be achieved by minimising the setup time of vehicles on the test rig. The stability of the car and good tire-roller contact during testing are essential for obtaining precise and accurate results [1].

Manuscript received on 30 December 2022 | Revised Manuscript received on 14 February 2023 | Manuscript Accepted on 15 March 2023 | Manuscript published on 30 March 2023.
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The use of twin rollers reduces the loading and unloading time of vehicles, and also due to good tire roller contact, vehicles remain stable in testing, quality of measurements can be enhanced [2]. Michael W. Leifer studied how the use of a mechanical flywheel can provide the necessary inertia to the roller system whenever required to simulate the varying road load of different vehicles, and it is possible to test a wide range of vehicle specifications on the same chassis dynamometer test rig [3-4]. For the system to work effectively, he observed that the sum of tractive force developed by the set of rollers must be equal to the road load force & this can be done by balancing the inertia of the vehicle and the rollers' arrangement [5].

Making provision for load compensation during vehicle acceleration and deceleration on twin rollers is crucial to obtaining accurate and precise results [6]. To provide appropriate load compensation, the vehicle equivalent inertia due to longitudinal motion should be equal to the rotational inertia of the roller system [7]. Christian Marron found that it is challenging to keep the shaft of a solid roller concentric through the roller due to its weight and length, resulting in measured power and speed being lower than the rated power and speed, which ultimately reduces measurement accuracy. The use of fabricated rollers not only reduces the weight on the shaft but also has low inertia, so vehicles with a wide range of test specifications, including speed, acceleration, and brake force, can be tested on a test rig [8].

The vehicle equivalent inertia required for simulation is calculated. The fabricated roller arrangement model is designed through analytical calculations, and other components, such as the plummer block, bearing, timing belt, timing pulley, pulse pickup sensor, and wheel, are selected according to the required specifications. The final 3D model is created on Solid Edge solid modelling software. After the manufacturing, assembly, and procurement of components, the final physical model is created. Experiments were conducted on a physical model to test vehicle parameters and validate the results.

The objectives of the present study are:

- To provide appropriate load compensation during acceleration to get accurate and precise measurement of testing parameters like speed, acceleration, brake force and brake distance, etc.
- To make the test rig versatile equipment for

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testing vehicles with a wide range of specifications, such as power, wheelbase, tire width, vehicle inertia and top speed.

II. ANALYTICAL WORK

To simulate a vehicle on a chassis dynamometer test rig, the vehicle's inertia due to its linear motion is converted into rotary equivalent inertia, as the car rolls over rollers during the experiment. This vehicle equivalent inertia must be equal to the inertia of the driveline on the chassis dynamometer to get accurate results and provide load compensation during acceleration.

One method to determine the rotational inertia of a vehicle that corresponds to the longitudinal displacement of the car is equalising the longitudinal kinetic energy of the car at a linear speed with the rotational kinetic energy of the roller at an angular speed, as shown in equation (1),

Linear kinetic energy = Rotational kinetic energy

$$I = mv^2/\omega^2 \quad (1)$$

where,

I is vehicle rotational inertia in kg · m².

m is a mass in kg and v is the linear speed of the vehicle in m/s,

$\omega = v/r$, In rad/s is the angular velocity of the roller.

From equation (1), the vehicle equivalent inertia calculated was 2.37 kg · m², and this must be equal to the chassis dynamometer system inertia to simulate the vehicle.

$$I = I_s \quad \text{and,}$$

$$I_s = I_b + I_f \quad (2)$$

where,

I_s is system inertia, I_b is the base inertia of the drive line & I_f is the Inertia of the flywheel system.

Once we know the mass of the vehicle to be simulated, we can calculate the vehicle's equivalent rotational inertia from equation (1), which is equal to the inertia of the roller system.

The fabricated roller is an assembly of a roller pipe, boss, flange, shaft, and other components. According to the ASTM 312M standard, the nominal roller pipe size, DN250 or NPS10, is available in the market with a diameter of 273.1 mm and a thickness of 12.7 mm. After machining, a roller pipe with a diameter of 265 mm and a length of 300 mm is obtained. Using a flange and boss, the shaft is assembled into a roller pipe to form a fabricated roller.

Tractive Force calculated from the Road Load Equation,

$$F = a + bv^2 + m(dv/dt) \quad (3)$$

where,

Component of rolling resistance, a=0.18*m

Aerodynamic drag, b=0.025,

Mass of the vehicle, m=135 kg, etc.

The tractive force calculated from equation (3) was 237.35 N.

Torque due to tractive force,

$$T = F*R \quad (4)$$

Torque value due to Tractive force was 31.45 Nm, calculated using equation (4).

Force due to the weight of the vehicle (W_v) and operator (W_o),

$$P = W_v + W_o \quad (5)$$

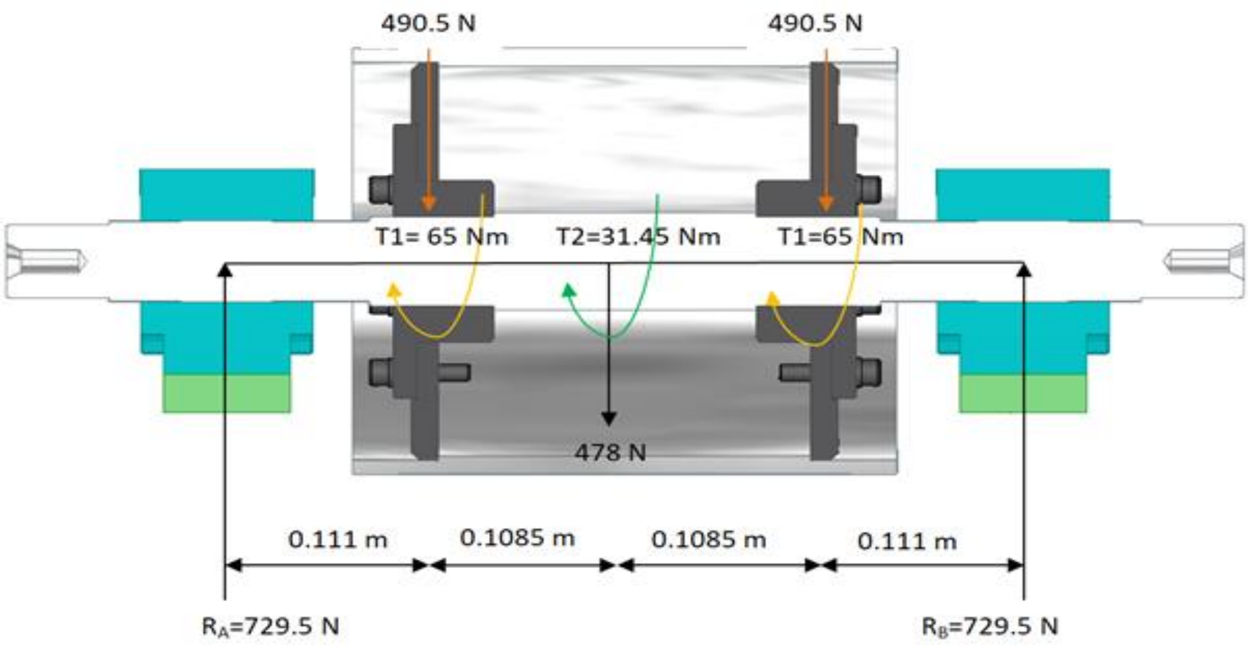

Fig. 1: Loading Diagram of Fabricated Roller

As shown in Fig. 1, the shaft of the fabricated roller was subjected to both bending and twisting. The load gets applied suddenly with major shocks when the vehicle is tested on twin rollers. There is a keyway on the shaft, which increases the stress concentration in running conditions. As a result of considering the above condition to which the fabricated roller was subject, the following equation (6) was derived to calculate the safe diameter of the fabricated roller's rotating shaft.

$$d = \sqrt[3]{pq} \tag{6}$$

Where, d is the safe diameter of the shaft in m, and factors p & q are calculated as per equations (7) & (8), respectively, given below,

$$P = \frac{16}{0.225\pi Y} \tag{7}$$

and,

$$q = \sqrt{\left[\frac{m}{4}(2SL - Pa)\right]^2 + [n(2T_1 - T_2)]^2} \tag{8}$$

where,
P is the total load on the fabricated roller in N,
S is a support reaction at any bearing end of the fabricated roller in N.
Y is the yield strength of the material in N/m²,
m & n are combined shock and fatigue factors due to the combined effect of bending and twisting moments in bending and twisting, respectively.
L is the CTC of bearing, a is the distance between the flange & centre of the shaft, and R is the outer radius of the roller pipe, in m.
T_1 and T_2 are torques due to load and tractive force, respectively, in Nm.
The most minor & safest diameter of the fabricated roller shaft calculated from equation (6) was 40 mm.

The base inertia of the driveline is the sum of all rotating components, including timing pulleys, brake disc, pulse pickup wheel, etc. Using 3D modelling software Solid Edge, this inertia was calculated to be 0.86 kg · m². Then, from equation (2), the additional inertia required for the system from the flywheel can be calculated, which is 1.51 kilograms · m².

Bearings and a Plummer block support the shaft, and its diameter at the bearing location is 45 mm, which is greater than the shaft's safe diameter. The SKF self-aligning ball bearing 1209 EKTN9 is selected, having a dynamic load rating of 7.8 kN and a limiting speed of 11000 rpm. This bearing is suitable for SKF's Plummer block SNL 509TL, which has a radial load carrying capacity of 160 kN.
The braking force calculated from the above equation,

$$F = ma \tag{9}$$

Where F is Brake force in N, m is the vehicle mass in kg, and a is the deceleration in m/s².

Given that the brake was applied at a speed of 20 km/h in two seconds, the calculated brake force from equation (9) was 375 N. The Kateel brake calliper KA-PA-150 was chosen because it has a brake disc thickness of 12.7 mm and can produce a minimum braking force of 1137 N when operated at 7 bar pressure.

The bearing should run for 30,000 hours at 1,800 rpm (90 km/h), and 90% reliability for the bearing should be considered.
The equivalent bearing load will be given by equation (10),

$$P = XF_r + YF_a \tag{10}$$

Where P is the Equivalent bearing load in N,

Fr & Fa are radial and thrust loads acting on the bearing, respectively, in N,
X & Y are factors.
As shown in the loading diagram, the bearing reactions are in the radial direction; therefore, only the radial load acts on the bearing, Fa=0, X=1, and equation (10) reduces to,
$P = XF_r$ and
$P = F_r = 490.5 + 478 + 490.5 = 1459N$
Dynamic load carrying capacity, C, can be calculated from the bearing life equation as follows,

$$L_{10h} = \frac{10^6}{60n}\left(\frac{C}{P}\right)^3 \tag{11}$$

Where, L_{10h} Is bearing life in hours, P is equivalent dynamic load in N, C is the Dynamic load carrying capacity in N, n is rpm of roller.

$$3000 = \frac{10^6}{60 * 1800}\left(\frac{C}{1459}\right)^3$$

$$C = 21589.22 \text{ N}$$

The dynamic load-carrying capacity of the SKF self-aligning ball bearing 1209 EKTN9 is 22900 N, with a bore diameter of 45 mm. The limiting speed of this bearing is 11000 rpm; therefore, this bearing is selected for this application. SKF Plummer block SNL 509TL is suitable for this bearing, which can support a radial load of 160 kN.
Dimensions of flywheel: dia. 400 mm and a length of 60 mm.
Material density 7850 kg/m³.
Mass of flywheel calculated from equation (12) given below,

$$m = \rho v \tag{12}$$

Where m is mass in kg, v The volume of the flywheel is in m³, and ρ is the material density in kg/m³.
$m = 7850 * (\pi/4)(0.4 * 0.4 * 0.06) = 59.2$ kg.
The Inertia of each flywheel is calculated as per the above equation (13),

$$I = 1/2 * mr^2 \tag{13}$$

$$I = (1/2) * 59.2 * (0.2)^2 = 1.2 \text{ kg.m}^2$$

For the two flywheels, the total inertia is 2.4 kg · m².

III. EXPERIMENTAL SETUP

A chassis dynamometer, also referred to as a twin-roller chassis dynamometer, served as the reference bench for this study and 3D model is shown in Fig. 2. The twin roller arrangement and its parts are carried by the roller base frame, which is built of ASTM A106 Grade B Steel tubes with a rectangular cross section of 61 mm by 122 mm. Two fabricated rollers with diameters of 265 mm and lengths of 300 mm are used; each roller is an assembly of flanges, shaft, bosses, hollow rollers, and other components made of EN24 T alloy steel.

These rollers are hard chrome-plated for long-lasting durability and are designed for testing two-wheeler vehicles with a maximum operating speed of 200 km/h. Four Plummer blocks, SKF make SNL 509TL, support the rotating

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shafts of twin rollers and are capable of carrying 160 kN of radial load. The SKF 1209 EKTN9 self-aligning ball bearing, assembled in Plummer blocks, has a dynamic load rating of 22.9 kN and a limiting speed of 11,000 rpm. To measure the speed of the roller, the 60-teeth pulse pickup wheel is keyed to the roller shaft, and the magnetic pulse pickup sensor is attached to the roller frame exactly below the pulse pickup wheel with a mounting bracket that works between a range of 5 and 15 volts. The gap between the pulse pickup wheel and its sensor was kept at 0.3 mm to measure the rpm of the pulse pickup wheel, or indirectly, the roller. The roller base frame is fitted with a normally open (NO) pneumatic brake from Kateel Engineering, model KA-PA-150. This brake assembly includes a brake caliper that is operated by a pneumatic air supply of 6-7 bar and an electric-operated solenoid valve that is employed to manage it on receiving a command from a PLC-based control system, and a brake disc made of mild steel that is keyed to the rotating shaft of a fabricated roller to bring the roller to rest after testing. Inertia flywheel arrangement consists of two inertia flywheels that are assembled on the same shaft, which is connected to the power roller shaft of the twin roller arrangement with the help of a timing belt and timing pulleys. These flywheels are made of cast iron, have a diameter of 400 mm, and a thickness of 60 mm, and can simulate an inertia equivalent of up to 2.4 kg/m2. A PLC Control Panel, including a commercial PC, is used to communicate with an operating pendant for controlling tests, a bar code scanner for vehicle model verification, a junction box and pneumatic system, etc.

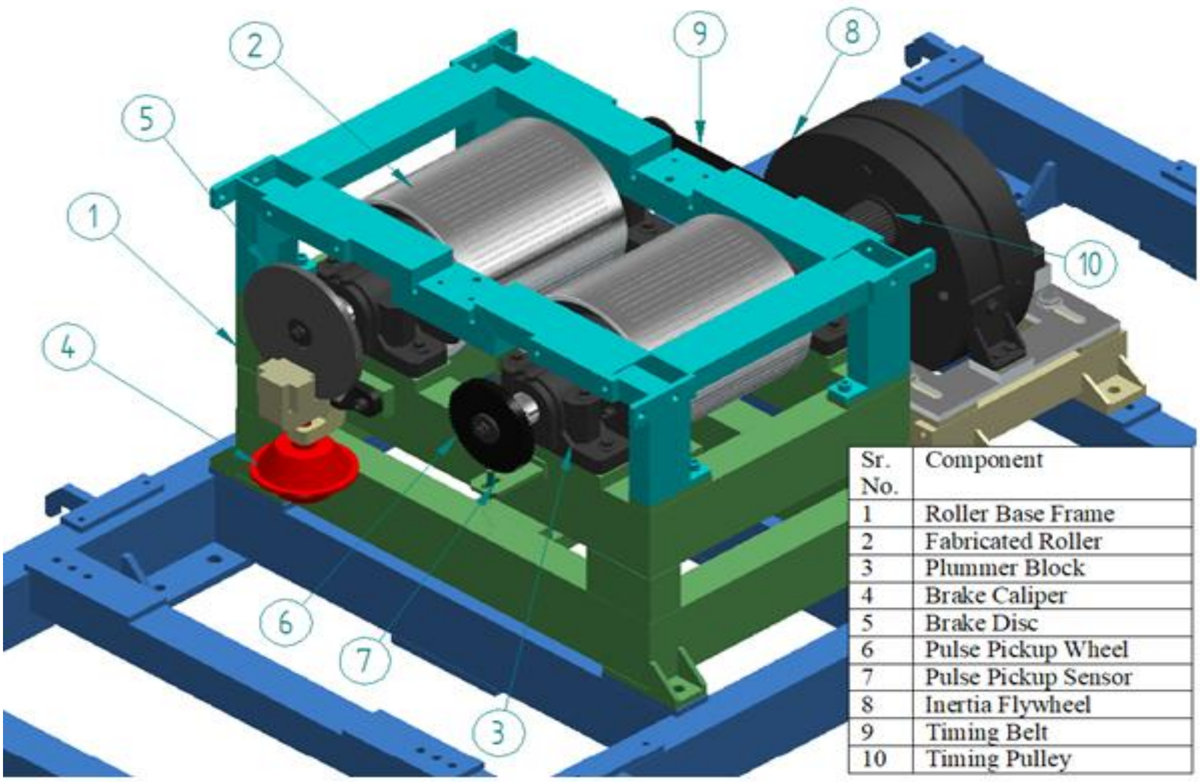

Fig. 2. Twin Roller Arrangement Setup

IV. EXPERIMENTAL RESULTS

To perform an acceleration test, a vehicle was run at different speed steps. The acceleration at each step was measured by determining the time required to reach each speed step, and the results are tabulated in Table I.

Table I. Acceleration Test.

Acc. Step	Lower Speed (km/h)	Upper Speed (km/h	Measu- red Time (sec)	Acceleration (m/s²)	Accele-ration Required (m/s²)	Varia-tion in Acc. (%)
1	0	10	0.76	3.65	3.70	1.35
2	10	20	0.85	3.26	3.26	0
3	20	30	0.85	3.26	3.08	5.84
4	30	40	0.93	2.98	2.93	1.71

The vehicle was driven at various speeds, and the readings on the speedometer were compared to the speed measured by the system; these values are shown in Table II.

Table II. Speedometer Verification & Max Speed Test.

Vehicle Speed on Speedometer (km/h)	Speed Measured on System (km/h)	Variation in Speed (%)
20	20	0
40	41	2.5
60	60	0
80	79	1.25
95	95	0

At 20 km/h, the brake is applied to bring the vehicle to a stop, and the time required to get the car to a halt is calculated. Then brake force and brake distance were calculated, and all measurements are shown in Table III.

Retrieval Number: 100.1/ijrte.E74290111523
DOI: 10.35940/ijrte.E7429.0311623
Journal Website: www.ijrte.org

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Table III. Brake Test.

Parameters	Required Value	Measured Value	Unit	Variation (%)
Brake Distance	7.5	7.48	m	0.27
Brake Force	280	279.8	N	0.07
Braking Time	2.75	2.68	s	2.55
Braking Speed	20	-	km/h	-

V. CONCLUSION

A chassis dynamometer prototype had been developed, and an electric two-wheeler vehicle was tested over it to evaluate its test parameters, which include speed, braking force, brake distance, and acceleration. The following conclusions were made from this study:

1) Load compensation was provided on the chassis dynamometer using an inertia flywheel during acceleration and speed tests. The maximum difference between the speed indicated on the vehicle's speedometer and the speed measured on a fabricated twin roller chassis dynamometer was observed to be 2.5%, which is very minor. Hence, it can be concluded that providing load compensation during acceleration is an effective method for obtaining accurate results in speed and acceleration tests.
2) The fabricated twin roller system's inertia was less than that of the vehicle being tested on it, and extra inertia was supplied from the flywheel.

   After performing acceleration tests, the maximum variation in acceleration was found to be 5.84%, and the brake force and brake distance values varied by 0.07% and 0.27%, respectively, when performing brake tests on the test rig, which are very close to the manufacturer's claim. Accurate results are produced by providing inertia via a flywheel; therefore, this equipment can be used to test different vehicles with a wide range of specifications, including vehicle inertia, power, and brake force.

ACKNOWLEDGEMENTS

I wish to express my sincere gratitude to SAJ Test Plant Pvt. Ltd., 72/76, Mundhwa, Pune, Maharashtra 411036, India, for providing me with the resources necessary for my study. I have great pleasure in expressing my deep sense of gratitude to Mr. Vikrant P. Jagtap for giving me the excellent opportunity to study in this organisation and encouraging me in my studies. I also express my appreciation to the working staff of SAJ Test Plant Pvt. Ltd., Pune, for providing me with valuable suggestions and guidance throughout my project work.

DECLARATION

Funding/ Grants/ Financial Support	Yes, this work is financially supported by the organisation SAJ Test Plant Pvt. Ltd., Pune-36, Maharashtra, India. (Website: https://sajdyno.com/)
Conflicts of Interest/ Competing Interests	No conflict of interest to the best of our knowledge.
Ethical Approval and Consent to Participate	No, the article does not require ethical approval or consent to participate, as it presents evidence that is not subject to interpretation.
Availability of Data and Material/ Data Access Statement	Not relevant.
Authors Contributions	All authors have equal participation in this article.

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