

Approval

The thesis titled "Analytical and Experimental Analysis of Wheel Alignment System for Light Vehicle" submitted by Riton Kumer Das, Student ID: 14PME002P, Session: 2014-2015, has been accepted as satisfactory in partial fulfillment of the requirements for the degree of Doctor of Philosophy in Mechanical Engineering on 2nd May 2023.

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Dedication

I dedicated this dissertation work to my beloved mother and wife, whose continuous assistance and encouragement helped me complete the thesis successfully.

List of Publications

Articles of the research work have been published in the following journals and conference proceedings:

List of Articles in the International Journal

1. R. K. Das, M. A. M. Hossain, M. T. Islam, and S. C. Banik , “ Effect of Tire Inflation Pressures, Loading and Shock Absorber Conditions on Fuel Consumption of Light Vehicle”, *Journal of Automation and Automobile Engineering*, (e-ISSN 2582-3159), Vol. 5, Issue 3, 2020.
2. R. K. Das, M. A. M. Hossain, M. T. Islam, and S. C. Banik, “Wheel Alignment and Fuel Performance Analysis of Light Vehicle Based on Road and Tire Conditions”, *Journal of Recent Trends in Mechanics*, (e-ISSN: 2582-3213), Vol. 5, Issue 1, 2020.
3. R. K. Das, M. A. M. Hossain, M. T. Islam, and S. C. Banik, “Effect of Front Right Toe-out Angles on Fuel Consumption for a Light Vehicle”, *South Florida Journal of Development*, Miami, Vol. 3, no 3. P.3724-3735, ISSN 2675-5459, DOI: 10.46932/sfjdv3n3-054, 2022.
4. R. K. Das, M. A. M. Hossain, M. T. Islam, and S. C. Banik, “Vehicle Dynamics, Lateral Forces, Roll Angle, Tire Wear and Road Profile States Estimation - A Review”, *International Journal for Engineering Modelling*, Vol. 35, no. 2, pp. 65-89, DOI: 10.31534/engmod.2022.2.ri.05b, 2022.
5. R. K. Das, M. A. M. Hossain, M. T. Islam, and S. C. Banik, “Effect of Tire Inflation Pressure and Front Total Toe Angle on Fuel Consumption for a Light Vehicle”, *International Journal of Automotive Technology*, Vol. 24, no. 5, pp. 1459-1473, DOI: 10.1007/s 12239-023-0118-2, 2023.
6. R. K. Das, M. A. M. Hossain, M. T. Islam, and S. C. Banik, “Effects of Front Total Toe-in Angle on Tire Wear and CO₂ Emission for a Light-Duty Vehicle”, Submitted with reviewers comment the International Journal of Vehicle performance.

Article in the National Journal

1. R. K. Das, M. A. M. Hossain, M. T. Islam, and S. C. Banik, “Effect of Wheel Alignment, Speed and Road Condition on Fuel Efficiency of a Light-Duty Vehicle”, *Mechanical Engineering Research Journal (MERJ)*, Vol. 12, pp. 1-7, CUET, Bangladesh, 2022.

List of Publications in Conference Proceedings:

1. R. K. Das, M. A. M. Hossain, and M. T. Islam, “Experimental Study on Vehicle Active Safety System Based on Wheel Alignment”, *National Conference on Energy Technology and Industrial Automation (NCETIA2018)*, 8th December, 2018, Chittagong, Bangladesh.
2. R. K. Das, M. A. M. Hossain, and M. T. Islam, “Experimental Study on Wheel Alignment System of Light Vehicles”, *Proceedings of the 5th International Conference on Mechanical, Industrial and Energy Engineering (ICMIEE2018)*, 23-24 December, 2018, Khulna, Bangladesh.
3. R. K. Das, M. A. M. Hossain, M. T. Islam, and S. C. Banik, “Experimental Study on Wheel Alignment System and Fuel Performance for Light Vehicle”, *Proceedings of the 4th International Conference on Mechanical Engineering and Renewable Energy (ICMERE2019)*, 11-13 December, 2019, Chittagong, Bangladesh.
4. R. K. Das, M. A. M. Hossain, M. T. Islam, and S. C. Banik, “An Experimental Study on Wheel Alignment and Shock Absorber Conditions for Fuel Performance Analysis of a Light Vehicle”, *Proceedings of the 6th International Conference on Mechanical, Industrial and Energy Engineering (ICMIEE2020)*, 19-21 December, 2020, Khulna, Bangladesh.
5. R. K. Das, M. A. M. Hossain, M. T. Islam, and S. C. Banik, “An Experimental Study on Front Toe-in Angles and Its Affect on Fuel Consumption for a Light Vehicle”, *Proceedings of the 5th International Conference on Mechanical Engineering and Renewable Energy (ICMERE2021)*, 12 – 14 December, 2021, Chittagong, Bangladesh.
6. R. K. Das, M. A. M. Hossain, M. T. Islam, and S. C. Banik, “Variation of Fuel Efficiency of a Light Vehicle Based on Wheel Alignment, Vehicle Speed and Braking”, *Proceedings of the 6th International Conference on Engineering Research, Innovation and Education School of Applied sciences & Technology (ICERIE2021)*, 26-28 February, 2021, SUST, Sylhet, Bangladesh.
7. R. K. Das, M. A. M. Hossain, M. T. Islam, and S. C. Banik, “An Experimental Study of the Front Left Toe-in Angle on Fuel Consumption and Traveling Cost for a Light Vehicle”, *Proceedings of the 7th International Conference on Mechanical, Industrial and Energy Engineering (ICMIEE2020)*, 22-24 December, 2022, Khulna, Bangladesh.
8. R. K. Das, M. A. M. Hossain, M. T. Islam, and S. C. Banik, “An Experimental Analysis of the Effects of Front Total Toe-out Angle on Tire Life and Wear for a Light Vehicle”, *Proceedings of the 5th International Conference on Mechanical, Industrial and Materials Engineering (ICMIME2022)*, 20-22 December, 2022, RUET, Rajshahi-6204, Bangladesh.

Approval by the Supervisor(s)

This is to certify that Riton Kumer Das has carried out this research work under our supervision and that he has fulfilled the relevant Academic Ordinance of the Chittagong University of Engineering and Technology, so that he is qualified to submit the following Thesis in the application for the degree of DOCTOR of PHILOSOPHY. Furthermore, the Thesis complies with the PLAGIARISM and ACADEMIC INTEGRITY regulation of CUET.

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Abstract

Wheel alignment is an important factor that influences the performance of automotive vehicles. This study experimentally investigated the effects of wheel alignment, especially front wheel toe angle, tire pressure, vehicle load, and brake application, on fuel consumption and tire travel life for two different models of light vehicles. According to the analysis, the vehicle's wheel started to become out of alignment when its travel distance increased over time. It is observed that vehicle wheels became out of alignment due to changes in different factors, including vehicle load, road condition, brake application, tire pressure, suspension condition, etc. The experimental findings also demonstrate that a number of variables, including engine rpm, rolling resistance, energy consumption, fuel consumption, and tire wear of the vehicles, are strongly correlated with front wheel toe angles, tire pressure, vehicle load, and brake application. The test result shows that the increase in rolling resistance and energy consumption led to an increase in fuel consumption, tire wear rates, and travel costs. It is found that due to the misalignment of the front wheel toe angle, the car travels about 4.77 km (for left toe-in, 2.53°), 5.12 km (for left toe-out, -2.53°), 7.37 km (for total toe-in, 5.06°), and 7.63 km (for total toe-out, -5.06°) less for the same amount of fuel, and the KPL reduction rate is up to 38.22%, 42.24%, 73.99%, and 79.31%, respectively. In comparison to the front wheel's left toe-in angle, the fuel consumption rate is 4.02% higher at the front wheel's left toe-out angle. Additionally, it is found that when both wheels are at a toe angle, fuel consumption is a little bit higher. In the second experiment, the effect of tire inflation pressure on the fuel performance of a light vehicle at a front wheel total toe angle of 0.00° , 1.44° (toe-in), and -1.44° (toe-out) is investigated. The experimental finding shows that when tire pressure changes from over-inflated to under-inflated while maintaining a constant front wheel toe angle, the vehicle's engine rpm, rolling resistance, energy consumption, fuel consumption, and travel cost rates increase. In comparison to a total toe angle of 0.00° , a front wheel total toe-in angle of 1.44° results in a 3.62% increase in fuel consumption, while a front wheel total toe-out angle of -1.44° results in a 4.63% increase. In the third experiment, a Toyota Echo Plus-2ZZ-GE-02 light-duty vehicle is used to investigate the effect of vehicle load on fuel performance. It is revealed that when the vehicle load rose from low to high while

maintaining a constant front wheel toe angle, the engine rpm, rolling resistance, energy consumption, fuel consumption, and travel cost rates increased. The analysis results also reveal that the rolling resistance varies non-linearly with the vehicle load under steady-state conditions, while the fuel consumption varies almost linearly. Interestingly, energy consumption also increases almost non-linearly with vehicle load. In the fourth case, the same vehicle is used to investigate the effect of brake application on fuel performance. According to the test results, increasing the number of brakes applied (0 to 100 times) results in increased rolling resistance and fuel consumption. The rates of rolling resistance coefficient and rolling resistance increased to 198.79% and 198.79%, respectively, and the rates of energy consumption and fuel consumption increased to 198.79% and 81.74%, respectively. In the fifth case, the effect of wheel toe angle on tire wear is investigated experimentally. It is shown that when the vehicle's front wheel total toe-in angle is changed from 0.00° to 4.80° for the Toyota Corolla-2E-86, the engine rpm, rolling resistance, energy consumption, tire wear, and tire traveling life reduction rate are increased. It is observed that the rate of loss in tire travel life with regard to a condition without misalignment is up to 99.08% when the front wheel total toe-in angle is out of alignment (from 0.00° to 4.80°). The rate of increase in rolling resistance is found to be about 167.84%, and the rate of carbon dioxide emissions is nearly 40% higher for the car as the front wheel total toe-in angle increased (from 0.00° to 4.80°). It is also seen that tire outside lateral groove wear, tire inside lateral groove wear, and tire circumferential groove wear change (from 0.03 mm to 3.01 mm, 0.03 mm to 0.29 mm, and 0.04 mm to 4.35 mm) with the changes in the front wheel total toe-in angle (from 0.00° to 4.80°) after the vehicle's traveling distance and tire travel life are recorded to be 3,500 km and 1537.50 km, respectively. Finally, a regression model is proposed using the test data. Such a model might be useful to explain the relationship between the related factors and determine the rate of fuel consumption, tire wear, and travel costs. The light automotive vehicle is expected to benefit from these findings in the form of improved fuel economy, less tire wear, lower energy usage, and prolonging the tire's useful life if the front wheel toe alignment, tire pressure, vehicle load, and brake application are maintained properly.

সারসংক্ষেপ

স্বয়ংচালিত যানের কর্মক্ষমতা প্রভাবের জন্য চাকার এ্যালাইনমেন্ট একটি গুরুত্বপূর্ণ বিষয়। এই গবেষণা কাজে দুইটি ভিন্ন ধরন মডেলের হালকা গাড়ীর বিশেষ করে সামনের চাকার টো-কোণ, টায়ারের চাপ, গাড়ীর লোড ও ব্রেক প্রয়োগের ফলে জ্বালানী খরচ এবং টায়ারের ভ্রমণ জীবনকালের উপর কতটুকু প্রভাব পড়ে তা পরীক্ষামূলকভাবে তদন্ত করা হয়েছে। এই পরীক্ষামূলক ফলাফলে দেখা যাচ্ছে যে কিছু পরিবর্তনশীল প্যারামিটার যেমন- ইঞ্জিন আর.পি.এম, ঘূর্ণায়মান প্রতিরোধ, শক্তি ব্যয়, জ্বালানী খরচ ও টায়ারের ক্ষয় যা চাকার কোণসমূহ, টায়ার চাপ, গাড়ীর লোড ও ব্রেক প্রয়োগের সাথে দৃঢ়ভাবে সম্পর্কযুক্ত। আরও একটি পরীক্ষার ফলাফলে দেখায় যে যখনই ঘূর্ণায়মান প্রতিরোধ ও শক্তি ব্যয় বেড়ে যায় তখনই জ্বালানী খরচ, টায়ারের ক্ষয়ের হার ও ভ্রমণ ব্যয় বেড়ে যায়। ইহাতে পাওয়া যাচ্ছে যে যখনই সামনের চাকার টো-কোণ মিসএ্যালাইনমেন্ট অর্থাৎ সামনের চাকার বামের টো-ইন কোণ 2.53° ও টো-আউট কোণ -2.53° এবং মোট টো-ইন কোণ 5.06° ও টো আউট কোণ -5.06° বেড়ে যায়, তখনই একই পরিমাণ জ্বালানী ব্যবহার করলে ইহাদের ভ্রমণ দূরত্ব যথাক্রমে ৪.৭৭ কিঃ মিঃ, ৫.১২ কিঃ মিঃ, ৭.৩৭ কিঃ মিঃ ও ৭.৬৩ কিঃ মিঃ কমে যায় এবং ইহাদের কে. পি. এল কমার হার ও যথাক্রমে ৩৮.২২%, ৪২.২৪%, ৭৩.৯৯% ও ৭৯.৩১% বেড়ে যায়। তবে ইহাদের মধ্যে জ্বালানী ব্যবহার হার তুলনা করলে দেখা যায় যে, সামনের বামের চাকার টো-আউট কোণে জ্বালানী ব্যবহার হার সামনের বামের চাকার টো-ইন কোণের চাইতে ৪.০২% বেশী। ইহাতে আর ও পাওয়া যাচ্ছে যে উভয় চাকার কোণ যুক্ত হয়ে গাড়ী চালালে তখনই ইহার জ্বালানী খরচ কিছুটা বেড়ে যায়।

দ্বিতীয় পর্যায়ে আরেকটি পরীক্ষামূলক ফলাফলে দেখা গেল যে, একটি হালকা গাড়ীর সামনের চাকার মোট টো-কোণকে যথাক্রমে 0.00° , 1.88° (টোন-ইন) ও -1.88° (টো-আউট) এ সেট করে স্থিরভাবে রেখে যদি টায়ার চাপকে অতিরিক্ত স্ফীত থেকে কম স্ফীতের দিকে পরিবর্তন করলে তখন ইঞ্জিনের আর.পি.এম, ঘূর্ণায়মান প্রতিরোধ, শক্তি ব্যয়, জ্বালানী খরচ ও ভ্রমণ খরচ বেড়ে যায়। ইহাদের মধ্যে তুলনা করে দেখা গেল যে মোট টো-কোণকে 0.00° রেখে গাড়ী চালালে যে পরিমাণ জ্বালানী খরচ হয়, তবে 1.88° (মোট টো-ইন)-এ চালালে সেক্ষেত্রে ৩.৬২% জ্বালানী বেশী খরচ হয় এবং -1.88° (মোট টো-আউট) এ চালালে সেক্ষেত্রে ৪.৬৩%, বেশী জ্বালানী খরচ হয়।

তৃতীয় পর্যায়ে আরেকটি পরীক্ষার একটি টয়োটা ইকো প্লাস টু ডবল জেড হালকা গাড়ীর জ্বালানী কার্যক্ষমতার উপর গাড়ীর লোডের প্রভাব তদন্ত করতে ব্যবহৃত করা হয়েছিল। ইহাতে প্রকাশিত হল যে যখনই গাড়ীর লোড কম থেকে বেশীর দিকে যায় তখনই গাড়ীর ইঞ্জিন আর.পি.এম, ঘূর্ণায়মান প্রতিরোধ, শক্তি ব্যয়, জ্বালানী খরচ ও ভ্রমণ খরচের হার বৃদ্ধি পায়। ইহাতে আর ও অনুধাবন করা গেল যে ঘূর্ণায়মান প্রতিরোধ গাড়ীটির লোডের সাথে অরৈখিক ভাবে পরিবর্তিত হচ্ছে এবং জ্বালানী খরচ প্রায় রৈখিকভাবে পরিবর্তিত হচ্ছে।

চতুর্থ পর্যায়ে, গাড়ীটির ব্রেক প্রয়োগের ফলে ইহার জ্বালানীর কার্যক্ষমতার উপর কতটুকু প্রভাব পড়ে তা অনুধাবন করা হল। উক্ত পরীক্ষার ফলাফল অনুসারে দেখা গেল যে, যখনই ব্রেক প্রয়োগের সংখ্যা ০ হতে ১০০ গুণবৃদ্ধি করা হল, তখনই ইহার ঘূর্ণায়মান প্রতিরোধ ও জ্বালানী খরচ বৃদ্ধি পায়। ইহাতে আরও দেখা গেল যে তাদের ঘূর্ণায়মান প্রতিরোধ সহগ, ঘূর্ণায়মান প্রতিরোধ, শক্তি ব্যয় ও জ্বালানী খরচের হার যথাক্রমে ১৯৮.৭৯%, ১৯৮.৭৯%, ১৯৮.৭৯%, ও ৮১.৭৪% বেড়ে যায়।

পঞ্চম পর্যায়ে টায়ার ক্ষয়ের উপর চাকার টো-কোণের প্রভাব পরীক্ষামূলকভাবে তদন্ত করা হয়েছে। ইহাতে দেখা যাচ্ছে যে, যখনই টয়োটা করোলা টু-ই গাড়ীটির সামনের চাকার মোট টো-ইন কোণ 0.00° থেকে 8.80° পরিবর্তন হল তখনই ইহার ইঞ্জিন আর পি.এম, ঘূর্ণায়মান প্রতিবোধ, শক্তি ব্যয়, টায়ার ক্ষয় ও টায়ার ভ্রমণকাল কমানোর হার বৃদ্ধি পায়। ইহাতে আরও অনুধাবন করা গেল যে, টায়ারের ভ্রমণকাল প্রায় ৯৯.০৮% কমে যায় যখনই সামনের চাকার মোট টো-কোণ 0.00° হতে 8.80° তে পরিবর্তন হয়। ইহাতে আরও পাওয়া গেল যে, যখন ঘূর্ণায়মান প্রতিবোধের হার বৃদ্ধি পেয়ে প্রায় ১৬৭.৮৪% হয়, তখনই ইহার কার্বন-ডাই-অক্সাইডের নির্গমের হার প্রায় ৪০% বৃদ্ধি পায়। তবে ইহাতে আরও দেখা গেল যে টায়ারের বাহিরের পার্শ্বে গ্রাভ ক্ষয়, ভিতরের পার্শ্বে গ্রাভ ক্ষয় ও মাঝখানের ক্ষয় যথাক্রমে (০.০৩ মিঃ মিঃ হতে ৩.০১ মিঃ মিঃ, ০.০৩ মিঃ মিঃ হতে ০.২৯ মিঃ মিঃ ও ০.০৪ মিঃ মিঃ হতে ৪.৩৫ মিঃ মিঃ) পর্যন্ত পরিবর্তন হল যখনই সামনের চাকার মোট টো-ইন কোণ 0.00° থেকে 8.80° তে পরিবর্তিত হল এবং পরবর্তীতে গাড়ীটির ভ্রমণ দূরত্ব এবং টায়ারের ভ্রমণকাল যথাক্রমে ৩,৫০০ কিঃ মিঃ ও ১৫৩৭.৫০ কিঃ মিঃ রেকর্ড লিপিবদ্ধ করা হল। অবশেষে, পরীক্ষামূলক ডাটাগুলি দিয়ে একটি রিগ্রেশন মডেল প্রস্তাব করা হয়েছে। এই ধরনের মডেলটি সম্পর্কিত কারণগুলির মধ্যে সম্পর্ক ব্যাখ্যা করতে এবং জ্বালানী খরচ, টায়ার ক্ষয় ও ভ্রমণ খরচের হার নির্ধারণ করতে ইহা উপযোগী ভূমিকা পালন করবে। পরিশেষে জানা গেল যে যদি একটি হালকা গাড়ীর সামনের চাকার টো-কোণ এ্যালাইনমেন্ট, টায়ার চাপ, গাড়ীর লোড ও ব্রেক প্রয়োগ সঠিকভাবে রক্ষণাবেক্ষণ করা যায়, সেক্ষেত্রে গাড়ীটির জ্বালানী খরচ, টায়ার ক্ষয় ও শক্তি ব্যয় অনেকাংশ কমে যায় এবং টায়ারের জীবনকাল অনেকগুণ বেড়ে যায় ফলে অর্থনৈতিকভাবে লাভবান হবে এবং পরিবেশ রক্ষায় গুরুত্বপূর্ণ ভূমিকা পালন করবে।

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Nomenclature

Abbreviation

α_i	Values of the Front Wheel (Left/Total) Toe-in and Toe-out Angle (deg.)
$\bar{\alpha}$	Mean of the Values of the Front Wheel Toe Angle (deg.)
f_c	Values of the Fuel Consumption (ml)
$\bar{f}_c$	Mean of the Values of the Fuel Consumption (ml)
$r_{\alpha f_c}$	Pearson's Correlation Coefficient (Dimensionless) for Front Wheel (Left/Total) Toe-in and Toe-out Angle and Fuel Consumption
$r_{T_p f_c}$	Pearson's Correlation Coefficient (Dimensionless) for Tire Pressure and Fuel Consumption
A_r	Angle Rate (%)
V_s	Vehicle Speed (km/h)
μ_{RRC}	Rolling Resistance Coefficient (Dimensionless)
F_R	Rolling Resistance (N)/Rolling Resistance Force(N)
V_W	Vehicle Weight (N)
KPL	Kilometer Per Liter
T_d	Travelling Distance (km)
T_p	Tire Pressure (psi)
m_1, m_2	Regressing Coefficients
k_1, k_2	Intercept Values
E_c	Energy Consumption (kJ)
α_t	Slip/Toe Angle (deg.)
IRI	International Roughness Index
PCR	Pavement Condition Rating
t_{cgw}	Tire Circumferential Groove Wear(mm)
$\bar{t}_{cgw}$	Mean of the Values of the Tire Circumferential Groove Wear(mm)

$r_{at_{cgw}}$	Pearson's Correlation Coefficient for Front Wheel Total Toe-in/ Toe-out Angle and Tire Circumferential Groove Wear
RPM	rev/min
LG_1	Tire Outside Lateral Groove Wear (mm)
LG_2	Tire Inside Lateral Groove Wear (mm)
$CG_1, CG_2, CG_3,$	Tire Circumferential Groove (mm)
CG_4	
T_{tl}	Tire Travelling Life (km)
T_{cgd}	Tire Circumferential Groove Depth (mm)
T_w	Tire Wear (mm)
$T_{c/km}$	Travelling Cost Per Kilometer (USD/BDT)
T_p	Tire Price (USD/BDT)
Fp	Fuel Price(USD/BDT)
Tc in USD	Travelling Cost in USD
Tc in BDT	Travelling Cost in BDT
$\bar{t}_p$	Mean of the Values of the Tire Pressure (psi)
V_l	Vehicle Load (kg)
$\bar{V}_l$	Mean of the Values of the Vehicle Load (kg)
$r_{v_l f_c}$	Pearson's Correlation Coefficient for Vehicle Load and Fuel Consumption
b_a	Brake Application (times)
$\bar{b}_a$	Mean of the Values of the Brake Application (times)
$r_{b_a f_c}$	Pearson's Correlation Coefficient for Brake Application and Fuel Consumption

Chapter 1

Introduction

1.1 BACKGROUND

Automobiles are the most essential mode of transport in today's world. One important thing to remember is that when an automobile travels on the road, two vital elements are used: fuel and tires. These two crucial elements are directly related to the cost of the vehicle's travel. When a vehicle is in motion, it consumes fuel, and tire life diminishes with the distance traveled. It's interesting to note that the car industry accounts for one-third of global fuel consumption, which substantially impacts the global economy. A number of factors affect the vehicle's fuel consumption. One main factor influencing a vehicle's performance and fuel consumption is wheel misalignment. The wheels mostly cause misalignment because the rolling resistance under the wheels increases as the vehicle travels along the road. A variety of factors influence the magnitude of rolling resistance, including variations in wheel angles, tire pressure, wheel loading, brake application, vehicle speed, and so on. Tires and road contact are essential to the vehicle's operation. Consequently, the tire is the medium of contact between the vehicle and the road. It is used to transfer forces between the vehicle and the ground. It affects the vehicle's wheel alignment. As a result, it emphasizes the importance of wheel alignment in vehicle dynamics. The wheel alignment of a vehicle significantly affects its suspension system's kinematics. Wheel alignment, also known as tire alignment, can help prolong the life and performance of vehicle tires. It can also help to handle it by avoiding pushing in one direction or unusual vibrations when driving. Alignment involves adjusting the suspension attachments that connect a vehicle to its wheels. The wheels and tires are not being adjusted. Adjusting the angles at which the tires make contact with the road is the key to good alignment. These adjustments are used to reduce tire wear and ensure that the vehicle moves straight and true (no "pulling" to one side). Alignment angles can also be changed beyond the manufacturer's specifications to provide a certain handling characteristic. Angles in off-road and motorsport applications may need to be changed far beyond "normal" for a variety of reasons. Therefore, wheel alignment/wheel angle is a crucial vehicle dynamics issues for the automotive sector.

It has a significant role in a number of safety-related factors, including tire wear and the transfer of forces between the tire and the ground [1,2]. The measurement of wheel angles may be carried out in various ways, from simple mechanical or electronic sensor-based systems to more advanced computer vision-based systems, which are increasingly used in automotive workshops [3]. Some researchers have done this to determine the user's primary misalignment with the image and calibration procedures. Using low-cost infrared depth imaging equipment, this technology is used to automate wheel loading procedures on trim and final vehicle assembly lines, as well as wheel hub movement and real-time alignment parameter measurement. This study investigated the contactless evaluation of wheel angle using a 3D stereo vision system, and a simple and low-cost method employing a consumer-grade depth-sensing camera such as the Kinect is demonstrated. It directly uses point clouds generated by its range picture stream, and the technique's practicability is defined by determining whether the orientation can be aligned to the desired orientation using simple point cloud data and camber angle data that can be communicated using RS232, Bluetooth, or Wi-Fi. The experimental results of camera calibration and pose estimation using a single camera for application to a vehicle's wheel alignment system are presented, and a systematic and iterative calibration technique, an image estimation algorithm, and a camera simulator are developed to check the calibration findings and compare them to pose estimation simulations [4–7].

The most recent analysis of findings from several studies shows that wheel misalignment affects vehicle accidents, reduces fuel efficiency, and increases tire wear. The most significant part of an automotive maintenance program is wheel alignment measurement, which shows that automobile stability and safety decrease with increasing vehicle speed. The wheel alignment active safety system is proportional to the steering functions [1,8]. Wheel misalignments include improperly setting and adjusting wheel alignment functions and improperly moving steering function systems. It was also discovered that vehicle fuel efficiency decreases with increasing tire wear caused by vehicle misalignment. However, many wheel alignment techniques have been developed to improve tire life and reduce vehicle tire wear. They can also verify that the vehicle's wheels are correctly contributing to its straight running position.

The vehicle's wheel alignment active safety system functions (caster, camber, toe, and steering axis inclination) can be much more easily measured using an IR sensor. This

technique uses a computerized wheel alignment machine to measure the proper wheel alignment of heavy and light vehicles. The system had a straightforward circuit with low costs, high resolution, and high working reliability. The computer-based wireless automobile wheel alignment process is used to quickly transfer data, which is better than the traditional system [9]. Regularly examining and adjusting the wheel alignment system are crucial tasks for the vehicle's active safety system. The machine vision process has been widely used as a simple and accurate method of determining wheel characteristic angles [10].

Researchers have analyzed the wheel misalignment state using a camera picture and calibration method [4–7]. However, this wheel alignment procedure is inaccurate in adjusting the wheel angles. Another group of researchers [9–13] has shown that computerized machine vision-assisted wheel alignment can decrease vehicle accidents, improve fuel efficiency, and prolong tire life due to timely identification of misalignment. With the progress of automobile technology, the speed of a car has dramatically increased; however, the travel safety and vehicle stability of the vehicle are now issues that are continually discussed. Automobile safety and stability depend on several factors, tires being one of them. The tire is a primary component that directly interacts with the road surface. Its main job is to guide the wheel on the road and support the associated forces. It is worth noting that despite the pavement's roughness and the vehicle's dynamic motion, the tire is loaded with a combined force, including longitudinal, lateral, and vertical forces. The tire quickly became worn out due to the combined effects of pitch, roll, and yaw. In the past, researchers [14–16] have focused on tire wear reduction for enhanced tire life and increased vehicle fuel efficiency. Research has demonstrated that tire life is inferred when the tread is worn to a minimum depth or is irregular [17,18]. Patrick [17] found that about 75.4% of the tires assessed had uneven wear profiles due to incorrect tire-road contact effects. It is well known that incorrect tire-road contact is associated with irregularities in the pavement surface. The researchers also showed that tire safety, road vehicle stability, and driver and passenger satisfaction are related to the quality of tires, tire material, and appropriate tire size [8]. According to the literature, most researchers measured tire wear using various models, and in most cases, model-based analyses supported the experimentally measured data [19]. The quantity of tire wear is predicted numerically for use in tire design. The tire dynamic rolling analysis of a 3D-patterned tire model computes the tire frictional

energy rates produced in each driving mode. Using a power function wear model is based on laboratory rubber abrasion testing [16,20]. The study focused on tires, which have always been regarded as one of the most important parts of a car. While simultaneously producing the longitudinal and lateral forces needed for acceleration, deceleration, and cornering, the tires support the vertical weight of the vehicle. Even though tires have a straightforward toroidal shape, the interaction between the tire and the road presents considerable difficulties to car engineers [21].

Tire inflation pressure is another critical parameter that significantly impacts rolling resistance, fuel consumption, and tire life. It is evident that tire contact on the ground has an adverse effect on the vehicle's performance. Many research investigations show that vehicle tire inflation pressure is a crucial issue for traveling conditions because tire pressure variation changes rolling resistance, causing fuel consumption to fluctuate, as well as the effect of wheel angles, creating wheel misalignment. Danso-acquah [22] investigated the impact of vehicle tire pressure on fuel consumption and discovered that a pressure of 10 kPa consumed 5.3 liters of gasoline per hour, which is higher than a pressure of 40 kPa, which consumed just 3.5 liters per hour. It can be seen that the lower the pressure, the larger the contact area and rolling resistance, resulting in increased fuel consumption. Usamo [23] has explained how tire inflation pressure significantly impacts power loss due to rolling resistance. When the tire pressure changed from 0.83 MPa to 0.9 MPa at a specific speed and distance, the fuel consumption of the Anbessa city bus increased by 9.74%. Other studies, on the other hand, have discussed the importance of maintaining tire pressure in automobiles because low tire pressure results in poor fuel efficiency and long-term tire wear, requiring tire replacement on a frequent basis. Despite the suggested tire pressures, at which the model reduces fuel consumption by 17.6% of the fuel cost, the studies revealed that any adjustment in vehicle tire pressure resulted in additional fuel consumption [24–25]. Varghese [26] investigated how tire inflation pressure affects fuel economy. Aside from the benefits of decreased fuel efficiency, tire pressure is also important in determining vehicle handling and passenger comfort. It was found that increasing tire inflation pressures from 2 to 3 bars reduced fuel consumption by up to 5%.

According to numerous researchers' findings, if the vehicle's loading is less or higher, the amount of rolling resistance is less or more. Thus, the fuel consumption is low or high, respectively. The extra weight fluctuation can affect tire pressure, negatively

affecting fuel consumption and vehicle performance [27–29]. It has been noted that due to overload, the tire contact surface experiences increased stress development, increasing rolling resistance.

Besides wheel alignment benefits for fuel efficiency, brake applications also reduce fuel efficiency and cause rapid wear and misalignment of the vehicle. Ahmed *et al.* [30] have investigated different speeds of vehicles and braking on the road based on the longitudinal tire slip and brake stop distances. The research has discussed the effects of driving speed and traffic safety [31]. Gregory *et al.* [32] and Sivaramakrishnan *et al.* [33] have presented their research work to identify the failure methods of various automobile components and focus on adequately replacing those methods. They say that regular maintenance can make the part last longer and work better. They also suggest a rigid ring-based dynamic tire model that can be used to study the effect of high-frequency, short-wavelength road disturbances on tire braking performance and the effect of the anti-lock braking system (ABS) on high-frequency tire vibration. Hou *et al.* [34] conducted the two most common V-B (Vehicle- Bicycle) conflicts that were discussed in previous studies. They used a driving simulator to investigate parameterized motion patterns and drivers' braking behaviors described by car speed, time-collision, bicycle speed, and lateral distance. Ryzhikov *et al.* [35] and Dhembare *et al.* [36] presented the car's differential braking system and maintained equal braking on all wheels. They commented that using the word "anxious" during braking action in the automobile industry over the past decades indicates a complex dynamic problem.

From the above discussions, the analysis results did not reveal that the light vehicles' front wheel toe angle changes affect fuel efficiency and tire wear at a different rate. As a consequence of the present study, trial results revealed that changing the stipulated front wheel toe angle has a distinct effect on fuel efficiency and tire wear rate. Also, the research has focused on tire pressure variation with toe angle change, load variation with toe angle change, and brake application variation with toe angle change. Numerous studies have been conducted to determine how rolling resistance, angle, lateral force, and longitudinal force affect wheel alignment functions. They mainly focused on individual relationships in their research. In this study, a number of correlation factors for the functions of wheel alignment, such as wheel angles, tire pressure, tire size, tire wear, suspension condition, road condition, load condition, and brake application, are considered. The experimental investigation is carried out to investigate the effect of

wheel alignment, specifically the front wheel toe angle, on rolling resistance, energy consumption, tire wear, and life, the lifespan of the suspension components, fuel efficiency, passenger comfort, vehicle handling performance, and overall maintenance costs for a light-duty vehicle.

1.2 EFFECT OF DIFFERENT PARAMETERS ON FUEL CONSUMPTION AND TIRE WEAR

1.2.1 Effect of Front Wheel Toe Angle on Fuel Consumption

According to the findings, the front wheel toe angle is proportionally correlated with the angle rate and rolling resistance. It shows that increases in the front wheel toe angle led to an increase in rolling resistance, fuel consumption, and energy consumption. This is probably due to the lateral acceleration over the curve and the local stiffness of the surface. However, the rolling resistance also depends on various factors, such as vehicle type, tire properties, tire materials, tire pressure, vehicle load, brake application, operating conditions, etc. Note that to maintain the vehicle's steady speed, the engine speed would have to be increased in order to overcome the excessive rolling resistance generated due to the misalignment of the wheels (front wheel toe angle). Hence, energy consumption would be higher as more fuel was burned. It's worth noting that a few degrees of toe angle, equivalent to moderate-high cornering accelerations, can nearly double the rolling resistance [37]. It was observed that tire rolling resistance has a considerable impact on vehicle fuel economy. It is important to note that a positive toe angle has advantages for greater vehicle stability when driving straight. According to Yurko [38], minor changes in toe-in angle (0° to 0.5°) should result in only a minor increase in rolling resistance. Therefore, it can be concluded that minor changes in toe angle are permissible. More changes in toe angle, however, lead to abrupt increases in fuel consumption, which is not recommended.

1.2.2 Effect of Tire Pressure on Fuel Consumption

Tire pressure plays an important role in determining vehicle handling and passenger comfort. Under or overinflated tires can cause accidents and excessive tire wear [39]. If the tire is operated at low pressure for an extended period of time, its life may be significantly shortened. Conversely, an over-inflated tire could cause damage when riding over irregularities in road surfaces, such as potholes and bumps, and comfort is

also marginally reduced. It is well known that rolling resistance is inversely proportional to tire inflation pressure. The increased tire inflation pressure results in decreased rolling loss, resulting in decreased fuel consumption.

1.2.3 Effect of Vehicle Load on Fuel Consumption

According to Madhusudhana and Nagaraja [40], vehicle load influences vehicle handling and accidents, as well as excessive tire wear and fuel consumption. The tire's life may be greatly prolonged if it is run at a low load for an extended length of time. An over-loaded tire, on the other hand, may cause damage while riding over abnormalities in the road surface, such as potholes and bumps, and comfort is also marginally diminished. The relationship between rolling resistance and vehicle load is widely understood. A higher vehicle load causes a higher rolling loss, which raises fuel consumption.

1.2.4 Effect of Brake Application on Fuel Consumption

It should be emphasized that brake application significantly impacts vehicle control, accidents, and fuel consumption [41]. The tire life may be greatly prolonged if the tire is operated with less braking application over an extended length of time. A higher braking application tire, on the other hand, may cause damage while riding over road surface imperfections like potholes and bumps, and comfort is also somewhat diminished. The relationship between rolling resistance and brake application is widely understood - more braking results in more rolling loss, which increases fuel consumption.

1.2.5 Effect of Front Wheel Toe Angle on Tire Wear

It is emphasized that front wheel toe angle is vital in determining vehicle handling as well as passenger comfort and safety. Excessive front wheel toe angle in cars can cause accidents and excessive tire wear [42]. If the tire is operated at the proper front wheel toe angle, the tire life may be greatly extended. On the other hand, an excessive front wheel toe angle tire may cause damage while riding over imperfections in road surfaces, such as pushing in one direction and unusual vibration. It is commonly understood that the front wheel toe angle is related to rolling resistance. A higher front wheel toe angle then results in more rolling resistance, which increases tire wear and decreases tire life.

1.3 IMPORTANCE OF WHEEL ALIGNMENT

Maintenance is crucial for the vehicle's performance and safe operation. Proper wheel alignment is one thing that plays an important role. Wheel alignment affects fuel efficiency, tire wear, performance, vehicle handling, and maneuverability. Wheel alignment is a significant factor in a vehicle's steering response. When tires and wheels are out of alignment, a number of problems can develop within a vehicle's various systems. The driver can easily observe whether the alignment is really bad or not. During misalignment, the vehicle pulls one way or the other. In other words, if the driver lets go of the steering wheel and the vehicle immediately steers to the left or right, there's likely an issue with vehicle wheel alignment. Misalignment may also cause vibration that a driver can feel, especially in the steering wheel. Simply put, it can be said that proper wheel alignments allow for safer driving. Proper wheel alignment also extends the life of vehicle tires. Incorrectly aligned wheels are one of the leading causes of uneven and/or premature tire wear. Uneven tread wear decreases the safe driving life of a tire or, in a worst-case scenario, may damage a tire beyond repair, requiring premature replacement [43].

1.4 ADVANTAGES OF WHEEL ALIGNMENT

The following are some of the advantages of wheel alignment:

- **Better fuel mileage:** One of the most significant advantages of wheel alignment is better fuel mileage. If the wheels are aligned the right way, the car does not need to work hard to move forward.
- **Easier to drive:** Another advantage of wheel alignment is that car driving becomes easier. When the wheels are aligned properly, the car does not keep pulling from side to side.
- **Tires will last longer:** If wheels are kept aligned regularly, tires will last much longer. When wheels are not kept aligned properly, tires wear down quickly, and replacing them can be expensive.
- **Less wear and tear:** Getting wheels properly aligned means less wear and tear. For example, if the wheels are not aligned, they need to press hard on the brakes to keep the vehicle straight, putting more pressure on them [44].

1.5 CAUSES OF WHEEL MISALIGNMENT

Wheel misalignment is one of the biggest issues for vehicle safety and stability. The wheels may go out of alignment due to a number of different factors. They frequently have to do with unsafe driving practices, severe road and weather conditions, etc., and also shield the car's suspension system from unplanned maintenance. Some of the common causes of wheel misalignment are listed below.

Wear and Tear: A car's suspension and steering systems can develop wear and tear over time. Bushings, ball joints, tie rods, and control arms are a few examples of parts that might wear out and lead to alignment issues. Due to worn or damaged suspension parts, such as ball joints or control arms, result in camber misalignment. Figure 1.1 shows a typical tie rod at the beginning stages of buckling or slight bending rather than at the catastrophic point of failure.

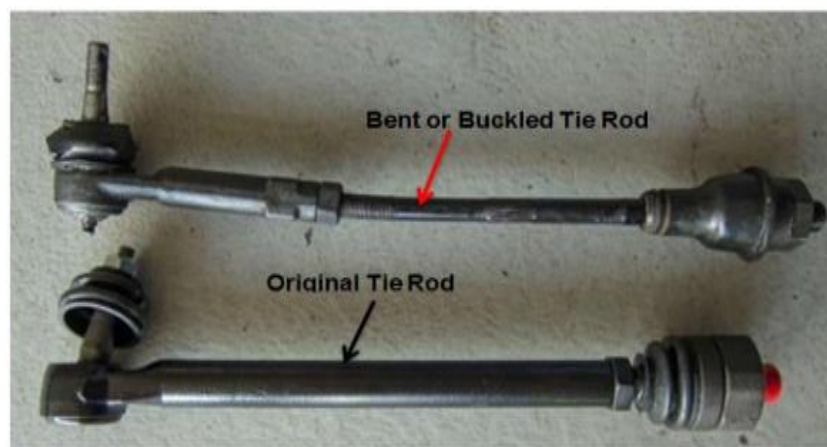


Figure 1.1 Illustrates a bent tie rod (top) to indicate failure with original one at the bottom [45].

Potholes and Bumps: Hitting potholes, speed bumps, or other road hazards can exert a significant force on the wheels and suspension system. These impacts can cause the suspension and steering components to shift, resulting in wheel misalignment. Such road hazards are more likely to be attempted when driving aggressively or off-road. When the suspension is not in good condition, it causes wheel misalignment because the alignment function is strongly connected to the suspension components.

Accidents and Collisions: The car's frame, suspension, or steering parts might be damaged in any collision or accident, no matter how slight. The damage may cause

wheel misalignment even if it is not immediately visible. Intense collisions can damage suspension parts like springs or struts, which can cause misalignment.

Improper Adjustment during Maintenance: Improper wheel adjustment may occur during manual wheel alignment checking, tire rotations, or suspension component replacements. These processes can potentially cause misalignment if the alignment is not properly calibrated or adjusted.

Incorrect Vehicle Body Height by Modified Suspension: Modifications to the suspension system, such as lowering or raising the car's vehicle height, can have an impact on wheel alignment. Altering the suspension geometry without compensating for the changes can lead to misalignment.

Uneven Tire Wear: Uneven tire wear patterns may be a sign of misalignment. Inadequate tire pressure, a lack of tire rotation, or worn-out suspension parts can all contribute to uneven tire wear. It might worsen the misalignment problem and cause further tire damage if ignored.

Environmental Effect: Due to environmental influences, the tire and car body will last a shorter period of time, which leads to unexpected wheel misalignment. Various joints and welding locations on the vehicle may be slack or worn out due to environmental effects.

Manufacturing Defects: In some cases, manufacturing defects or errors in the assembly process can cause wheel misalignment. These defects can include improperly drilled holes for suspension components or inaccurately positioned subframes. Though relatively rare, such issues can cause misalignment from the beginning.

1.6 THE EFFECT OF DIFFERENT ALIGNMENTS ON WHEEL PERFORMANCE

Tire wear is a highly complicated process. Several factors can contribute to tire wear. These include improper inflation (low tire pressure due to outer edge wear), poor alignment, vehicle overloading, and old shocks or struts. In conventional research, experiments are used to estimate tire wear. On the other hand, tire vibration and modal analysis can predict tire wear. Figure 1.2 shows various alignment settings on the tire.

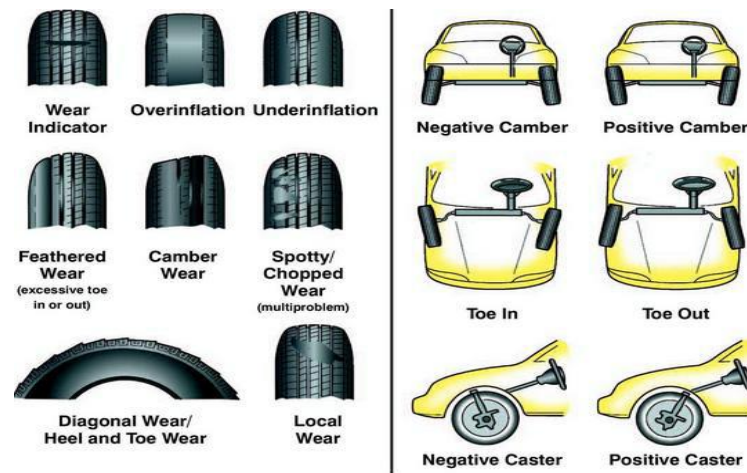


Figure 1.2 Effect of different alignment angles on tire [46].

1.6.1 Toe Angle's Impact on Vehicle Performance and Tire Life

The toe angle significantly impacts tire tread life, straight-line stability, and corner entry handling characteristics, which are a vehicle's three main performance aspects.

- Excessive toe-in, tire outside edge wear increases.
- Excessive toe-out, tire inside edge wear increases.
- Uneven tire wear increased rolling friction on the wheels

1.7 TIRE PRESSURE IMPACTS ON VEHICLE PERFORMANCE AND SAFETY

Tire pressure is an important consideration for any vehicle. It primarily causes rolling resistance, which results in uneven tire wear and increased fuel consumption. It is obvious that tire pressure is an important aspect of a vehicle. It is worth noting that maintaining standard tire pressure helps to maintain vehicle safety, improves vehicle performance, reduces fuel costs, and ensures tire lifespan. The following section discusses how three types of tire inflation pressure impact a vehicle's performance, as shown in Figures 1.3 and 1.4.

Corrected Inflation: Properly inflated tires improve vehicle control by increasing contact with the road surface. It improves traction, which is necessary for safe driving. In wet or slippery weather, traction is very vital. It has increased contact results in better braking and turning performance. Tires that are correctly inflated will also last longer and provide a smoother ride. All of these factors contribute to a more enjoyable and safe driving experience.

- Effect of proper inflated tire:
 - One of the most significant advantages of maintaining proper tire pressure is that it can help increase tire life.
 - Another advantage of maintaining proper tire pressure is that it may help customers save money on travel.
 - When tires are correctly inflated, fuel economy improves because there is less rolling.



Figure 1.3 Inflation pressure of tire [47].

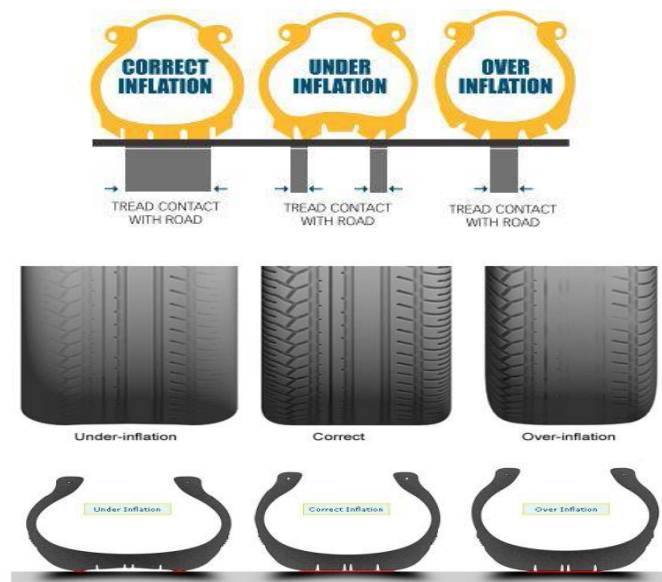


Figure 1.4 Inflation pressure of tire [40].

Over-inflation: Overinflated tires provide a bumpier ride. When a tire is overinflated, the outer tread pattern rounds out, causing the tire's center to wear out much faster than the outside edges.

- Effect of over-inflated tire:
 - It has a smaller contact area with the road.
 - It may affect the braking function.
 - By creating uneven tread wear, the center of the tire wears out faster.
 - Handling is a compromise.
 - It produces an unpleasant ride since the tire is unable to "soak up" the bump.

Under-inflation: One of the most common causes of tire failure is under-inflated tires. Low pressure indicates that too much of the tire is in contact with the road, significantly causing friction and reducing tire life.

- Effect of under-inflated tire:
 - It is possible to increase rolling resistance.
 - Increased fuel consumption as a result of rolling resistance, which causes the engine to work harder.
 - Uneven tread wear causes the tire's edges to wear out faster.

1.8 PERSPECTIVE OF THE RESEARCH ON WHEEL ALIGNMENT SYSTEM IN BANGLADESH

Very few automobile workshops currently maintain wheel alignment measurements and adjustments in our country. Nowadays, many light-duty vehicles on the road are found to be out of alignment. With the help of measuring the different angles of vehicles and making angle adjustments, the directional control of vehicles can be improved. Tire life and fuel performance, or mileage, depend on each other during road transportation. A routine wheel alignment check is the most effective way to control tire wear and tear. It can also impact fuel performance, suspension components, and driver fatigue. Therefore, it is necessary to maintain the wheel alignment measurement and adjustment according to standard mileage, and it is also necessary to find out the systems to overcome this situation very soon. In this research, the vehicle alignment system primarily focuses on analyzing the proper solution and guidelines to increase light vehicles' tire life and fuel performance.

1.9 MOTIVATION

Many researchers have identified vehicle misalignments using imaging tools, 3D stereo vision, camera calibration, and posture assessment approaches as on [4,6,7, 11-13]. It should be mentioned that wheel alignments consume a lot of time during the automotive assembly process and are crucial for optimum vehicle performance; moreover, the measuring tool's accuracy and the operator's expertise need improvement. Automobile manufacturers are trying to improve the assembly line process since wheel alignment could be a production bottleneck. The assessment of vehicle wheel alignment is influenced by a number of factors, including the alignment equipment, suspension design, and operator protocols [48]. Furthermore, recent investigations have revealed that misaligned wheels cause car accidents, reduce fuel efficiency, and increase tire wear. Thus, wheel alignment measurement plays a significant role in automotive maintenance programs. It has been demonstrated that automobile stability and safety decrease as vehicle speeds increase. Wheel alignment is accurately measured and adjusted with computerized wheel alignment, as mentioned in [1, 5-11] to increase vehicle performance. However, their analysis did not reveal that the light vehicles front wheel toe angle changes affect fuel efficiency and tire wear at different rates with respect to different toe angles. As a consequence of the study, trial results revealed that changing the stipulated front toe angle has a distinct effect on fuel efficiency and tire wear rate, and the research has focused on tire pressure variation with toe angle change, load variation, and brake application variation.

It has been the subject of numerous investigations to discover how rolling resistance, angle, lateral force, and longitudinal force affect wheel alignment functions. Their findings suggested that the effort was concentrated and found the maximum individual correlation. This investigation considered various correlation factors for wheel alignment functions such as wheel angles, tire pressure, tire size, tire wear, suspension condition, road condition, load condition, and brake application.

Finally, the research observed that a wheel alignment-related multiple correlation system has not yet been developed. However, some theoretical and analytical studies are available, but experimental research in this area has not yet been accomplished. For the experimental issue in this research, light vehicles are considered. Though light vehicles of electric and hybrid types are running on the road, only light vehicles powered by the S.I. engine are used in the study. While driving, the vehicles generally

encounter various drive resistances like rolling, air, and grade resistance. For simplicity, in this study, only rolling resistance is considered.

1.10 SPECIFIC RESEARCH OBJECTIVES AND POSSIBLE OUTCOMES

The specific research objectives and possible outcomes are

- To find out the causes of light vehicle wheel misalignment and solution to get vehicle active safety system.
- To analyze the light vehicle front wheel alignment method that greatly contributes to increase the tire life, fuel performance to ensure the travelling and driving stability.

1.11 OUTLINE OF THE THESIS:

This thesis paper, titled "Analytical and Experimental Analysis of Wheel Alignment System for Light Vehicle," constitutes the detailed experimental procedure, mechanics of the wheel alignment, analysis of the fuel performance of light vehicles as a function of wheel alignment, and different factors. The whole thesis incorporates six chapters, which are listed below.

Chapter 1: Introduction

This chapter discusses the background of the research, gives a brief introduction to vehicle wheel alignment, tire wear, load condition, brake application, tire inflation pressure, and indicates the motivation for this thesis. Additionally, the importance of wheel alignment and its effects on the state-of-the-art fuel consumption of the vehicle are highlighted.

Chapter 2: Literature Review

The scope and factors mentioned regarding wheel alignment in the preceding chapter are represented in this chapter. The detailed description of the literature review on wheel alignment, road condition, vehicle suspension condition, vehicle speed condition, tire pressure condition, tire wear condition, vehicle load condition, brake application, vehicle fuel performance condition, rolling resistance condition, and energy consumption condition is presented in this chapter.

Chapter 3: Mechanics of Wheel Alignment

In this chapter, the mechanics of wheel characteristics as well as the mechanics of wheel alignment measurement are presented. The details of determining different factors associated with vehicle performance are also presented in this chapter.

Chapter 4: Experimental Setup and Methodology

This chapter contains detailed experimental procedures, including the construction of a wheel alignment pit, detailed specifications of the wheel alignment machine, descriptions of equipment and useful hand tools, and experimental factors discussed.

Chapter 5: Results and Discussion

This chapter contains a detailed discussion of experimental results. Moreover, vehicle performance based on wheel alignment is discussed through proper graphs and tables.

Chapter 6: Conclusions and Recommendations of the Research

In this chapter, the research findings are summarized, along with its drawbacks and recommendations for further research.

Chapter 2

Literature Review

2.1 INTRODUCTION

Among the factors that affect the overall fuel consumption of a vehicle, wheel alignment is one of them. Vehicle wheel alignment is a crucial aspect of the automobile industry. It refers to the functions of the vehicle's active safety system, such as caster, camber, and toe angle. When the vehicle's active safety system is not properly functionalized, the overall performance of the vehicle is decreased, which means that the vehicle's performance is mostly dependent on fuel consumption, tire life, suspension component life, safety, and comfort. The vehicle's active safety system function changes due to wheel misalignment, which increases the rolling resistance of the automobile wheel and affects its ability to move straight. Besides this, the vehicle requires more energy for the accelerating pedal, resulting in higher fuel consumption and uneven tire wear. It is worth noting that the transportation industry consumes one-third of total global fuel consumption, which significantly impacts the world economy.

Numerous studies [4–7] show that wheel alignment detection systems can significantly improve vehicle performance. There are a number of techniques that have been reported in the literature to assess the wheel alignment of vehicles [5–13]. According to current research [9–13], the computerized machine vision wheel alignment method significantly influences the wheel misalignment condition and efficiently solves the vehicle's active safety system functions. This method also addresses the fact that proper wheel alignment can reduce accidents and tire wear and increase fuel efficiency. Though several techniques were available to adjust the misalignment of the wheels [1–13, 48–83], fuel efficiency needs to be investigated along with the causes of wheel misalignment. It is obvious that a single factor cannot define the fuel economy of an automotive vehicle; however, to a certain extent, it can be defined by several factors, including engine performance, tire size, tire pressure, tire wear, suspension conditions, load conditions, road conditions, speed, braking application, etc. This section includes the literature reviewed for the purpose of understanding the wheel alignment of the vehicle and its impact on fuel consumption, as well as how it is influenced by various

parameters such as road condition, load condition, suspension condition, vehicle speed, tire pressure, etc.

2.2 PREVIOUS STUDIES

2.2.1 Wheel Alignment System

The wheel alignment measurement system is one of the most important features of the vehicle maintenance schedule. The alignment system analyzes the wheel functions, which are an important component of the vehicle's stability and safety systems. Recent research shows that most road accidents are caused by improper wheel adjustment and that the steering function is not in a turning position. The condition becomes even more critical in the case of underdeveloped countries due to the lack of adequate planning and maintenance programs for wheel alignment. Nowadays, modern passenger vehicles have introduced different wheel alignment techniques for security reasons. The main purpose of the wheel alignment is to ensure the vehicle is running in a straight and correct position without pulling to one side or the other [50]. The vehicle wheel alignment system functions are the caster, camber, toe, and steering axis inclination (SAI) alignment, which can be easily measured using IR sensors [9]. A computerized wheel alignment machine is used to measure wheel alignment for heavy and light vehicles [52–61]. The process has some benefits, like a low-cost, simple electronic circuit, high resolution with better working reliability, and a better data transfer system. In recent years, the machine vision process has been extensively used as an easy and proper way to know the characteristic angles. The process is used to measure the active function of the vehicle safety system as well as to adjust the characteristic wheel angles [1–13, 78–83]. Wang and Tu [49] have explained the traditional four-wheel alignment techniques, including issues such as the requirement for several sensors, difficult calibration, particular operation, low inspection speed and accuracy, etc. Senjalia *et al.* [50] concentrated on a method for conveniently measuring wheel alignment using camera calibration and laser triangulation. Laser projection is common in structured contexts, and using this technology, a mobile operation can collect the data required for calibration. Yu *et al.* [66] reported on the state of the development of four-wheel aligners for automobiles for detecting wheel alignment parameters. Kim and Lee [51] analyzed a unique and effective method for inspecting vehicle wheel alignment for

suspension modules. Two laser modules and various mechanical jigs are used for wheel alignment inspection. The inspections were found to be accurate and cost-effective.

Xu *et al.* [53] focused on a new way to match calibration target photos, and their experiments show that the method's precision and reliability meet the requirements for vehicle wheel alignment for the vision detection system. Hu [54] suggested two new approaches for calibrating two cameras' extrinsic parameters. These two systems have distinct advantages, and the same data is collected from the two cameras in a non-public visual environment. Shao *et al.* [55] presented an explicit formula for estimating kingpin parameters, such as caster and kingpin inclination angle (KIA), in four-wheel alignment. The data showed significant changes due to local locations with high fractional number dimensionalization. The findings provide a scientific foundation for the establishment of alignment calibration standards.

There are numerous ways for measuring wheel angles currently in use, from simple mechanical or electronic sensor-based systems to modern systems based on computer vision, which have been considered one of the most precise techniques and are thus utilized by many automobile workshops [56]. Minaz *et al.* [57] observed that wheel alignments are often performed on both the front and back wheels of a modern person. Each tire analysis is specific and carried out by sensors using Raspberry Pi and Python concepts. Patel *et al.* [58] investigated the effort to understand the influences on static wheel alignment measurement accuracy for road vehicles. Vehicle alignments are the bottleneck in automobile and truck assembly lines; therefore, a better understanding of the difficulties is extremely beneficial. The alignment equipment used in their study was modified and adjusted to reduce external factors and vehicle data. The measurement procedures can be the most critical factor influencing vehicle suspension alignment assessment and adjustment accuracy and repeatability. Pundir [59] analyzed trends for the causes of car accidents, with special attention paid to the effects of wheel alignment. Wheel adjustment, which includes proper checking of wheel attribute angles versus vehicle company details, is an important task in the automobile industry because it limits irregular tire wear and has an impact on car handling and damage. Their work includes the design and evaluation of a 3D laptop computer view-based machine for the touchless restoration of automobile wheel calculation as well as a precise hint to attribute planes. Wach [60] evaluated the effect of tire pressure and additional vehicle weight that is not in accordance with the manufacturer's requirements on the outcomes

of wheel alignment measurement. Liang [61] discussed the advantages of a wireless optical car wheel alignment measuring system based on PSD (Position Sensitive Detector), including a simple circuit, low cost, high resolution and precision, high operating dependability, etc. The infrared transmission technique is used to transmit command and measurement data between the master and slave computers.

The wheel alignment substantially impacts a vehicle's active safety since it impacts stability and steerability. Wach and Skrzyniowski [62] examined the effect of measurement conditions on the results of a wheel geometry check. They suggest that the wheel geometry needs to be checked and adjusted on a regular basis. Another study [63] described the influence of kingpin inclination angle and wheel offset on several vehicle performance measures, such as steering effort, vehicle handling, and steering system vibration. It should be noted that vehicle steering and handling characteristics can be simulated in a virtual environment, which saves product development costs and time. Steering Axis Inclination (SAI) angles have the benefit of enhancing steering wheel returnability and lowering steering effort [64]. The higher steering axis inclination (SAI) angles improve the steering wheel return ability and reduce steering effort [65]. Kashyzadeh *et al.* [67] examine the fatigue life of a vehicle steering knuckle with variations in wheel primary angles, such as camber and toe angles. They observed that the steering knuckles highest and lowest fatigue life are related to the values of ± 2 degrees of camber, respectively. The effects of altering caster offset and kingpin offset on a long-wheelbase bus are investigated by Jambukar and Sujatha [68]. Handling reaction metrics are assessed for three different maneuvers: straight-line driving, steady-state circles, and double-lane changing. It is noted that the handling characteristics of a car while cornering are more critical than when traveling straight [69]. Vijayakumer and Ganesh [69] investigated the potential use of vehicle lateral and yaw dynamics to examine vehicle handling performance. Park and Sohn [70] proposed a new electronic camber suspension mechanism to improve cornering performance by controlling suspension geometry rather than brake and driving torques. The proposed system reduced camber angle formation and camber thrust in the vehicle. This allows the car to move straight forward. Cho [71] analyzed the steering drift, which is determined by gyro and wheel tracker measurements of dynamic wheel alignments. After braking force is applied at high speed, the toe angle at the front wheel is initially in a toe-in direction owing to longitudinal braking force. The toe angle then becomes

toe-out owing to the wheel bump stroke. Gajek and Strzepiek [72] reported the theoretical approach and results of an investigation of the wheel alignment check method on the sliding slip plate stand. They also estimated the precision of the measurements and the impact of the test conditions on the plate stand. Dong *et al.* [73] demonstrated that the front-wheel alignment parameters of a vehicle are generally determined based on dynamic suspension system performance or empirical equations. Their results showed that the proposed approach synchronously ensured braking and handling stability.

According to Wei *et al.* [74] investigation, the front and rear wheels' toe-in and camber are two crucial criteria influencing automobile tire abrasion, driving and braking stability, front wheel shimmy, etc. Their observations suggested that it is challenging to establish the exact relationship between toe-in and camber angle. The KTH project, which Karanja and Skoog [76] presented, looked at how camber and toe angles affected rolling resistance. Sleipner, a prototype car competing in the Rotterdam Shell Eco Marathon, was used for the tests. Researchers found that a toe-in improves straight-line stability while a negative camber angle enhances cornering stability. Nasada [77] outlined how improving a vehicle's stability, controllability, and ride comfort involves paying close attention to suspension compliance, compliance steer, and wheel alignment variance. A shock absorber and a knuckle are exposed to the bending force from the tire in the McPherson strut suspension, and the stiffness affects compliance, steer, and alignment variation.

Das *et al.* [78–83] presented a computerized and computer vision-based automobile wheel alignment measuring system to investigate the fuel consumption of an automotive vehicle by considering several factors. Their results suggest that the regular checking of wheel alignment by advanced technologies significantly improves fuel performance, tire life, tire safety, and vehicle handling satisfaction. Das *et al.* [83] analyzed that when the vehicle's front wheel toe-in angle is misaligned from 0.00° to 2.53° , the automobile traveled 4.76 km less for the same amount of fuel. In their studies, they created a mathematical model to predict how much fuel various car models will consume.

2.2.2 Effect of Road Condition on Vehicle Performance

The road surface condition, or roughness, is another important factor that directly affects the tires' lifespan and the vehicle's fuel efficiency. Pavement surface roughness is normally measured using irregularities in the pavement surface through the International Roughness Index (IRI). It should be noted that the lifetime of the tires is inferred when the tread is worn to a minimum depth or has uneven wear [21]. Patrick [17] found that about 75.4% of the tires assessed had uneven wear profiles due to incorrect tire-road contact effects. It is well known that incorrect tire-road contact is associated with irregularities in the pavement surface. The researchers also showed that tire safety and road vehicle stability, as well as driver and passenger satisfaction, are related to the quality of the tire, tire material, and appropriate tire size [2,8].

Hong and Oguchi [84] demonstrated a well-balanced design policy that takes into account drivers' perceptions and behaviors. They noticed that using the five-minute mean speed, detectors in the median lane on Japan's 7,000 km network of expressways only detect one passenger car. Ambros and Velentová [85] conducted GPS technology research, which has been chosen for data collection with the goal of using alignment consistency levels. Data processing is adapted to determine the core trajectory, horizontal alignment elements and segmentation, consistency measures, and identify inconsistencies. They reported their results in terms of the consistency level of individual curves, followed by confirmation using accident data analysis results. Nama *et al.* [86] investigated the geometric consistency of steep routes for traffic with poor lane discipline. On a four-lane split of National Highway 40 in India, 16 sites in steep terrain with distinct horizontal alignment and vertical profile features were chosen. According to the findings, 79% and 93% of the areas met Lamm's Criteria I and II, respectively. Sajjadi *et al.* [87] stated that over the last two decades, research has increased on optimizing roadway alignment. This report provides an overview of highway optimization models available in most parts of the world. It also shines a light on the elements that influence highway alignment optimization, with a great focus on safety characteristics. Taylor *et al.* [88] conducted fuel efficiency tests on a variety of pavement types and weather conditions in Ontario and Quebec, Canada. A highway tractor, a van semi-trailer, and a passenger vehicle were driven on the concrete, asphalt, and composite (asphalt over concrete) paved roads, and multiple regression analysis was used to analyze the data, which formed the basis for a set of predictive mathematical models.

Acqua *et al.* [89] found that the number of crashes decreases significantly as a road's design consistency increases. The study was conducted on the Tirrenia Inferiore State Highway in southern Italy without spiral transition curves between geometric tangent and circular elements on the horizontal alignment. Cheng *et al.* [90] conducted a road situation analysis, which included estimating and forecasting the position and size of numerous on-road impediments. They observed that current object detection and tracking techniques have trouble with real-time processing when given incomplete and confusing information. This study presented a development framework and unique algorithms for road situation analysis based on driving action behavior. The safety scenario was evaluated by simulating real-world driving behaviors. M. Di and associates [91] addressed the fact that the road centerlines of geometry definition can be determined using existing maps, static measurements (traditional surveys), or dynamic measurements (a GPS receiver mounted on a car). Their work aimed to establish the dependability of road alignment geometry based on the final application of road geometry recognition and to define the inherent boundaries of this integrated data measurement and processing approach.

Jimenez *et al.* [92] reported the numerous advanced driver assistance systems (ADAS) aimed at enhancing road safety that can benefit from digital mapping. This study talks about a fast and accurate way to measure using an on-board inertial device, as well as an algorithm for segmenting and fitting curves that meet current standards for highway design. Sümbül and A. Bögrek *et al.* [93] in their research discussed the road conditions and how vehicle acceleration-deceleration metrics were measured using a mems-based accelerometer. The driving qualities of the driver might be observed in this manner. The warning system engaged when the specified acceleration level exceeded, and the driver was notified. This resulted in lower fuel consumption, lower exhaust emissions, and a longer vehicle life. Das *et al.* [94, 95] experimentally investigated the effect of wheel alignment and fuel efficiency on the performance of light vehicles with the variation of vehicle speed under different road pavement conditions. The experimental results showed that the vehicle's fuel efficiency depends on the road's state and the wheels' alignment. However, good road pavement conditions with a recommended speed limit can significantly enhance the safety of suspension system components, reduce tire wear, boost vehicle mileage, and increase driver satisfaction. As shown in Figure 2.1, there is a little variation in fuel consumption on both fair and poor road pavement; however, the percentage of fuel losses became much higher in very poor road pavement conditions.

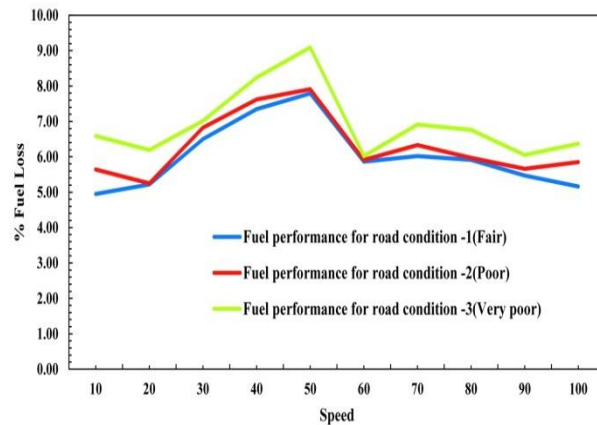


Figure 2.1 Comparison of fuel efficiency with respect to speed and road condition [95].

2.2.3 Suspension Condition on Vehicle Performance

The suspension system is an important issue for uneven and bumpy roads relating to wheel alignment functions. Researchers [96,97] have explained the vehicle's optimum performance based on suspension factors such as roll moment, understeer, oversteer, etc. Ikechukwu *et al.* [45] analyzed the tie rod function and tire wear conditions and found that the tie rod function is an important issue for wheel misalignment and driving stability. Consequently, the researcher Vadhe [98] suggested that the proper suspension components can greatly improve vehicle stability and fuel performance, considering cornering acceleration and bumping conditions. Belkhode *et al.* [99] demonstrated that almost all current automobiles feature independent front suspension, which implies that each front wheel is connected to the vehicle chassis separately. The steering geometry of the suspension determines the performance of the vehicle's front suspension, and kingpin angle, camber angle, caster angle, scrub radius, and toe-in and toe-out are characteristics of the steering geometry. Harsha *et al.* [100] found that all-terrain vehicle (ATV) suspension performance is critical. Their research aimed to create an ATV suspension system using the multibody dynamics technique and the Lotus Shark software. The suspension components have been modeled in Solid Works, and the analysis is performed in ANSYS Workbench. The camber and toe angles were evaluated in Lotus software, and specific toe and camber angles were chosen depending on performance. Deo and Suh [101] eliminated the linkage in the steering and suspension systems by making the WAP independent of suspension travel. The proposed solution employed a six-bar suspension, which eliminated the linkage. A new steering system compatible with the new suspension system had also been proposed, and this identified other couplings and DP redundancy

in the vehicle system and provided a framework for the construction of unique vehicles. Wang and Rui [102] described how the dynamics of the front road wheel directly impacted tire wear and steering wheel vibration, which in turn had a detrimental impact on customer satisfaction. The dynamic toe varied with parameters when there was an out-of-phase, non-coupled tire mass imbalance, as graphically illustrated in Figure 2.2. The outcome suggested that the dynamic toe is significantly or somewhat affected by the steering arm length, gear stiffness, ball joint friction, gear mesh load, and kingpin offset. Das *et al.* [103] analyzed the fuel performance with consideration of the effects of wheel alignment and shock absorber conditions. Their analysis results reveal that shock absorber conditions have greatly affected the wheel alignment. Wheel misalignment under tire inflation pressure and improper shock absorber conditions resulted in an increase in fuel consumption. In Figure 2.3, it is seen that the fuel consumption rate is much higher when the shock absorber is in bad condition (working load capacity of approximately 10%).

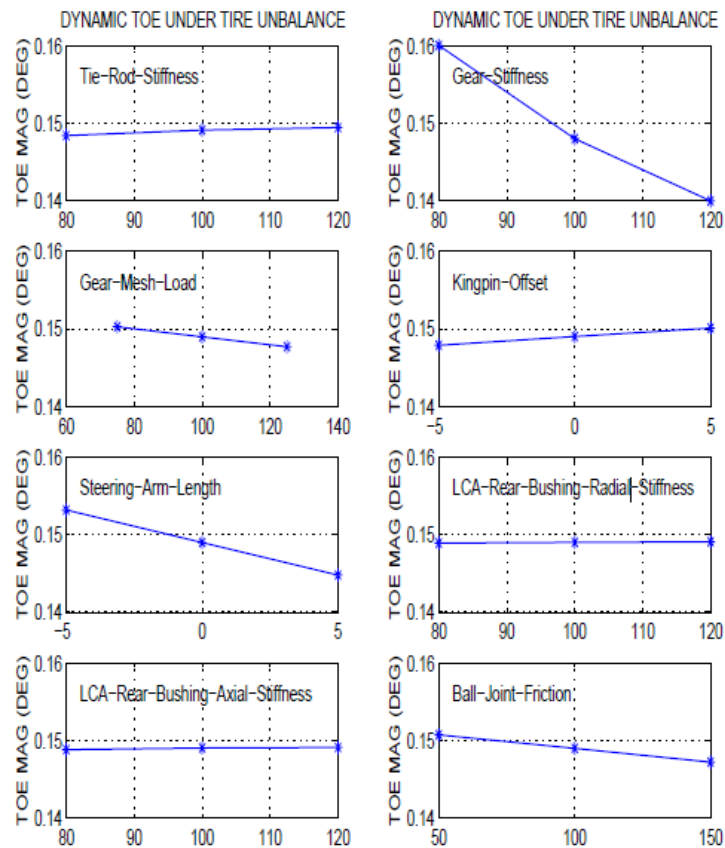


Figure 2.2 Effects of parameters under tire unbalance [102].

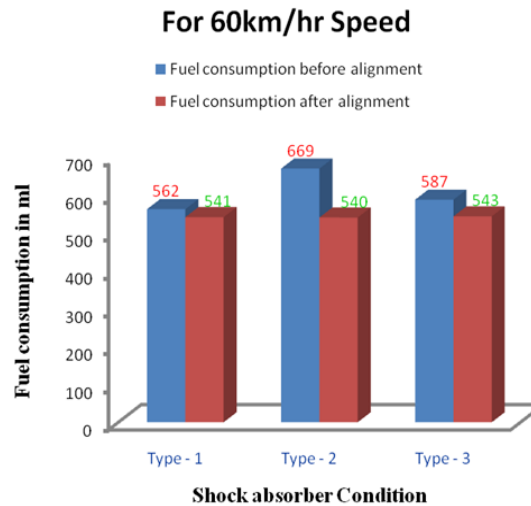


Figure 2.3 Shock absorber condition for fuel consumption [103].

2.2.4 Effect of Vehicle Speed on fuel consumption

Automobile researchers have tried to develop driving behaviors that ensure engine efficiency and road safety. Most studies characterized drivers' behavior according to their aggressiveness while driving. According to the study, aggressive drivers often cause unexpected speed changes that are dangerous on the road by driving at high engine revolutions and prompting fast starts and quick acceleration [104]. Due to this, different research studies have been conducted to control vehicle speed. Eco-driving leads to gradual acceleration, coast-down slowing, maintaining a constant speed, and avoiding high speeds. In eco-driving, a smooth and experienced pattern burns fuel statistically meaningfully. The design and size of speed control devices are the most crucial elements for effectiveness in lowering the maximum speed between devices, with speed humps/tables achieving a reduction of 50%–75% and trapezoidal shapes (speed tables) achieving a decrease of 10%–65% [105]. Baxter and Bumby [106] determined the underlying relationships between a number of measured variables and fuel consumption. For purely urban, suburban, and highway travel and an overall journey, 93.8, 92.4, 88.2, and 95 percent of fuel consumption were explained by the respective regression models. In tests over urban sections, the significant variables were found to be vehicle speed and deceleration, frequency of the R gear change, and ambient pressure. Moitzi *et al.* [107] found that fuel consumption increased linearly with working depth for both the moldboard plough and the short disc harrow but disproportionately for the subsoil. The influence of engine speed is shown in a separate

experiment with a universal cultivator. Li and Peng [108] discussed the fuel-efficient driving strategies of a traditional vehicle equipped with a continuously variable transmission in car-following scenarios. A minimal fuel consumption probability is formulated and numerically solved using the Gauss pseudo-spectral method. The optimal maneuvers demonstrated two types of behavior depending on the vehicle speed: pulse-and-gliding (PnG) and constant speed (CS). The differences in behavior are primarily determined by engine fueling characteristics and road load.

Wang *et al.* [109] found that fuel consumption per unit distance is optimum at speeds between 50 and 70 km/h, with fuel consumption increasing significantly with acceleration. In their study, a VSP-based model has been developed to calculate vehicle fuel consumption and produced good results compared to the measured data. Road transportation is a major consumer of fuel in China, as graphically illustrated in Figure 2.4.

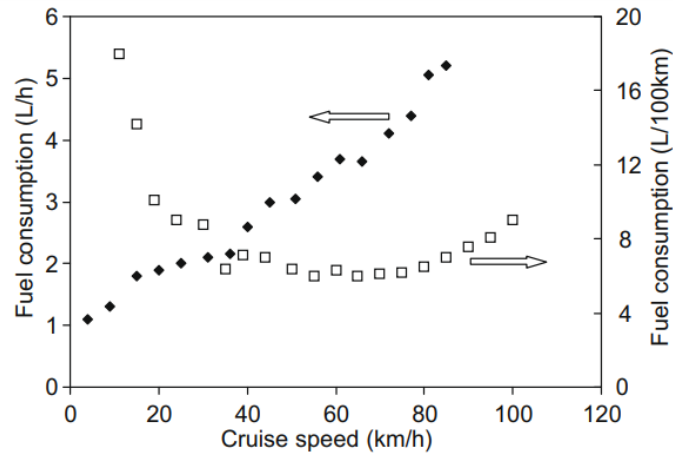


Figure 2.4 Influence of cruise speed on fuel-consumption rate for 2 test vehicle [109].

Wang and Yin [110] demonstrated that fuel consumption per 100 km decreases at first and then increases as vehicle speed increases. There is a quadratic fit between the two kinds of speed, which is consistent with the curve of the power-based emissions model, and PM_{2.5} is significantly related to instantaneous fuel consumption. However, no typical function can be fitted, indicating that vehicle emissions are not a decisive factor. Fleming *et al.* [111] focused on eco-driving assistance systems that incorporated predictive or feed-forward information as a promising technique to increase energy efficiency and reduce CO₂ emissions from road transportation. This work gives details of a system recently developed by the authors that uses real-time data from GPS and

automotive radar to perform a predictive optimization of a vehicle's speed profile. J. Hine [112] presented the New Highway Development and Management Model, HDM-4, which attempts to predict fuel consumption in combination with a simulated vehicle speed profile. It found that the IHCM model appeared to overestimate the Kijang's speed, although a close estimate of the truck's speed was obtained. In view of the promising results, it is proposed that relatively simple multiple regression models be used to predict fuel consumption under congestion.

2.2.5 Influence of Tire Inflation Pressure on Fuel Consumption

Apart from the benefits of wheel alignment, tire pressure also plays an important role in ensuring the stability and safe driving of the vehicle. It has been found that raising tire inflation pressure reduces local tire deformation and, consequently, the loss of hysteresis caused by deformation. The 660 automobiles surveyed in five different areas in the Kumasi Metropolis showed that 96.52% of the vehicle tire pressures deviated from the recommended tire pressures [113]. That means only 3.48% of these vehicles had measured tire pressures that conformed to the recommended tires. Adequately inflated tire pressure can save tire life up to 20%, which is nine months more of its lifespan, and it can also save fuel from 4% to 10% by increasing the braking efficiency up to 20% [114,15]. Though the process of tire wear is very complex, many researchers have shown that tire wear is due to incorrect inflation, alignment issues, vehicle overloading, and worn-out shocks and struts [1]. It is known that tire rolling resistance and inflation pressure are two factors that directly influence the fuel consumption of an automobile, and these factors are interrelated. Ejsmont *et al.* [115] indicated that tires with high rolling resistance are more sensitive to inflation pressure changes than tires with low rolling resistance. Mahendra and Rao [116] explained that lower tire pressure causes increased tire wear, heat production, and fuel consumption. Conversely, keeping tires at optimum pressure helps lessen running obstacles, reduce tire wear, and improve fuel efficiency. Eckert and Santiciolli *et al.* [117] discussed the rolling resistance force, which is primarily the result of tire deformation and is calculable as a function of various variables including tire structure, tire geometry, tire material, temperature, and filling pressure. At low speeds and on hard pavement, rolling resistance is the primary

resistance force. This research aims to study the influence of tire pressure in the calculation of the vehicle's power consumption.

Elfasakhany [118] concentrated on the tire pressure checking framework (TPCF), which is a system framework for measuring tire pressure. They discussed the dangers of having low and high tire pressures. The direct TPCF method has been claimed to be more accurate, but battery life and framework costs are affected. The indirect method used velocity and vibration sensors to monitor the tire's pressure. Laz and Elnour [119] are concerned about pneumatic tire services in Sudan, particularly in most developing countries. It also gives the condition of maintaining the correct cold tire inflation rate for different sizes, which helps in optimizing tire performance and fuel economy. Self-inflation and automatic tire pressure control devices have been developed to assist in maintaining constant tire pressure, which provides high fuel efficiency and reduces accident rates [120,121]. A pressure drop is detected by a hissing sound made by the tire, and the system will start refilling the tire automatically according to the tire's requirements [120,122]. Babu *et al.* [123] addressed potential improvements in gas mileage, tire wear, tire wear reduction, and tire handling and tire performance in diverse road conditions. Automatic inflation system that ensures the tires are inflated constantly. A central compressor will supply air to all four tires through hoses and a rotary joint between the wheel spindle and hub at each wheel hub. Singh *et al.* [124] developed the tire inflation system, which is based on the idea of maintaining uniform tire pressure on an automobile to reduce tire wear, increase fuel efficiency, and increase overall vehicle safety. Reiter and Wagner [125] demonstrated that the under-inflation of pneumatic tires, a typical problem in sedans and light-duty vehicles, also affects the vehicle's handling characteristics. An automated tire monitoring and inflation system can better accommodate handling requirements, and a quadruple lane change test revealed that the required steering wheel angle increased by up to 47.7% for front axle tire inflation pressures at 70% of nominal values.

Mushiri *et al.* [126] focused on the design in the report as functioning to restore the tire pressure on vehicles so that they are kept at optimum pressure levels, thus extending their lives and, at the same time, saving the owner from fuel costs and maintenance

costs incurred with underinflated tires. It is self-sustaining, using an independent power source within the vehicle. The system is monitored and controlled by a Java or Android program that detects low pressure and initiates compressor ON / OFF states. Mahesh *et al.* [127] stated that the pneumatic tire is a flexible structure in the shape of a tire filled with compressed air. The running tire is essential to studying ground vehicles' performance, ride, and handling behavior. Their work aims to examine the rolling characteristics of tires produced from different materials. Variations in tire inflation pressure, camber angle, and slip angle are used to study the dynamic behavior of tires. Ambrosio and Vitolo [128] presented inflation pressure management through an automatic, electro-pneumatic, central tire inflation system as a means of improving fuel consumption in passenger vehicles, as well as safety and drivability.

Elnour and Laz [129] mentioned that the correctly inflated tire provides appropriate support from the contained air pressure to provide an even distribution of load across the footprint. This significantly impacts tire wear, rolling resistance, and durability, as well as providing drivers with acceptable vehicle comfort and performance. In this research, they have taken into consideration all factors and the condition of the tire that have a direct effect on the tire's performance, stability, and safety. Machave *et al.* [130] emphasized the use of an experimental method to study the influence of pressure and load on wheels using RFT and finite element techniques. The most commonly used considerations in the design of the rotating body are elucidated. A potentially viable technique for modeling wheels subjected to loading is highlighted.

Das *et al.* [131] found that a drop in tire pressure from the recommended inflation pressure decreased fuel efficiency by 2.38 percent (when the front tire pressure is 30 psi and the rear tire pressure is 27 psi), 5.73 percent (when the front tire pressure is 28 psi and the rear tire pressure is 25 psi), and 8.84 percent (when the front tire pressure is 26 psi and the rear tire pressure is 23 psi). However, fuel consumption has significantly improved after adjusting the wheel alignment, even when the tire pressure is underinflated.

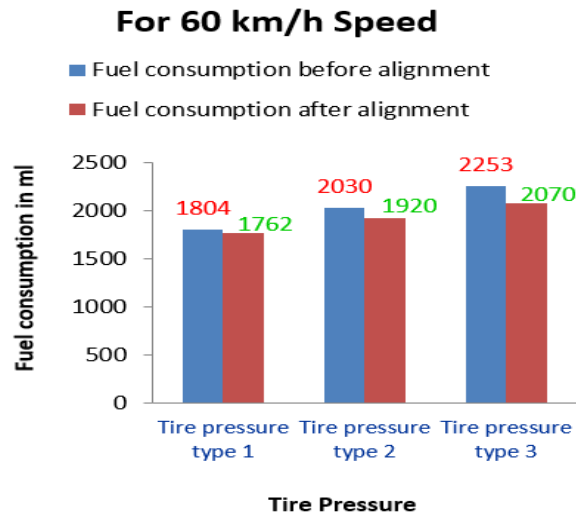


Figure 2.5 Fuel consumption with respect to tire pressure [131].

2.2.6 Effect of Rolling Resistance on Fuel Consumption

As discussed in an earlier section, the fuel efficiency of an automobile depends on a number of variables; among them, the speed of the vehicle, rolling resistance, the condition of the tires, tire pressure, and the alignment of the wheels are particularly frequent problems that affect fuel efficiency. Although rolling resistance has less emphasis in this work than wheel alignment-related vehicle fuel consumption, it is nonetheless important to look at how rolling resistance affects fuel economy. Also, it is important to look into how misaligned wheels affect rolling resistance. According to several studies, a 10% reduction in tire rolling resistance might have resulted in a 1-3% reduction in fuel consumption [132]. When a tire rolls on the road, mechanical energy is converted to heat as a result of the phenomenon referred to as rolling resistance. Rolling resistance plays an important part in increasing vehicle fuel consumption.

Figure 2.6 shows the variation of the rolling resistance coefficient with speed for different types of tire construction. In the case of radial ply tires, the increase in rolling resistance coefficient is very low until 80–100 km/h, beyond which the increase is slightly higher. For bias-ply tires, the rolling resistance coefficient increases significantly with an increase in speed. Figure 2.7 shows a decrease in RRC with an increase in the vertical load for different speeds. Michelin correlates this to the slight decrease in visco-elasticity due to an increase in load, as the higher bending and shearing at high loads results in increased temperatures.

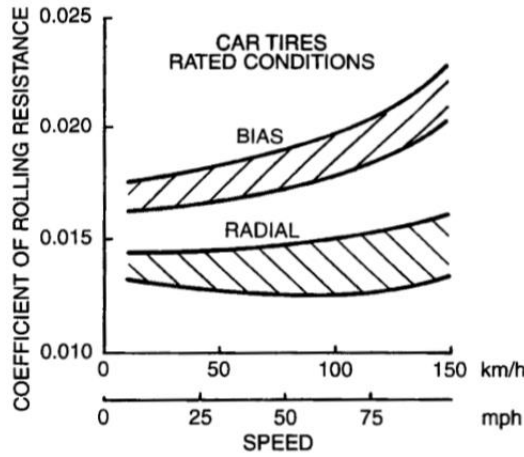


Figure 2.6 Coefficient of rolling resistance change with change in speed for bias and radial ply tires [133].

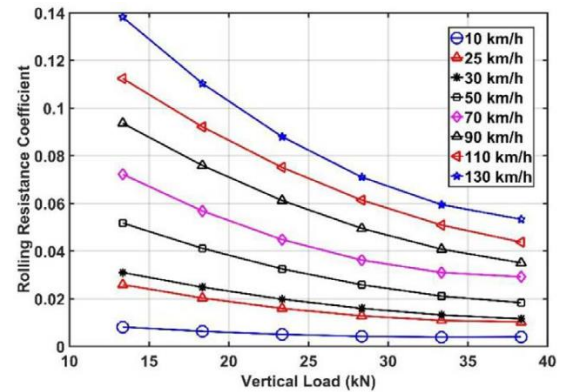


Figure 2.7 Variation of rolling resistance coefficient with vehicle load [134].

Ejsmont *et al.* [135] reported that 25% - 30% of the vehicle's energy consumption is lost due to rolling resistance. It should be noted that the rolling resistance coefficient used for pneumatic tires is directly related to tire inflation pressure. It has been discovered that raising tire inflation pressure reduces local tire deformation and, consequently, the loss of hysteresis caused by deformation. Ejsmont *et al.* [136] indicated that tires with high rolling resistance are more sensitive to changes in inflation pressure than tires with low rolling resistance. Mahendra and Rao [116] explained that lower tire pressure causes increased tire wear, heat production, and fuel consumption. Conversely, keeping tires at optimum pressure helps lessen running obstacles, reduce tire wear, and improve fuel efficiency. According to research by Vijaya *et al.* [24], there are noticeable variations in fuel consumption, tire wear, and maintenance costs when the tire pressure is low or high. Rajopadhye and Raju [137] presented the importance of improving vehicle energy efficiency in modern automobiles. It is even more vital for electric vehicles, which have limited range. The driving resistance forces of the vehicle determine its energy efficiency, and the rolling resistance makes up a sizable portion of that driving resistance and directly affects the vehicle's fuel consumption and energy efficiency. In addition, inflation pressure, speed, and load variation affect the rolling resistance and energy consumption of free-rolling passenger car tires. This impact is especially pronounced for short-distance travel, during which the tire does not completely warm up. An additional contribution due to the warm-up phase is 25-30 percent for short distances of 5 km and reduced to 2.5 percent at 100 km. The RRC

shows a negative correlation with changes in inflation pressure, load, and speed but a positive correlation with speed. Mashadi *et al.* [138] presented a large potential for decreasing energy losses due to tire rolling resistance, significantly reducing fuel consumption and greenhouse gas emissions, and improving tire tread durability. Grappe *et al.* [139] presented that the coefficient of rolling resistance for pneumatic tires depends on hysteresis loss from tire deformation, which is affected by the vertical force applied to the tires and the tire's inflation pressure. The coefficient of rolling resistance was determined with the coasting deceleration method from measurements performed in a level hallway. A simulation model shows that the changes in magnitude examined here have an important impact on competitive cycling performance. Wiel *et al.* [140] measured the rolling resistance of various wheels under a variety of camber and toe alignments. Their findings show that rolling resistance increases parabolically with toe misalignment and is only minimally affected by camber. Also, they found that a 1° toe-in/toe-out misalignment results in a 25.5% increase in rolling resistance.

Steyn and Ilse [141] discussed how tire rolling resistance has a major impact on vehicle fuel consumption. Rolling resistance is the loss of energy due to the interaction between the tire and the pavement surface. Their research investigated the relationship between pavement surface texture depths and tire/surfacing contact stress and area. The contact stress between the surface and the base layer and how texture affects the contact stress are discussed. According to research by Kelly [142], a 2.5% improvement in fleet fuel economy can lead to yearly energy savings of 3.8 billion gallons. Lower rolling resistance tires and better tire pressure maintenance can help achieve this improvement. The National Renewable Energy Lab has developed a vehicle systems analysis tool, the Advanced Vehicle Simulator (ADVISOR), that models vehicle performance over any predefined driving cycle. Kühlwein [143] focused their research on how the driving characteristics of a light-duty vehicle (LDV) have a direct impact on fuel consumption and CO₂ emissions. Reducing the main parameters of mass, aerodynamic drag, and rolling resistance improves fuel efficiency and reduces total carbon dioxide emissions. Cutting driving resistances can contribute considerably to reaching the common and manufacturer-specific emission targets and mitigating climate change effects. Their study comprehensively investigates all vehicle-based parameters influencing LDV driving behavior.

It should be noted that the toe angle (wheel slip angle) has a negative effect on rolling resistance. With a few degrees of slip, the rolling resistance coefficient can almost double, as said by Thomas [144]. In addition, improper wheel toe angle leads to an accident and rapid tire wear, which is not anticipated. Yurko [38] concentrated on road-tested vehicles; in some cases, the fuel economy benefit may exceed 7%. With more than 105 million registered passenger vehicles in the U.S. that travel almost 1 trillion miles each year, proper wheel alignment could save the country over 5 million barrels of gasoline, or about \$0.3 billion each year.

2.2.7 Tire Wear

Automobile safety and stability depend on several factors, tires being one of them. Tires are a key sub-system of vehicles that are responsible for comfort, fuel consumption, and traffic safety. The tire is a primary component that directly interacts with the road surface. The main job is to guide the wheel on the road and support the associated forces. Despite the pavement's roughness and the vehicle's dynamic motion, it is worth noting that the tire is loaded with a combined force, including longitudinal, lateral, and vertical forces. The tire becomes worn out rapidly due to the combined effects of pitch, roll, or yaw. In the past, researchers [145,146] have focused on tire wear reduction for enhanced tire life and increased vehicle fuel efficiency. Patrick *et al.* [17] demonstrated that tire life is inferred when the tread is worn to a minimum depth or is irregular. They found that about 75.4% of the tires assessed had uneven wear profiles due to incorrect tire-road contact effects. They also confirmed that incorrect tire-road contact is associated with irregularities in the pavement surface.

Li *et al.* [147] developed the theory of tire wear, which provides a theoretical foundation to predict tire wear for different types of tires under different running conditions. The parameter's sensitivity analysis determines the main impact factors for tire tread wear. The results showed that the established model of tire wear was feasible and reflected the wear conditions of tires. In Figure 2.8, it can be seen that the wear quantity increases with an increase in speed. This is because as tire deformation and heat generation capabilities increase, the tread temperature rises and tread wear worsens. Figure 2.9 shows that tire wear has steadily decreased as tire pressure has increased because more substantial tire stiffness prevents tire deformation, energy loss, and heat generation. Research done by Sreeraj *et al.* [148] concentrated on doing finite element analysis on

tire tread wear for its performance under steady rolling situations in order to forecast its behavior and wear under on-road circumstances. Any tire that touches the road at its topmost point has tread. This study examines the impact of tire inflation pressure on tire wear as well as the analysis of the wear on various tire tread rubber types. Lee *et al.* [149] developed a numerical technique for predicting the uneven wear of passenger car tires. It involved applying a single layer of reflective paint to the tread surface by spraying and measuring the intensity of light reflected from a matrix of blocks on the unworn tire. The proposed numerical technique is illustrated through the method of testing the wear on the tread surface of a tire by applying a single layer of reflective paint. Singh and Taheri [42] described how knowledge of tire-road friction conditions is critical for many vehicle control systems. An integrated method based on an intelligent tire-based friction estimator and a brush tire model-based friction estimator is described. The shape of a tire's force-slip characteristics gives information about the slip values at which maximum friction is achieved.

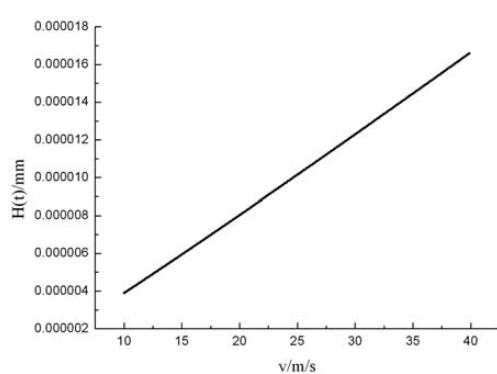


Figure 2.8 The changes of wear quantity with the variation of speed [147].

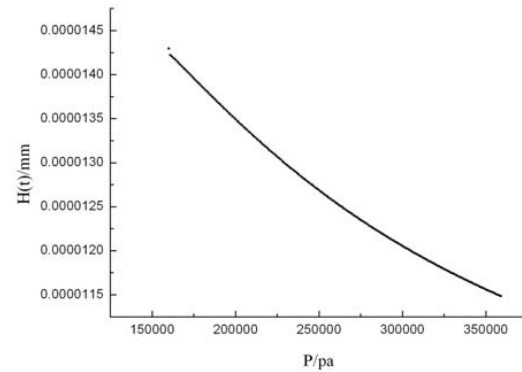


Figure 2.9 The changes of wear quantity with the variation of tire pressure [147].

Kowshik *et al.* [150] addressed tread wear as a familiar tire problem. Several factors affect tires' tread wear, like tire geometry, operating conditions, tire materials, etc. The current work aimed at investigating the effect of three operating condition variables, viz., inflation pressure (30–36 psi), speed (20–60 km/h), and distance (100–300 km), on the response (tread wear). Xi and Ke [151] demonstrated that the length of automobile tire service life, in addition to the design of the tire, determined quality. Under normal conditions, travel truck tires rated at 6–6.5 million meters were acceptable. However, a lot of car tires greatly exceed the quota, reaching 100,000 km

and even 200,000 km. Keeping tire pressures normal, minimizing tire excess load, driving safely, prompt maintenance, tire curing, and suitable tire installation are typically the only issues most people concentrate on.

2.2.8 Effect of Vehicle Load on fuel consumption

Tire inflation pressure and wheel load play a significant role in controlling the traction performance of a tractor. A decrease in tire pressure and an increase in wheel load resulted in higher drawbar pull; however, only the former produced improvements in terms of coefficient of traction, tractive efficiency, and specific fuel consumption [152]. It is known that rolling resistance is directly proportional to tire load and inversely proportional to tire inflation pressure. The rolling loss is increased with overloading; hence, the fuel consumption will be higher. Figure 2.10 shows the fuel consumption rates as a function of vehicle loading when the vehicle speed is 60 km/h. It found that the fuel consumption rate is much higher when vehicles are overloaded. Therefore, vehicle overloading is another important factor directly involved in fuel consumption. Fuel consumption increases even more when the vehicle's wheels are not properly aligned. However, fuel consumption has significantly improved after adjusting the wheel alignment for different loads.

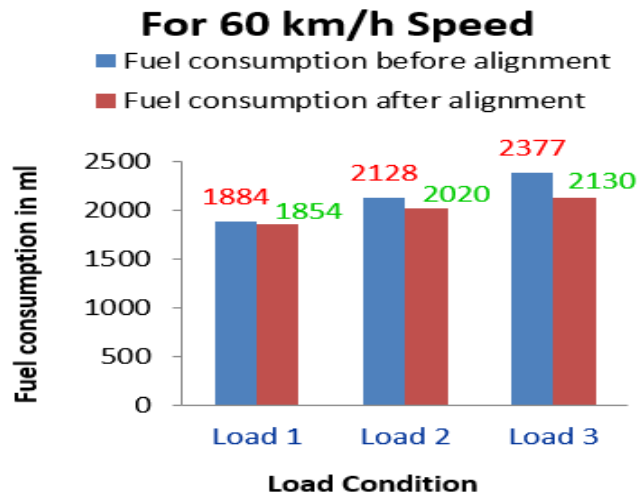


Figure 2.10 Fuel consumption with variation of vehicle load [131].

Raper *et al.* [153] looked at how the new load-inflation pressure tables affected soil deformation and contact stresses using an 18.4 R38 R-1 radial tractor tire with inflation pressures of 41 kPa and 124 kPa and dynamic loads of 13.1 kN and 25.3 kN. Inflation

pressure and dynamic load effects were found on rut width, contact length, and contact area. Increased levels of soil-tire interface stress were found near the center of the tire. Madhusudhana and Nagaraja [40] investigated the effect of tire overload and inflation pressure on rolling loss and resistance on the fuel consumption of cars using vehicle loads as well as a person's weight. Their study examined the relative change in fuel consumption of Skoda Rapid and Ford Classic car tires for different loads and pressures. Gangadhar [154] presented that fuel consumption and tire rolling loss in all types of automobiles have become increasingly important because of the adverse environmental and economic effects. The work finds the effect of rolling resistance and overload on the fuel consumption of automobile tires. Pillai [155] demonstrated the effect of rolling resistance on the fuel consumption of radial passenger and truck tires. An equation for the rolling loss is directly proportional to tire load and inversely proportional to inflation pressure, which has been discussed in their research. They suggested a possible method of optimizing fuel use by adjusting tire load and pressure conditions. The author felt that the same methodology is also applicable for bias tires. Information about friction can be used to enhance the performance of wheel slip control systems. Knowledge of the shape of the force-slip characteristics of a tire provides information about the slip values at which maximum friction is obtained. This research aims to estimate the tire-road friction coefficient using a well-defined tire behavior model. Ejsmont *et al.* [135] conducted studies indicating that tire load and inflation pressure are important factors controlling the rolling resistance of road vehicles. Knowledge of rolling resistance characteristics is important for modeling car dynamics as well as fuel consumption. Establishing proper test conditions for the future standardized on-road method of measuring rolling resistance is also necessary. Results indicate that tires with high rolling resistance are more sensitive to changes in inflation pressure than tires with low rolling resistance.

2.2.9 Effect of Brake Application on Vehicle Performance

Vehicle misalignment may occur as a result of the sudden shock that appears due to the abrupt application of the brakes when the vehicle drives on the road surface. Frequent braking on the road results in faster tire wear and may significantly raise fuel consumption. Ahmed *et al.* [30] have investigated different speeds of vehicles and braking on the road based on the longitudinal tire slip and brake stop distance with

vehicle speed. Wang *et al.* [31] have discussed the effect of driving speed on traffic safety. Sivaramakrishnan *et al.* [33] made a rigid ring-based dynamic tire model to study the effect of high-frequency short-wavelength road disturbances on tire braking performance and the effect of the anti-lock braking system (ABS) on high-frequency tire vibration. Hou *et al.* [34] conducted the two most common V-B (vehicle-bike) conflicts that were discussed according to previous studies. They used a driving simulator to study parameterized motion models and driver braking behaviors described by car speed, collision time, bike speed, and lateral distance. Ryzhikov *et al.* [35] presented the differential braking system of the car. They maintained equal braking on all the wheels and analyzed the results obtained from the mathematical model of a car's braking system. Bella and Silvestri [156] investigated the drivers' braking behavior while approaching zebra crossings under different safety measures (curb extensions, parking restrictions, and advance yield markings) and without treatment. The speed reduction time is used to describe the driver's behavior. This result showed that the driver could receive clear information and better adapt to his approaching speed because of the improved visibility of the pedestrian allowed by the curb extensions. Asadi and Vahidi [41] proposed using upcoming traffic signal information within the vehicle's adaptive cruise control system to reduce idle time at stop lights and fuel consumption. The control objectives are timely arrival at the green light with minimal use of braking, maintaining a safe distance between vehicles, and cruising at or near a set speed. Three example simulation case studies were presented to demonstrate the potential impact on fuel economy, emission levels, and trip time. Caban [157] emphasized the viability of eco-driving in city traffic and compared its impacts with conventional driving. The effects of a few characteristics are reported, including average speed, driving time, distance, and kind of road, stops, and amount of braking. According to the investigation, eco-driving tactics provide some advantages in city traffic. Sterna *et al.* [158] presented traffic waves as phenomena that emerge when vehicular density exceeds a critical threshold. It showed that intelligent control of an autonomous vehicle can dampen stop-and-go waves that can arise even without geometric or lane-changing triggers. These experimental findings suggest a paradigm shift in traffic management. Flow control will be possible via a few mobile actuators (less than 5%), long before the majority of vehicles have autonomous capabilities. Niu and Sun [159] demonstrated the new in-vehicle systems that provide the driver with speed advice through variable message signs and an in-car display. It has been found

that the number of stops is significantly reduced for vehicles with the GWSGS and EDSGS. Fuel consumption and CO₂ emissions can be reduced by 25% and 13%, compared with vehicles without speed guidance. Figure 2.11 shows the fuel consumption rates as a function of the number of braking actions on the road. It is found that the fuel consumption rate is much higher when more brakes are applied, and it becomes worse in rough road conditions.

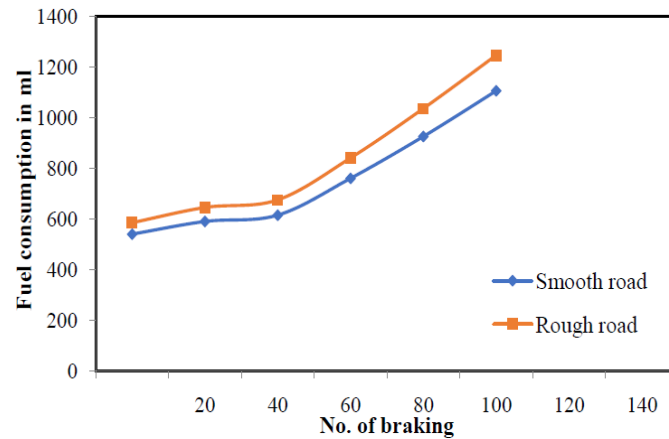


Figure 2.11 Fuel consumption at number of braking [160].

2.2.10 Effect of Vehicle Characteristics on Fuel Efficiency

The components of fuel consumption due to engine operation, rolling and air drag, inertia, etc., and the effect of vehicle characteristics on these components are estimated using the instantaneous model. A different approach, which has been used by Biggs Akcelik [161], is to relate fuel consumption to the individual components that contribute to fuel consumption, e.g., drag, inertial, and grade forces. These components are then related to vehicle characteristics. Al-Momani and Badran [162] have presented that fuel consumption depends on the altitude at which the vehicles are driven, and that can be improved by reducing load, aerodynamic drag, and rolling resistance (tire improvements), as well as by improving maintenance conditions and using efficient fuel types. A well-designed diffuser (clearance) underneath the vehicles could decrease the drag due to the optimized airflow distribution around the vehicle body. Svenson and Fjeld [163] studied the effects of gradient, curvature, and road surface roughness and discovered their correlation to functional road class. Davari [164] presented that the resistive forces for a vehicle to overcome are aerodynamics, inertia, internal friction, and rolling resistance forces. Improvement in tire rolling efficiency is one of the key enablers for lower energy consumption. Their study aimed to develop a simulation

environment to simulate rolling losses (the energy dissipated from the tire) and investigate the potential of controlling different chassis parameters to reduce rolling losses. Over-actuation can be used to reduce tire energy losses during driving. Utilizing the CSC function for a chosen vehicle in a maneuver shows a significant improvement of about 60% in rolling loss reduction while maintaining path tracking. It has been shown that by using this function, the tire forces can be distributed more evenly while maintaining the global force, which is especially beneficial when driving at the limit.

A secure energy resource is one of the prime factors driving the growth of any country, and it becomes more critical for developing countries. In India and other Asian developing countries, fossil fuels will continue to dominate the energy supply. These countries are facing the challenges of preventing environmental degradation as a result of hydrocarbon combustion and the long-term challenges of energy availability and sustainability [165]. According to studies by Nasir *et al.* [166], the transportation industry's greenhouse gas emissions are a significant global source of concern. Researchers are working hard to replace fossil fuels with alternative fuels and are experimenting with different driving techniques to help traffic flow. Utilizing intelligent transportation system (ITS) technologies can save fuel usage and pollutant emissions. In their research, Barrand and Bokar [167] established a simple law to predict fuel savings between two tire configurations. This approach has been evaluated in practice for passenger car tires fitted on one gasoline and one diesel midsize car. The results also show how low-rolling-resistance tires can help decrease CO₂ emissions and local pollutants such as CO emissions for gasoline engines and NO_x emissions for diesel engines. Barth and Boriboonsomsin [168] presented their research on how fuel consumption and CO₂ emissions can be lowered through a variety of strategies that reduce congestion, smooth traffic flow, and reduce excessive vehicle speeds. Eco-driving typically consists of changing a person's driving behavior by providing general static advice to the driver (e.g., do not accelerate too quickly, reduce speeds, etc.). The idea of "dynamic eco-driving," in which drivers receive real-time directions based on shifting traffic circumstances nearby, is examined in this study. By giving drivers timely information, it is possible to minimize fuel usage by 10–20% without noticeably lengthening journey times.

Borcherts *et al.* [169] conducted research that shows that fuel economy can be improved by reducing weight, decreasing aerodynamic drag (better streamlining), decreasing rolling resistance (tire improvements), and increasing engine and power train efficiency. As applied to a six-passenger, 3700 lb. present-day vehicle powered with a

250-CI six-cylinder engine through an automatic transmission, the improvements in EPA M/H fuel economy for 1% reductions in these parameters are projected to be 0.75%, 0.35%, and 0.28%, respectively. According to the National Research Council [170], Americans spend approximately \$20 billion annually on tire replacement for cars and light trucks. About 200 million replacement passenger tires are purchased annually in the United States. Each time they replace their tires, motorists spend several hundred dollars and must choose among tires varying in price, style, and many aspects of performance. The maintenance of proper inflation pressure is especially important. Hunt *et al.* [171] developed a mathematical model to predict the energy consumption of a heavy vehicle. Novel, low-cost testing methods have been developed to determine engine and drivetrain characteristics. The model can predict fuel consumption for a 37 ton tractor-trailer vehicle over a 4 km drive cycle to within 1%. Bechtold and Regency [172] presented a broad overview of passenger car fuel economy. The main components of vehicle power requirements (aerodynamics, rolling resistance, and inertia) are discussed for steady-state and driving cycle conditions. Estimates of the fuel economy of a vehicle with regenerative braking are made, and fuel effects (alcohols vs. hydrocarbons), engine types, and differences due to combustion systems are discussed. Internal combustion engines (ICEs) currently power 99.8% of global transportation, and liquid petroleum fuels account for 95% of transport energy, according to Leach *et al.* [173]. Engine and conventional powertrain developments alone could reduce fuel consumption in light-duty vehicles (LDVs) by more than 30%. Hybridization and lightweighting could reduce fuel consumption by 50% compared to the current average for LDVs. Current after-treatment technology can ensure that exhaust pollutant levels meet the most stringent current emissions requirements. According to Sina *et al.* [174], research focused on the importance of energy efficiency has grown rapidly in recent years, considering the global energy crisis and climate change. Their study aimed to introduce a method to increase powertrain efficiency based on minimizing tire power loss. The effectiveness of the proposed online control system, called the tire inflation pressure regulation system, is simulated in some driving cycles. It is shown that up to 2% improvement is achievable.

Chapter 3

Mechanics of Wheel Alignment

3.1 WHEEL CHARACTERISTIC ANGLES

The term "wheel alignment" refers to the reference of three basic measurements: caster angle, camber angle, and toe angle. In addition to that, thrust angle and steering axis inclination angle are also related to wheel alignment. There are some standard values for these angles that a technician can utilize as objectives for adjustment [175–178].

3.1.1 Caster

Caster is the tilting of the topmost point of the steering axis forward or backward (when viewed from the side of the vehicle), as shown in Figure 3.1. A backward tilt is positive (+), whereas a forward tilt is negative (-). Casters impact steering directional control but do not affect tire wear and are not changeable on this vehicle. Keeping the body at its designated height is crucial because vehicle height has an impact on casters. Casters will be affected if the vehicle is overloaded or the rear spring is weak or drooping. When the vehicle's rear is lower than its prescribed trim height, the front suspension shifts to a more positive caster. Suppose the vehicle's rear is higher than its specified trim; the front suspension shifts to a less positive caster. While there is insufficient positive caster, steering becomes sensitive at high speeds, and wheel returnability suffers while exiting a corner. If one wheel has a more positive caster than the other, the wheel with a greater positive caster will draw toward the center of the vehicle. Because of this, the vehicle will pull or lead to the side with the least positive caster [179]. The caster angle value for typical road cars usually ranges from $+3^{\circ}$ to $+5^{\circ}$, and it is often more prominent on rear-wheel-drive vehicles.

3.1.2 Camber

Camber is the angle at which the wheels deviate from the vertical as viewed from the front of the vehicle, as illustrated in Figure 3.2. Camber is positive (+) when the wheels tilt outward at the top. Camber is negative (-) when the wheel tilts inward at the top. The tilt is measured in degrees relative to the vertical. Camber adjustments affect directional control as well as tire wear. Too much positive camber may cause early tire wear as well as excessive wear on the suspension components. Too much negative camber may cause early tire wear as well as excessive wear on the suspension components. It is worth noting that a tire at a relatively small negative camber angle

generates the most cornering force, generally around negative 1/2 degree. This is due to the camber thrust, which creates a lateral force in addition to the elastic deformation of the tread rubber at the place where it hits the road (the contact patch). When there is an unequal side-to-side camber of 1° or more, the vehicle will pull or lead to the side with the highest positive camber [177].



Figure 3.1 Caster angle [176].



Figure 3.2 Camber angle [177].

3.1.3. Toe

Toe is a measurement of how much the front and/or rear wheels are turning in or out from a forward position, as shown in Figure 3.3. When the wheels turn in, the toe is positive (+). When the wheels are turned out, the toe is negative (-). When the wheels are turned, the purpose of the toe is to ensure that the wheels roll parallel. Toe is a measurement of how much the front and/or rear wheels are rotated in or out from a straight flection of the wheel support system that occurs while the vehicle is rolling forward. In other words, while the vehicle is stationary and the wheels are set up with toe-in, the wheels prefer to roll parallel on the road when the vehicle is going. Improper toe adjustment can result in early tire wear and steering instability [178].



Figure 3.3 Toe-in and toe-out [178].

The toe-in or toe-out can be determined by comparing the distances between the front and rear ends of the tires, as indicated in Figure 3.4. After measuring the distances A and B, the toe-in and toe-out can be identified as follows: if $A > B$: toe-out; if $A < B$: toe-in; if $A = B$: zero toe. The following equation can then be used to calculate the toe-in or toe-out angle for each wheel:

For toe-out, $\delta = 2 \sin^{-1} \left(\frac{A-B}{2d} \right)$ (when $A > B$)

For toe -in, $\delta = 2 \sin^{-1} \left(\frac{B-A}{2d} \right)$ (when $B > A$)

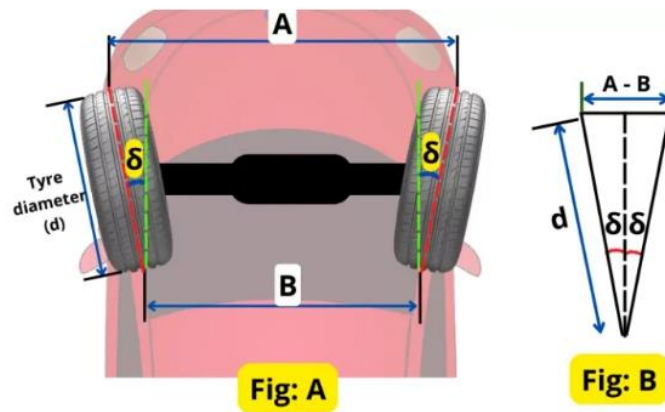


Figure 3.4 Toe in and toe out measurement [180].

3.1.4 Thrust Angle

The thrust angle, as shown in Figure 3.5, is the angle formed between the thrust line and the centerline. The angle is considered positive when the thrust line is to the right of the centerline. The angle is negative if the thrust line is to the left of the center. It results from the rear wheels or axles being out of alignment, which causes the steering to pull or lead to one side or the other. It is the most common cause of a crooked or off-center steering wheel. The rear axle or toe alignment must be corrected to remove the thrust angle. If this is not practicable, utilizing the thrust angle as a reference line for aligning the front toe can help restore center steering [181].

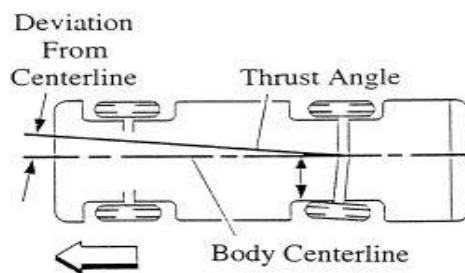


Figure 3.5 Thrust angle [181].

3.1.5 Steering Axis Inclination

The angle between the higher and lower steering pivots with respect to the vertical is known as the steering axis inclination (SAI), as depicted in Figure 3.6. The line goes through an SLA suspension's upper and lower ball joints. The line on a MacPherson strut suspension that passes between the lower ball joint and the top strut mount or bearing plate can deviate from the vertical line. From the front, SAI is also the inward tilt of the steering axis. It, like the caster, offers directional stability. However, it also decreases steering effort by shortening the scrub radius. SAI is a non-adjustable angle that is used in conjunction with camber and the included angle to identify bent spindles, struts, and misallocated crosses [182].

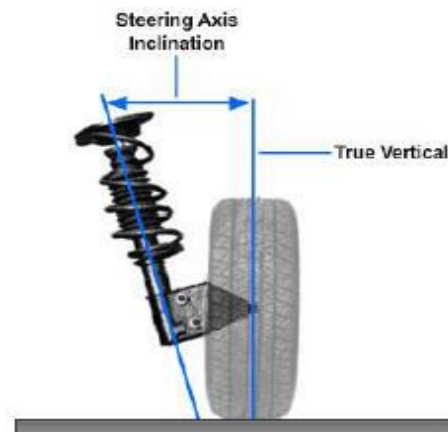


Figure 3.6 Steering axis inclination (SAI) [182].

Currently, various wheel alignment adjustment techniques have been established for the alignment of wheels. The alignment adjustment procedure comprises manual wheel alignment checks with all-wheel alignment-related parts. The car is then aligned with computerized wheel alignment equipment.

3.2 WHEEL ALIGNMENT

Wheel alignment is an important component of any automotive maintenance program because it adjusts the characteristic angles of the wheels to the vehicle manufacturer's guidelines. These modifications are made to prevent tire wear and to make sure that the vehicle travels straight and true without pulling to one side. The alignment angles can be acquired in accordance with the vehicle manufacturer's standards, which address specialized handling and suitable adjustment of characteristic angles for standard application. Nowadays, advanced driver assistance systems (ADAS) are employed in

new automobiles, which include electronic stability control, anti-lock brakes, lane departure warning, adaptive cruise control, traction control, and related features [183, 184]. Mechanical alignment modifications can currently damage the systems, and many automobile manufacturers have implemented electronic resets for this system following a mechanical alignment.

The suspension design determines wheel alignment, and not all vehicles are completely or easily changeable. To make up for vehicle damage or an alignment change brought on by the use of other aftermarket performance kits, some vehicles require aftermarket adjustment kits. Front-end alignment, thrust angle alignment, and four-wheel alignment are the three forms of alignment that have been explained and detailed in the following sections.

3.2.1 Front-end Alignment

Front-end alignments, as the name implies, solely measure and change the angles of the front axles. It is the most fundamental and basic alignment approach. Front-end alignments are beneficial for some automobiles with a solid rear axle. Even if the front axle angles are aligned, ensuring the front tires are properly positioned in front of the rear tires is critical. Front-end alignment is insufficient on most current vehicles and is regarded as archaic owing to its limited nature [185].

3.2.2 Thrust Angle Alignment

This alignment goes a step further by doing a thrust angle alignment in addition to the front-end alignment described above. A thrust angle alignment is conducted to ensure that all four tires are in a rectangular shape with each other. A thrust alignment is required for all cars with a solid rear suspension to get better outcomes on the road in terms of tire wear and safety. Thrust angle alignments help identify a vehicle's "dog track" along the road with the rear offset from the front. The thrust angle is defined by an imaginary line drawn perpendicular to the centerline of the rear axles, as shown in Figure 3.7. It determines whether the rear axle is aligned with the vehicle's centerline or not. It also determines if the rear axle is parallel to the front axle and ensures that the wheelbase is the same on both sides of the vehicle. A misaligned axle or improper toe settings might result in incorrect thrust angles. Incorrect thrust angles cause the vehicle to respond differently while turning left vs. turning right [185].

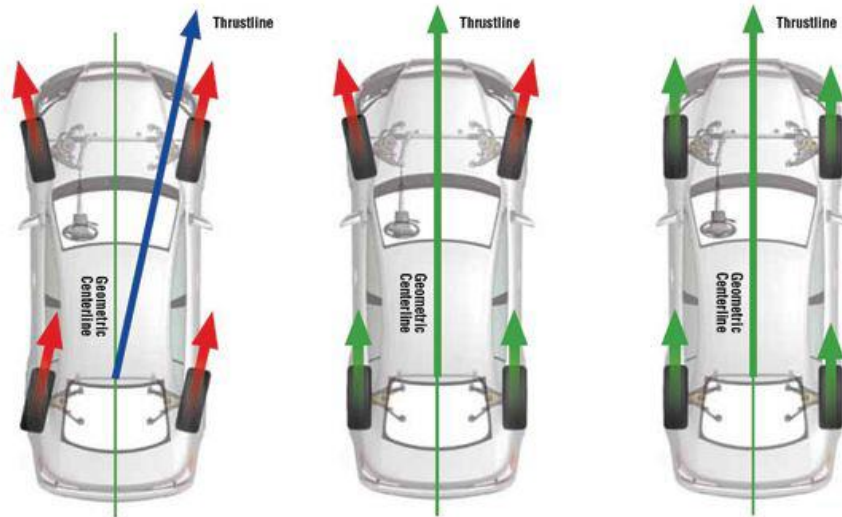


Figure 3.7 Thrust angle alignments [186].

If the thrust angle on a vehicle with a solid rear axle is improper, it may be essential to straighten the frame and place the rear axle appropriately.

3.2.3 Four-wheel Alignment

A four-wheel alignment is advised for cars with four-wheel independent suspensions or front-wheel drive vehicles with adjustable rear suspensions. Because the technique measures and changes rear axle angles with front axle angles, it causes the need for an extra workforce. Today, four-wheel alignment is the most prevalent type of alignment [185]. This alignment goes a step further by measuring and modifying the rear axle angles, combining front-end and thrust angle alignments. This alignment approach returns all four corners to the manufacturer's requirements [186]. The most extensive alignment necessitates the use of specialized equipment and experienced labor.

3.3 WHEEL ALIGNMENT SYSTEM

There are two types of wheel alignment systems: mechanical wheel alignment systems and computerized wheel alignment systems. These two types are discussed in Sections 3.3.1 and 3.3.2, respectively.

3.3.1 Mechanical Wheel Alignment System

Mechanical wheel alignment can be accomplished in various ways, including using strings, angle gauges, aluminum bars, turn plates, and other tools. Because of its simplicity and repeatability, racing teams primarily employ this strategy. The

mechanical measuring approach is useful since contemporary computerized methods are insufficient for aligning race vehicles. Toe measurements can be taken with a straight aluminum rod in addition to wheel alignment measures. With a straight aluminum bar and appropriate alignment expertise, one may alter the toe with results that outperform wheel-clamp-based aligners. The use of aluminum bars has the advantage of allowing for more considerable variations over longer distances [187].

Toe measurements by aluminum bars are now often substituted by strings, popularly known as the ‘String Method’. With the right equipment, the string method provides accurate toe measurements and may be performed on all four wheels simultaneously. It also ensures that all the wheels are square, so that the rear axle runs true and parallel to the front axle. As shown in Figure 3.8, the strings are linked to the car and move with it; hence, the vehicle may be rolled to settle the settings after the modifications are performed without altering the measurements [188].

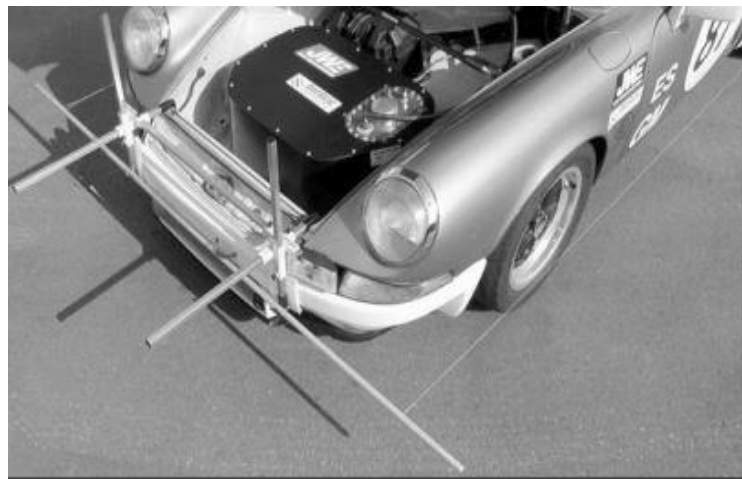


Figure 3.8 Toe alignment measuring strings [188].

3.3.2 Computerized Wheel Alignment System

Lasers or infrared sensors are used for measurements in computerized alignment systems. Camber, caster, toe, and thrust angle information are presented on a computer screen. The equipment has a built-in calibration mechanism that the operator can calibrate. The functioning of a computerized wheel alignment machine may be separated into two primary principles: internally and externally referenced measuring methods [187]. Internally referenced measurement systems have all their components placed on the vehicle, such as the sensor heads employed in this study, as shown in Figure 3.9 (a).

Externally referenced measurement systems have the laser targets mounted on the wheel while the laser sources are mounted away from the car on a separate stand, as shown in Figure 3.9 (b).



(a) Internally referenced



(b) Externally referenced

Figure 3.9 Computerized wheel alignment measurement systems of (a) Internally reference and (b) Externally reference [189].

The main difference between the two systems is that the readings of the sensors in externally referenced measuring systems vary as the vehicle travels. As a result, the system must continuously determine the car's location and take it into account when calculating accurate toe and camber values. In the case of internally referenced systems, the toe readings remain relatively constant even if the vehicle is pushed on the turn plates, provided the steering angle remains constant. The internally referenced systems evaluate at least six different measurement lengths. These systems begin by measuring the location of the wheel centers in relation to one another by utilizing the six different concurrently measured angles illustrated in Figure 3.10 [187].

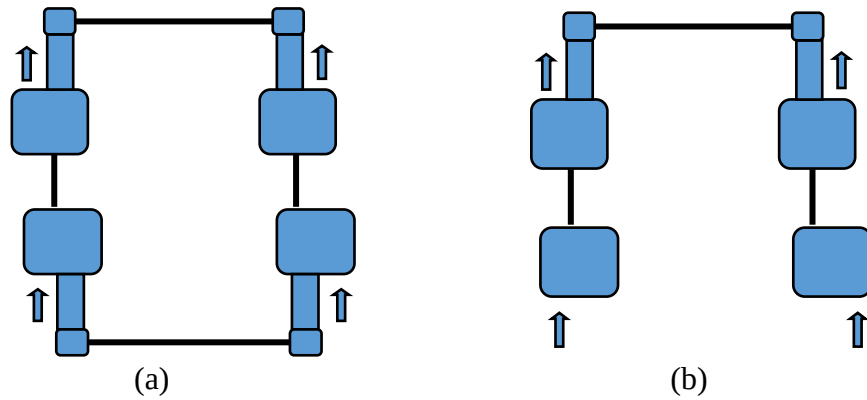


Figure 3.10 (a) 8-Sensor system and (b) 6-Sensor system [187].

After determining each wheel center's position, the wheels' angles with respect to the wheel center are calculated. As illustrated in Figure 3.10, a one-liner delivers two laser beams in opposite but not coinciding directions, and a minimum of six measurement lines are required to measure four separate toe values: four transversal and two lateral. Most alignment machines now employ a full 8-sensor measuring system, as seen in Figure 3.10(a). When the front transversal toe sensors no longer interact with one another, this permits the coarse toe measurement necessary for larger steering angles. Caster sweeps of 20 can be performed on turn tables equipped with an 8-sensor system without the need for angle sensors [187].

Internally referenced measuring methods only know about angles rather than real lengths. They estimate distances by employing observed angles and reference distances from sensor head data. Externally referenced measuring methods, on the other hand, know the vehicle's dimensions based on the measurements made. Externally referenced systems include a separate set of cameras that are not linked to the vehicle and easily identifiable pattern-printed targets fastened to the wheels. The measuring method is based on the notion that a specific form, such as a circle, will appear larger or smaller depending on the distance from the camera lens, and the same circle will appear oval from a different angle. The software analyzes the deformations using the image from the camera lens and reports the results. The quality of the output is also dependent on how the clamps with the printed pattern are mounted.

3.4 BRIEF DISCUSSION ON WHEEL ALIGNMENT MECHANICS

The behavior of tires is strongly influenced by applied forces, moments, and angles. These forces can be considered road forces and moments acting on the tire, as shown in Figure 3.11. The camber and slip angles are the first two important angles to consider. The slip angle is the difference in direction and heading of the wheel, whereas the

camber angle is the inclination angle from its vertical position. The lateral force is associated with these two angles. The lateral force in the Y direction, the normal force in the Z direction, and the longitudinal force in the X direction are all types of forces. The tire's force on the road produces longitudinal force (F_x), which decreases when braking. During cornering, the forces produced by a non-zero camber angle and a non-zero slip angle combine to form the lateral force (F_y). The opposite of the upward vertical force is known as the normal force (F_z). Overturning, rolling resistance, wheel torque, and alignment moments are among the moments. The vertical load shifts to the side during cornering, which results in an overturning moment (M_x). Rolling resistance (M_y), which typically results from a number of causes, is what causes an energy loss. When the slip angle is not zero, the aligning moment (M_z), also known as the self-aligning torque, produces a restoring moment on the tire to realign the direction of travel with the direction of heading. It should be mentioned that the axle on the wheel also produces a moment. Finally, but not least, it should be noticed that the wheel torque (T_{in}) points in the opposite direction of Y when the camber angle is zero [190].

The slip angle (α) generates a component of lateral force if the direction of travel separates from the wheel heading (if the wheel's angular displacement separates from the path the tire is travelling) (F_y). This lateral force will exert itself at a point behind the wheel's center in a way that aims to realign the tire. It should be noticed that the slip angle and the steering angle are not the same [190].

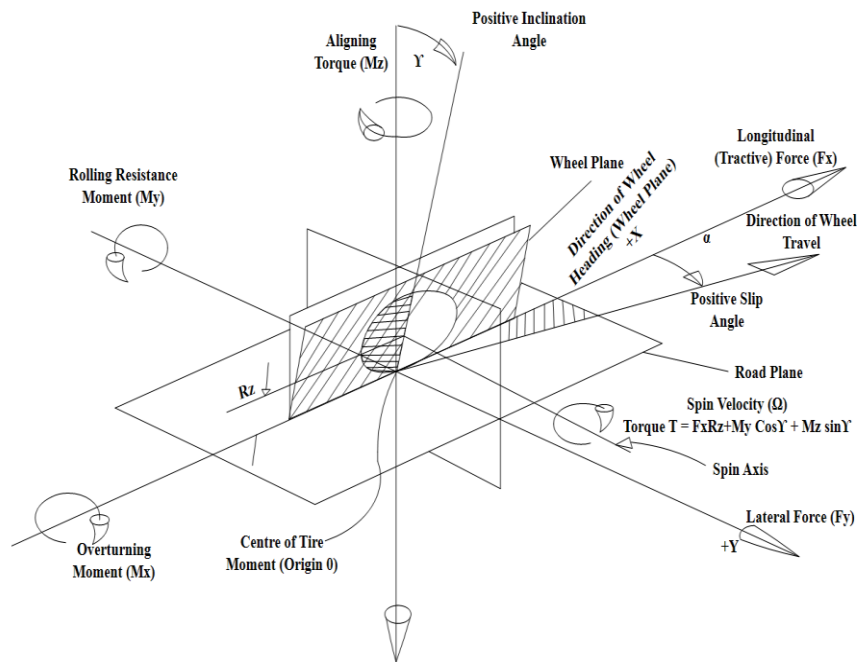


Figure 3.11 Forces acting on an automobile wheel [190].

3.5 ROLLING RESISTANCE

3.5.1 Determination of Rolling Resistance

The forces opposing the motion of a vehicle, known as the driving resistance forces, are often recognized as the resistance to the vehicle's motion. The resistance force created by the deformation of the tire in contact with the road surface is characterized as rolling resistance. The rolling resistance (F_R) can be determined using a simple equation as follows:

$$F_R = \mu_{RRC} \times V_W \quad (3.1)$$

where, μ_{RRC} is the rolling resistance coefficient and V_W is the vehicle weight.

Several factors affecting rolling resistance make it virtually very difficult to develop a formula that considers all variables. Over the years, several equations have been developed to estimate rolling resistance. This study used the rolling resistance coefficient (μ_{RRC}) in relation to slip/toe angle graphically presented by Clark [191]. Using the graph (Figure 3.12), a general polynomial equation of 2nd order ($y = ax^2 + bx + c$) is considered, where x is the toe angle (α) and y is the rolling resistance coefficient (μ_{RRC}). Then the equation can be written as

$$\mu_{RRC} = a\alpha^2 + b\alpha + c \quad (3.2)$$

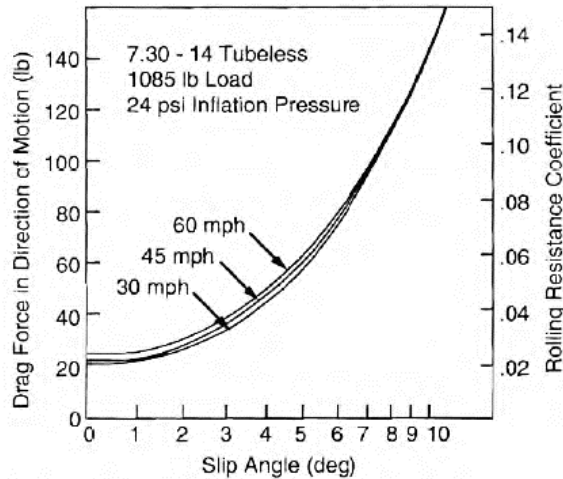


Figure 3.12 Rolling resistance coefficient versus slip angle [144].

Tire deformation while rolling is the main cause of rolling resistance. Losses arise because the energy required to deform the section of the wheel in contact with the surface is greater than the energy recovered when that component returns to its former shape. The energy that is lost is dispersed as heat. Additionally, factors like tire type,

surface deformation, normal force, and wheel alignment affect rolling resistance. Because of this intricacy, the rolling resistance and its coefficient are generally measured empirically by Ron and Michelin [192,193]. Figure 3.13 is from Tire Digest, a Michelin website, and depicts the components and their contribution to rolling resistance in the tires of an automobile.

What	Surface of tire and air	Tire tread			Sidewall and bottom part	
How	Air circulation	Slippage on ground	Deformation hence dissipation of energy			
			bending	compression	shearing	bending shearing
						
Contribution	< 15%		60 to 70%			20 to 30 %

Figure 3.13 Constituents of rolling resistance [193].

The coefficient of rolling friction is related to normal pressure and wheel width and is inversely proportional to radius. It also increases when the surface of contact rises [194].

Further, the correlation between rolling resistance force and fuel consumption is developed using the Lagrange Interpolation (Unequal Interval) formula as below (Hoffman, 1992) [195]:

$$\begin{aligned}
 Y = f(x) = & \frac{(x-x_1)(x-x_2)(x-x_3)(x-x_4)}{(x_0-x_1)(x_0-x_2)(x_0-x_3)(x_0-x_4)} Y_0 + \frac{(x-x_0)(x-x_2)(x-x_3)(x-x_4)}{(x_1-x_0)(x_1-x_2)(x_1-x_3)(x_1-x_4)} Y_1 + \\
 & \frac{(x-x_0)(x-x_1)(x-x_3)(x-x_4)}{(x_2-x_0)(x_2-x_1)(x_2-x_3)(x_2-x_4)} Y_2 + \frac{(x-x_0)(x-x_1)(x-x_2)(x-x_4)}{(x_3-x_0)(x_3-x_1)(x_3-x_2)(x_3-x_4)} Y_3 + \\
 & \frac{(x-x_0)(x-x_1)(x-x_2)(x-x_3)}{(x_4-x_0)(x_4-x_1)(x_4-x_2)(x_4-x_3)} Y_4
 \end{aligned} \quad (3.3)$$

3.6 DETERMINATION OF ENERGY CONSUMPTION

Energy consumption is the energy loss resulting from a wheel's rolling resistance when in contact with the road surface. The energy consumption (E_c) can be determined using a simple equation as follows:

$$E_c = (F_R \times T_d) \quad (3.4)$$

where, F_R = Rolling resistance force (N) and T_d = Travelling distance (km).

A rubber tire consists of various polymers which are formed in a tangled mass. Rubber polymers are tangled around each other and their neighbors, and when the material is stressed, the polymers drag their chains across each other. Friction is created in this way, which is then dissipated as heat. The energy used to deform the tire at the start of the contact patch is greater than the energy recovered at the end of the contact patch. Figure 3.14 shows the energy loss produced by the generation of heat in the rubber as the tire deforms.

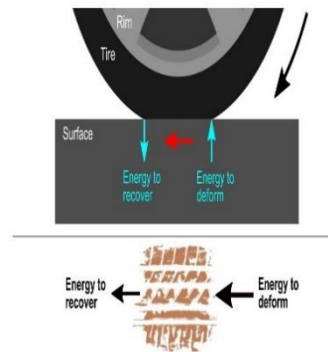


Figure 3.14 Constituents of energy consumption [196].

The energy efficiency of transportation is the product of the usable traveled distance of people, goods, or any other type of load and the total energy put into the transport propulsion means. Depending on the propulsion, the energy input can take numerous forms, most commonly being liquid fuels, electrical energy, or dietary energy [197, 198]. Energy intensity is another term for energy efficiency [199]. Energy consumption in transportation is the inverse of energy efficiency in transportation.

Energy efficiency in transportation is frequently characterized in terms of fuel consumption, which is the reciprocal of fuel economy [198]. However, fuel consumption is associated with a mode of propulsion that employs liquid fuels, but energy efficiency applies to any mode of propulsion. To minimize misunderstanding and compare any vehicle's energy efficiency, experts often measure energy in the International System of Units, i.e., joules. As a result, under the International System of Units, energy efficiency in transportation is measured in meters per joule (m/J), but energy consumption in transportation is measured in joules per meter (J/m). The more efficient the vehicle, the more meters it can cover with one joule (higher efficiency) or the fewer joules it requires to travel one meter (lower consumption).

3.7 GOVERNING EQUATION

The following equations are used to calculate the angle rate, rolling resistance, fuel consumption and energy consumption.

$$\text{Angle rate } (A_r) = \frac{\alpha_t \times V_s}{T_d} \text{ (deg. /sec) [200]} \quad (3.5)$$

where, α_t = toe angle (deg.), V_s = Vehicle speed (km/h), T_d = Travelling distance (km)

$$\text{Rolling resistance force } (F_R) = \mu_{RRC} \times V_W \text{ (N)} \quad (3.6)$$

where, μ_{RRC} = Rolling resistance coefficient and V_W = Vehicle weight (N)

$$\text{Energy consumption } (E_c) = (F_R \times T_d), \text{ kJ} \quad (3.7)$$

where, F_R = Rolling resistance force (N) and T_d = Travelling distance (km)

$$\text{Engine RPM rate} = \frac{RPM - RPM_{(min)}}{RPM_{(min)}} \times 100 \quad (3.8)$$

$$\text{Rolling resistance coefficient rate} = \frac{\mu_{RRC} - \mu_{RRC_{(min)}}}{\mu_{RRC_{(min)}}} \times 100 \quad (3.9)$$

$$\text{Rolling resistance force rate} = \frac{F_R - F_{R_{(min)}}}{F_{R_{(min)}}} \times 100 \quad (3.10)$$

$$\text{Fuel consumption Rate} = \frac{F - F_{min}}{F_{min}} \times 100 \quad (3.11)$$

$$\text{Energy consumption rate} = \frac{E_c - E_{c_{(min)}}}{E_{c_{(min)}}} \times 100 \quad (3.12)$$

$$\text{KPL reduction rate} = \frac{KPL_{(max)} - KPL}{KPL_{(max)}} \times 100 \quad (3.13)$$

3.7.1 Correlation Formula

3.7.1.1 Pearson's correlation coefficient

The Pearson correlation coefficient (r) is the most often used method for finding a linear correlation. The magnitude and direction of the relationship between two variables are stated as numbers ranging from -1 to 1. The Pearson's correlation coefficient [201] can be defined as

$$r = \frac{\sum (x_i - \bar{x})(y_i - \bar{y})}{\sqrt{\sum (x_i - \bar{x})^2 \sum (y_i - \bar{y})^2}} \quad (3.14)$$

Where, r = correlation coefficient, x_i = values of the x-variable in a sample, $\bar{x}$ = mean of the values of the x-variable, y_i = values of the y-variable in a sample, $\bar{y}$ = mean of the values of the y-variable

The Pearson's correlation coefficient for two variables can be written as

Between toe angle and fuel consumption

$$r_{\alpha f_c} = \frac{\sum_{i=1}^n (\alpha_i - \bar{\alpha})(f_c - \bar{f}_c)}{\sqrt{\sum_{i=1}^n (\alpha_i - \bar{\alpha})^2 \sum_{i=1}^n (f_c - \bar{f}_c)^2}} \quad (3.15)$$

Between tire pressure and fuel consumption

$$r_{t_p f_c} = \frac{\sum_{i=1}^n (t_p - \bar{t}_p)(f_c - \bar{f}_c)}{\sqrt{\sum_{i=1}^n (t_p - \bar{t}_p)^2 \sum_{i=1}^n (f_c - \bar{f}_c)^2}} \quad (3.16)$$

Between vehicle load and fuel consumption

$$r_{v_l f_c} = \frac{\sum_{i=1}^n (v_l - \bar{v}_l)(f_c - \bar{f}_c)}{\sqrt{\sum_{i=1}^n (v_l - \bar{v}_l)^2 \sum_{i=1}^n (f_c - \bar{f}_c)^2}} \quad (3.17)$$

Between brake application and fuel consumption

$$r_{b_a f_c} = \frac{\sum_{i=1}^n (b_a - \bar{b}_a)(f_c - \bar{f}_c)}{\sqrt{\sum_{i=1}^n (b_a - \bar{b}_a)^2 \sum_{i=1}^n (f_c - \bar{f}_c)^2}} \quad (3.18)$$

Between toe angle and tire wear

$$r_{\alpha t_{cgw}} = \frac{\sum_{i=1}^n (\alpha_i - \bar{\alpha})(t_{cgw} - \bar{t}_{cgw})}{\sqrt{\sum_{i=1}^n (\alpha_i - \bar{\alpha})^2 \sum_{i=1}^n (t_{cgw} - \bar{t}_{cgw})^2}} \quad (3.19)$$

3.7.1.2 Regression Analysis and Correlation Equation

Regression analysis is one of the most powerful statistical methods for developing the correlation between two or more interests. The method most frequently employed by sociologists is that of linear equations. This multiple linear regression can be written in the form [202]:

$$Y = a + b_1 X_1 + b_2 X_2 + b_3 X_3 + b_k X_k, \quad (3.20)$$

where Y is the dependent variable, which is to be determined; $X_1, X_2, X_3, \dots, X_k$ are the known independent variables on which the predictions are to be made and $a, b_1, b_2, b_3, \dots, b_k$ are the coefficients that are determined through the least squares method.

Chapter 4

Experimental Setup and Methodology

4.1 INTRODUCTION

Several techniques had already been used for the alignment of wheels, which were discussed in the literature review chapter. This thesis used a simple, easy, and effective computerized wheel alignment machine (the Best 5800 Model). The construction of a wheel alignment pit, detailed specifications of the wheel alignment machine, useful hand tools, and experimental factors are discussed in this chapter. The experimental procedure to determine the wheel misalignment of light vehicles is also discussed in this chapter.

4.2 DESIGN AND CONSTRUCTION OF WHEEL ALIGNMENT PIT FOR LIGHT VEHICLE

The detailed design layout of the wheel alignment pit (length = 11.0 ft. 10 inches, width = 2 ft. 11 inches, depth = 5 ft.) for a light vehicle is given in Figure 4.1. The wheel alignment pit is constructed in accordance with the design layout, as shown in Figure 4.2.

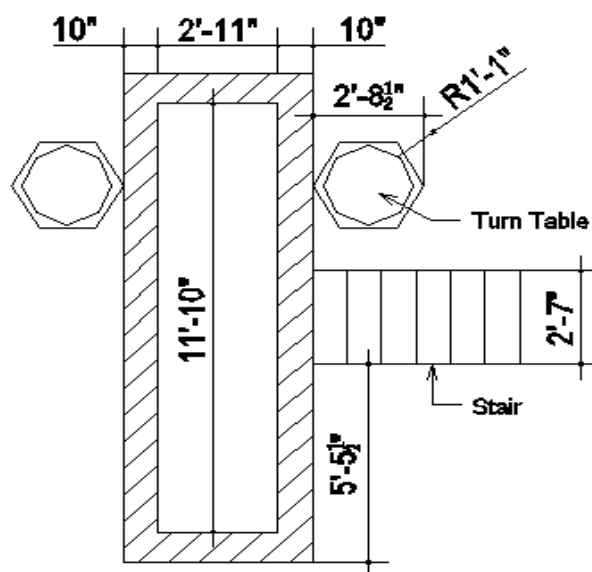


Figure 4.1 Wheel alignment PIT design for light vehicles.

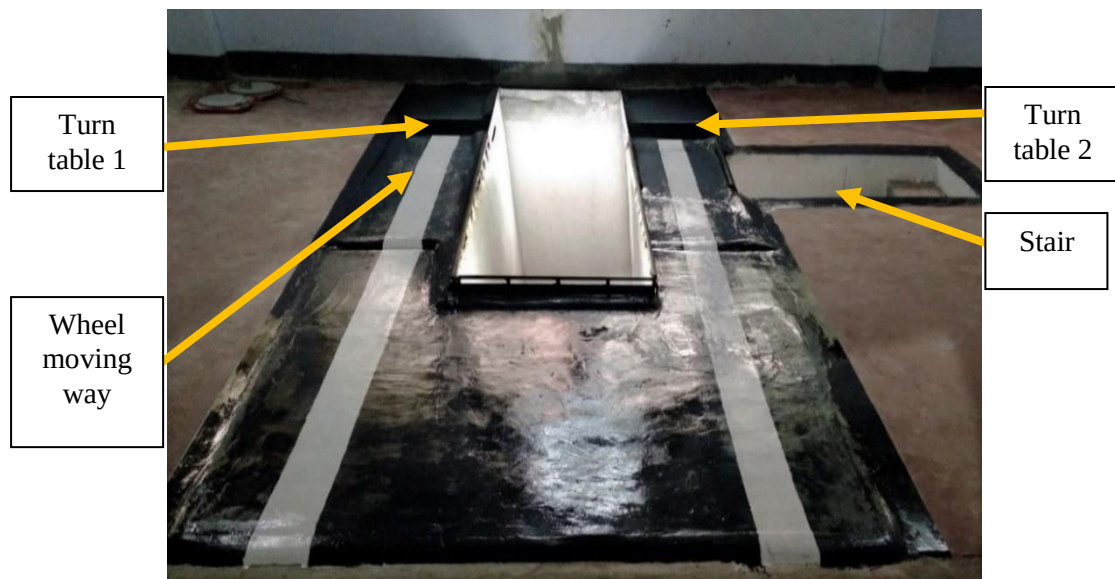


Figure 4.2 Constructed a PIT for light vehicle alignment.

4.3 DESCRIPTION OF WHEEL ALIGNMENT MACHINE

A computerized wheel alignment machine (the Best 5800 model, manufactured by E-Modern Co., Ltd., South Korea), as shown in Figure 4.3, is used in this research work. A brief description of the working principle of the PSD sensors and the specifications of the Best 5800 model machine are summarized in Table 4.1.



Figure 4.3 Computerized wheel alignment machine (Best 5800 Model).

Table 4.1 Specification of computerized wheel alignment machine (Best 5800 Model)

Sl. No	Specification
1	Sensor head Communication method Infrared ray
2	Measurement method PSD sensor
3	Wheel head with rechargeable 4 battery
4	Head sensor 8 nos
5	4 wheel fixing clamp Fix Up to 22" wheel, adjustable.
6	Caster +/- 30 ⁰
7	Camber +/- 10 ⁰
8	Toe +/- 12 ⁰ (+/- 150mm)
9	Set back +/- 22 ⁰
10	Thrust +/- 22 ⁰
11	S.A.I +/- 30 ⁰
12	Temperature range (4°C to 45°C)
13	Power supply 220V AC 50 Hz,
14	Operating platform with PC and Windows software
15	17" LCD Color display
16	Printer
17	2 turntable (Heavy steel made)
18	1 set break lock lever
19	1 set steering lock lever

The vehicle alignment system consists of pairs of sensors acting as active and passive sensors that cooperate with each other in order to determine the wheel alignment angles. The alignment software has an intuitive alignment sequence that provides a smooth and easy alignment procedure. The kit is equipped with a pair of long-arm sensors and a pair of short-arm sensors. This system provides reversible setups, i.e., interchangeable front and rear sensors, to avoid obstructions such as air dams or spoilers. The sensors come with rolling compensation. This provides precise alignment measurements without jacking up the vehicle. The sensors provide a continuous run-out compensation procedure to ensure accurate alignment angles, even if the wheels rotate after compensation. Being a wired system, it has cables to connect. The sensors are equipped with rechargeable batteries and can be charged when they are mounted on the aligner. The sensors are easy to handle because of their light weight and strong magnesium bodies. The sensors are compatible with all vehicles, even those with special adapters. It comes with a sensor locking mechanism to hold the sensors firmly. The alignment values do not change because of the interrupted measurement retention feature. The first step is to turn on the computer, after which the main menu with the alignment start

option appears on the screen. After starting the software, the next step is to enter the customer's identification. In this scenario, the driver's name and the test number were entered into the "Customer ID/Work Order ID" tab each time. This ensures that the test is correctly identified while sorting the measurement database. The customer identification screen is displayed at the start of the alignment program and includes the vehicle's make, model, and year selection. The software displays a list of available car models after selecting the vehicle's make. Finally, the wheel angle is adjusted to ensure proper alignment of the light vehicle.

4.4 LIST OF TOOLS REQUIRED FOR EXPERIMENTS

The details of the tools used in this study are summarized in Table 4.2.

Table 4.2 Necessary tools required for the experiments

SL. No	Name of the tool	Function
1	Digital Vernier Caliper	Measure the tire tread depth
2	Tire Tread & Brake lining Gauge	
3	Manual Vernier Caliper	
4	Measuring Tape	Measure the alignment pit hole
5	Air Compressor	Fill the air into tire
6	Tire Pressure Gauge	Measure the tire pressure
7	Hydraulic Jack	Lift the vehicle
8	Digital Temperature and Humidity Meter	Measure the air temperature and humidity
9	Engine RPM Meter	Measure the engine rpm
10	Digital Depth Gauge	Measure the tire tread depth
11	Digital Thermometer	Measure the engine outer surface temperature
12	Wheel Socket Wrench	Open and fitting the wheel
13	Oil Funnel	Fill the oil into fuel tank
14	Oil Measurement Biker	Measure the oil quantity
15	Open end ring spanner-14inch	Adjust the tie rod nut
16	Adjustable Wrench-12inch	
17	Open end ring spanner-19inch	
18	Open end ring spanner-17inch	
19	Open end ring spanner-13inch	
20	Open end ring spanner-12inch	
21	White Paint	Mark the tire wear measurement
22	Wheel Alignment Machine	Measure the wheel angles
23	Load Scale	Measure the vehicle load

4.5 LIST OF EXPERIMENTAL ANALYSIS PARAMETERS

In this study, various dynamic characteristics of the vehicle, such as wheel angles, tire wear, tire pressure, load condition, braking application, and road condition, are considered to analyze the vehicle's performance. These factors are briefly discussed below.

Wheel Angles: The study took into account four different types of wheel angles, including the front wheel left toe-in angle, the front wheel left toe-out angle, the front wheel total toe-in angle, and the front wheel total toe-out angle.

Tire Wear: Tire wear is investigated based on two different conditions: front wheel total toe-in angle and front wheel total toe-out angle.

Tire Pressure: A total of 21 different types of tire pressure, including over- and under-inflation, are taken into account to investigate the effect of tire pressure on fuel consumption in a light vehicle.

Load Condition: A total of twenty different load conditions are considered to analyze the effect of fluctuating vehicle loads on fuel consumption in a light vehicle.

Brake Application: In this study, the application of brakes while driving is taken into account to find out the effect of brake application on fuel consumption for a light vehicle.

Road Condition: For the analysis, three road pavement conditions, such as fair road pavement, poor road pavement, and very poor road pavement, are considered, as shown in Figure 4.4. The International Roughness Index (IRI) is a common tool for measuring road pavement roughness. The Bangladesh Roads and Highways Department (RHD) conducts regular nationwide surveys to obtain IRI data [203]. Their statistical data showed that the IRI values for fair, poor, and very poor road pavement are 1.8, 3, and 5.5, respectively. The Pavement Condition Rating (PCR) is also a well-known methodology for assessing the service life of flexible pavement conditions. The IRI data for several road conditions is then converted to PCR (pavement condition rating) values. For classifying conditions as "very poor", "poor", "fair", "good", and "very good", a range of PCR of 0–50, 51–70, 71–80, 81–90, and 91–100 is used, respectively [203].

Using the equation below, we can easily determine the state of the road pavement.

$$\begin{aligned}
 \text{Pavement Condition Rating (PCR)} &= 100 * e^{-0.18(\text{IRI})} \\
 &= 100 * e^{-0.18 * 1.8} \\
 &= 72.32 \text{ (fair pavement condition)}
 \end{aligned}$$



a) Fair road pavement condition



b) Poor road pavement condition



c) Very poor road pavement condition

Figure 4.4 Road pavement condition for vehicle testing.

4.6 EXPERIMENTAL PROCEDURE OF WHEEL ALIGNMENT

Computerized wheel alignment systems use PSD sensors at each wheel, and the geometrical relationship between the sensors is used to calculate the wheel alignment angles. The vehicle is set using the alignment pit and moves on the stands with turn plates, with the wheels adequately centered on the turn plates. It is ensured that the rear wheels are not bound to their end positions (for rollover protection). After placing the vehicle on the alignment pit, the vehicle transmission is put in park, and parking brakes are applied to steady the wheels. The alignment stands are leveled prior to placing the vehicle to ensure proper alignment. The vehicle tire pressures are adjusted to the manufacturer's specifications. The suspension and steering linkage components are inspected for wear, looseness, or damage. In the first step, turning on the computer

generates the computer screen, as shown in Figure 4.5. The second step after starting the software is to enter the customer identification, as shown in Figure 4.6. In this case, the driver's name and the test number are noted down every time in the "Customer ID/Work Order ID" tab. This ensures test identification while sorting the measurement database. Figure 4.6 shows the customer identification screen at the start of the alignment software, which includes selecting the make, model, and year of the vehicle under study. After selecting the make of the vehicle, the software lists different models of the car for selection.



Figure 4.5 Wheel alignment starting menu screen [204].

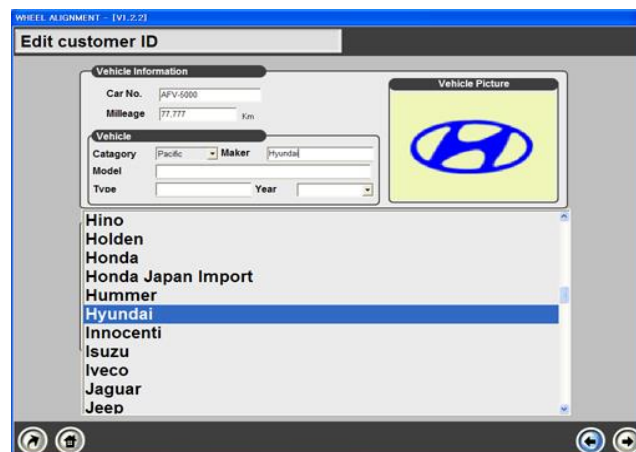


Figure 4.6 Customer identification screen [204].

The next step includes selecting the steering wheel positions, as shown in Figure 4.7. It is always important to maintain the centering status of the handle and lock up the fixing utensils. By turning the steering wheel, the center position of the left and right wheels is fixed, allowing the correct reading to be obtained. When the wheel angle adjustment is successful, the wheel is in the correct alignment position.

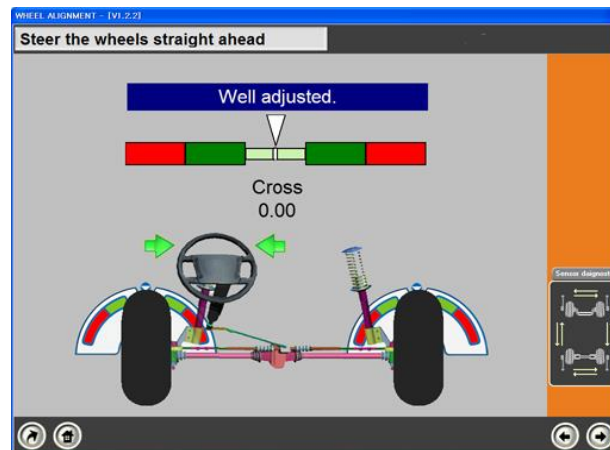


Figure 4.7 Steer the wheels straight ahead screen.

The next screen shows the brake pedal adjustment as in Figure 4.8. It has maintained brake function, locking up the vehicle. After completing the wheel angle adjustment function, the brake lock lever is attached to the brake pedal, and it is used to ensure that the vehicle wheel does not move during the wheel angle adjustment, which is important for wheel angle adjustment.



Figure 4.8 Vehicle brake pedal adjustment screen [204].

In Figure 4.9, the experimental system looks at the light vehicle's wheel angle position on the computer screen to see how much it differs from the standard data. It then uses the tie rod function to fix the misalignment and checks the alignment again to make it easy to move to a straight line. After completing the alignment, an alignment summary is printed by pressing the "Print" option.



Figure 4.9 Wheel alignment setup screen at CUET.

A detailed overview of the wheel alignment measuring system is shown in Figure 4.10, along with a flowchart of the steps.

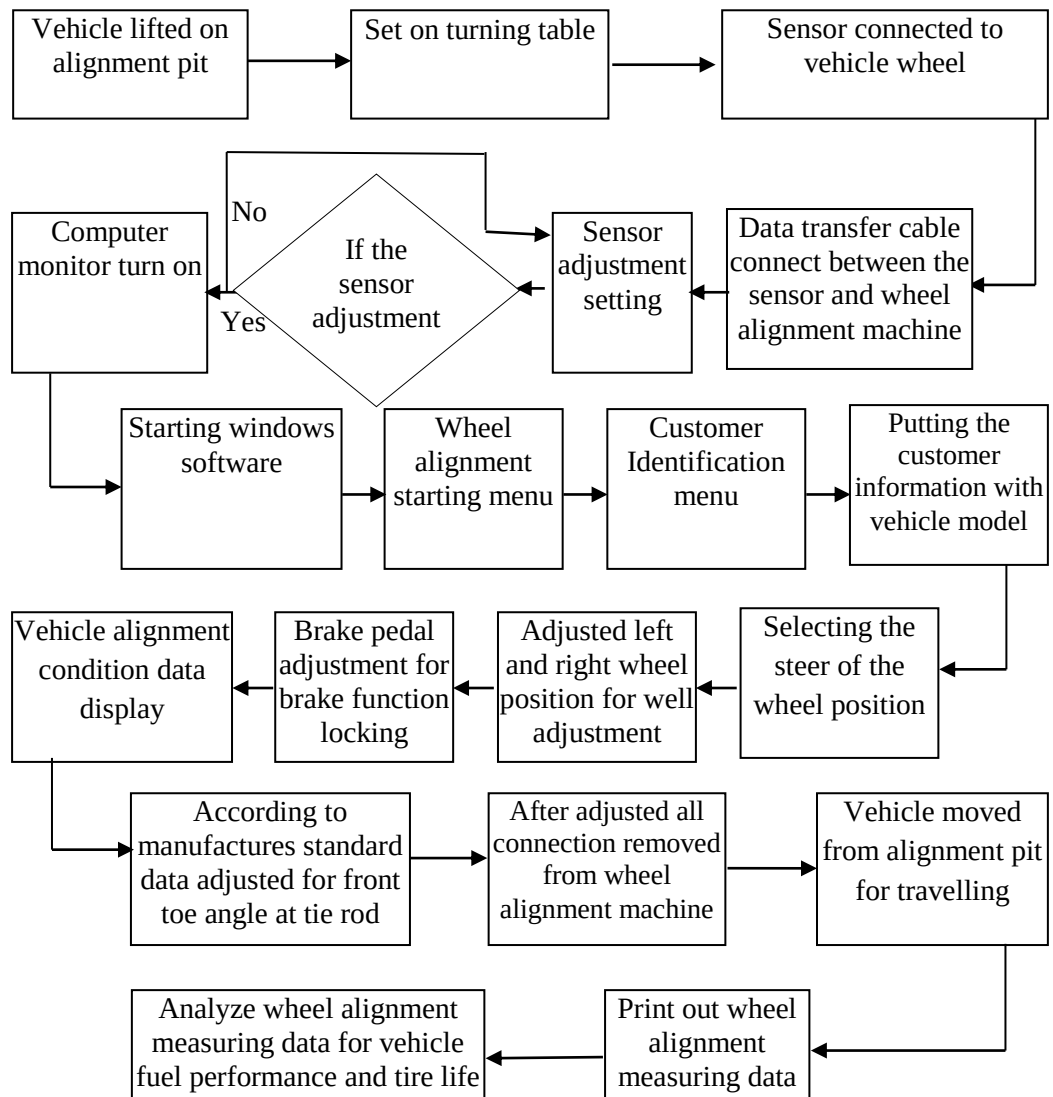


Figure 4.10 Flowchart of wheel alignment measuring system.

Measuring Steps

At first, the vehicle is lifted onto the alignment pit, and then a turning table is set up for wheel position. The sensors are connected to the vehicle's wheels and wheel alignment equipment, after which the sensors' adjustment settings are checked. After the settings have been completed, the computer software and wheel alignment start menu will run, displaying a customer identification menu in which the customer's identification will be entered. The steering wheel position is then adjusted to obtain correct wheel alignment, and the monitor display data is compared to the manufacturer's data before all connections are removed and the vehicle is moved from the alignment pit for traveling, as shown in flow chart Figure 4.10.

4.7 THE EXPERIMENTAL TEST CONDITIONS

Two different types of vehicles, the Toyota Echo Plus-2ZZ-GE-02 and the Toyota Corolla-2E-86, are considered for fuel performance and tire wear analysis based on wheel misalignment function and other related conditions. As mentioned below, the vehicle's characteristics, the suspension's condition, the engine's cylinder performance, and other alignment parameters are considered while measuring the correct toe angle.

4.7.1 Experimental Vehicle Type-I

The Toyota Echo Plus-2ZZ-GE-02 vehicle is used in the first test, as shown in Figure 4.11.



Figure 4.11 Experimental vehicle type-I.

During the test performance, the following testing environment and related parameters are maintained:

Table 4.3 Vehicle's characteristics and alignment parameters for vehicle type-I

<i>Vehicle Particulars:</i> Vehicle Model: Toyota Echo Plus-2ZZ-GE-02 Engine type: VVT Engine displacement: 1300 cc Tire Size: 175/70R14 Vehicle weight: 965 kg Air conditioning system: Non air conditioning Gear position: Auto transmission Type of fuel: Octane Tire pressure (Front and Rear) = 32 psi Vehicle speed = 30 km/hr. Travelling distance per test = 6 km Driver and one passenger weight = 125 kg <i>Alignment Conditions:</i> Caster angle (Left) = 0.20° Caster angle (Right) = 0.23° Camber angle (Left) = -0.25° Camber angle (Right) = -0.37°	<i>Suspension Conditions:</i> LH FRONT Suspension Weight: 1.49 kN Adherence: 57%, RH FRONT Suspension Weight: 1.76 kN Adherence: 60%, LH REAR Suspension Weight: 1.1 kN Adherence: 45%, RH REAR Suspension Weight: 1.12 kN Adherence: 51% <i>Engine Cylinder Performance Conditions:</i> Cylinder-1 = 80.63% Cylinder-2 = 80.10% Cylinder-3 = 80.63% Cylinder-4 = 80.10%
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4.7.2 Experimental Vehicle Type-II

In the second test, the Toyota Corolla-2E-86 vehicle is used, as shown in Figure 4.12.



Figure 4.12 Experimental vehicle type-II.

Similarly, during test performance, the following testing environment and related parameters are maintained:

Table 4.4 Vehicle's characteristics and alignment parameters for vehicle type-II

<i>Vehicle Particulars:</i> Vehicle Model: Toyota Corolla-2E-86 Engine displacement: 1300 cc Tire Size: 165/80R13 Vehicle weight: 830kg Air conditioning system: Non air conditioning Transmission condition: Manual Gear position: 2 Front tire pressure = 40 psi, Rear tire pressure = 40 psi Speed = 40 km/hr. Driver and one passenger weight = 125 kg <i>Alignment Conditions:</i> Caster angle (Left) = 0.12° Caster angle (Right) = 0.10° Camber angle (Left) = 0.17° Camber angle (Right) = -0.25°	<i>Suspension Conditions:</i> LH FRONT Suspension Weight: 1.43kN Adherence: 50%, RH FRONT Suspension Weight: 1.53kN Adherence: 45%, LH REAR Suspension Weight: 1.24kN Adherence: 42%, RH REAR Suspension Weight: 1.34kN Adherence: 19% <i>Engine Cylinder Performance Conditions:</i> Cylinder-1 = 78.84% Cylinder-2 = 79.37% Cylinder-3 = 79.37% Cylinder-4 = 77.78% <i>Tire Particulars</i> DOT, 20E9, EEC, 2721, SNI (83T), (E4), 027277, S2WR2, 0285667 Tubeless Radial, Max. Load - 487kg (1074lb), Max. Pressure - 44psi (300 kpi), MA-P₃-04 Tread: 1 Polyester + 2 Steel + 1 Nylon, Sidewall: 1 Polyester Treadware 440, Traction - A, Temperature – A, Made in Thailand
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For both types of vehicles, the above-mentioned experimental test conditions, such as vehicle particulars, suspension conditions, engine cylinder performance conditions, alignment conditions, road conditions, and tire particulars, remain constant, whereas weather temperature, humidity, engine outer temperature, and number of test runs are varied.

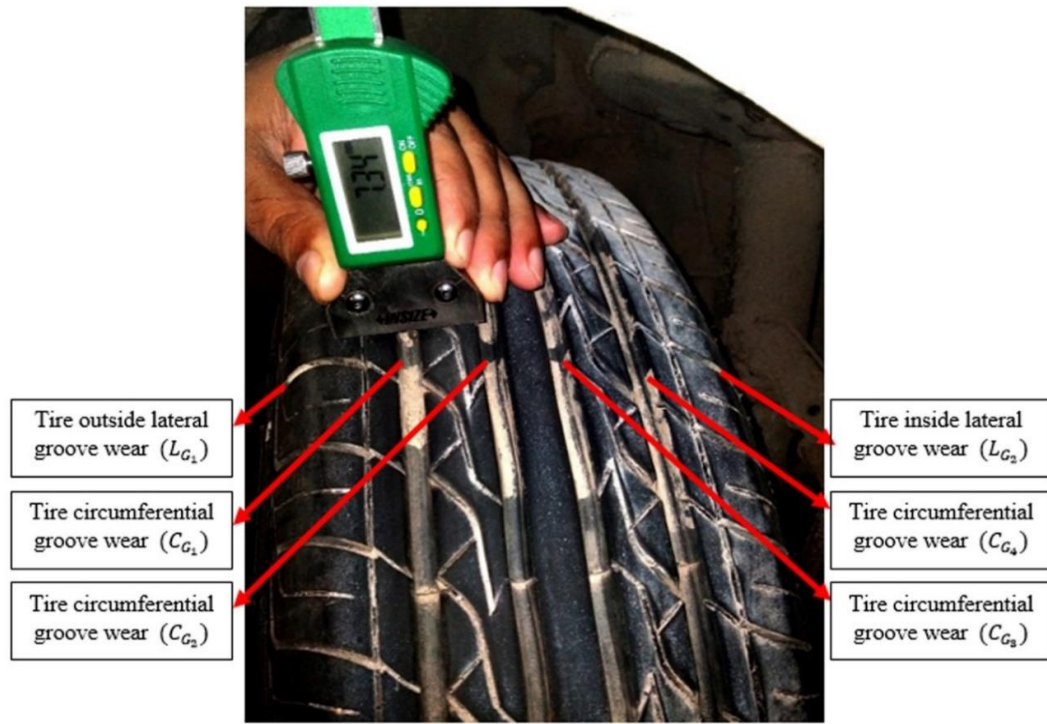
4.8 TIRE WEAR MEASUREMENT PROCESS

The Maxxis band tire (165/80R13, for vehicle type II) is used in this analysis, as shown in Figure 4.13(a). The tire tread is made up of two types of grooves: lateral grooves and

circumferential grooves. After each set of test runs, the tire wear is measured using a digital depth gauge, as shown in Figure 4.13(b). Many tests are performed over a period of time to get an overview and aim to minimize the error in the results. The tire wear is measured based on the wheel front total toe angle with a tire pressure of 40 psi and a vehicle speed maintained at 40 km/hr. In the first case, the depth of the lateral and circumferential grooves is measured after a travel distance of 500 km with one driver and one passenger at a 0° front wheel total toe angle. Then, the depth of the grooves is subsequently measured by the change in the front wheel's total toe angle. A light-duty vehicle weighing 955 kg, including the driver and one passenger, is used for all tests. The data is recorded, and details of it are stated in Table D.15 and Table D.17 of Appendix D. After traveling 3500 km by driving on two tires with front wheel total toe-in angles of 0.40°, 1.00°, 1.60°, 4.20°, and 4.80°. The values of tire wear are shown in Table D.15 and Table D.17 of Appendix D, which are consistent in nature. Similarly, the tire wear measurement of front wheel total toe-out angles at 0.00°, -0.40°, -1.00°, -1.60°, -4.20°, and -4.80° is recorded for 4000 km by incorporating two new tires, as shown in Table D.17 of Appendix D.



(a)



(b)

Figure 4.13 (a) Photograph of new wheel tire and (b) Tire tread depth measuring process.

Chapter 5

Results and Discussion

5.1 BACKGROUND

Vehicle safety and stability assurance are important factors in the automobile industry that ensure the comfort of passengers and drivers. The proper adjustments of wheel characteristics angle significantly positively affect vehicle performance. In addition, it is believed that wheel alignment is connected to driving comfort, fuel efficiency, and tire life. It should be mentioned that improper wheel alignment might cause an accident and unexpectedly quick tire wear. Additionally, it makes the car less controllable. The referenced literature in Chapter 2 represents a great deal of research conducted to improve the fuel efficiency of motor vehicles. The fuel efficiency of an automobile depends on several variables; among them, the speed of the vehicle, rolling resistance, the condition of the tires, tire pressure, and the alignment of the wheels are particularly frequent problems that affect fuel efficiency and tire life. Even though many experimental studies have been carried out to examine the fuel consumption of vehicles while accounting for various parameters, work related to wheel misalignment, which is thought to be an important factor affecting fuel consumption and vehicle performance, is relatively rare in the open literature. This chapter covered several aspects obtained from experimental results that affect wheel alignment, tire wear, and the fuel efficiency of light vehicles.

5.2 FINDINGS THE CAUSES OF WHEEL MISALIGNMENT

The automobile wheels are initially aligned, tested, adjusted, and verified by the manufacturer. There should not be any problems with wheel alignment in a new car. However, most drivers will notice that their wheels become out of alignment over time. Wheel misalignment has a wide range of causes, as discussed in Section 1.5. Here, some of the factors are examined using experimental data, which are as follows:

5.2.1 Wheel Misalignment Due to Increased Travel Distance

The vehicle's running distance is one of the factors that have been considered to analyze the wheel misalignment of different vehicles. The experimental data for three different

light vehicles are presented in Table 5.1. The vehicle is initially aligned properly with the help of a wheel alignment machine, and the wheel alignment is checked after a particular amount of travel distance. According to the analysis, the vehicle's wheel started to become out of alignment when its travel distance increased over time. As shown in Table 5.1, the toe angle of the Toyota Premio-2013 model changes by 00.03° (left toe-in angle) and 00.27° (right toe-in angle), respectively, with a travel distance of 943 km. After traveling 4645 km, the toe angle is measured again, and the results show that the angle has increased to 00.19° (left) and 00.31° (right).

Similarly, the toe angle is examined for the Toyota Corolla 2E-86 and the Toyota Echoplus 2ZZ-GE-02, showing that the vehicle toe angle changes with travel distance. It is well known that improper tire-road contact is associated with irregularities like potholes and bumps in the pavement surface, which may cause the vehicle wheel to become misaligned over time. A study by Rahimafrooz Auto Center has found that when the running distance is between roughly 4000 km and 5000 km, the vehicle's wheel alignment starts to become out of alignment [78]. Therefore, as part of the vehicle's preventive maintenance program, it is suggested that the wheel alignment be checked every 4000 to 5000 km of running distance or sooner.

Table 5.1 Test results for wheel alignment conditions

Sl. no	Vehicle Model	Travelling Distance (km)	Change of Front Wheel Toe Angle (Misalignment)
1	Toyota Premio-2013	943	00.03° (Left toe-in) and 00.27° (Right toe-in)
		4645	00.19° (Left toe-in) and 00.31° (Right toe-in)
2	Toyota Corolla-1986	400	00.06°(Left toe-in) and -00.18°(Right toe-out)
3	Toyota Echo Plus-2003	1427	-00.06° (Left toe-out) and 00.12° (Right toe-in)

5.2.2 Effect of Vehicle Load on Wheel Alignment

The weight of the vehicle has a significant impact on wheel misalignment. The effect of vehicle load on wheel alignment of the Toyota Corolla 2E-86 model is shown in Table 5.2. According to test results, the wheel's left toe-in angle and right toe-out angle are observed to be 00.06° and -00.18°, respectively, when the car is driving 400 km with a vehicle weight of 900 kg and tire pressure of 40 psi on a fair pavement road

condition. However, when the vehicle load increased to 1080 kg with the same tire pressure and road condition, the wheel toe angle changed to 00.87° (right toe-in angle) and -00.05° (left toe-out angle), respectively. This is most likely because increased tire-road contact pressure increases rolling resistance.

Table 5.2 Test results of vehicle load on wheel alignment

Vehicle Model	Travelling Distance (km)	Vehicle Load (kg)	Change of Front Wheel Toe Angle (Misalignment)
Toyota Corolla-2E-86	400	900	00.06°(Left toe-in) and -00.18°(Right toe-out)
Toyota Corolla-2E-86	406	1080	-00.05° (Left toe-out) and 00.87° (Right toe-in)

5.2.3 Effect of Road Condition on Wheel Alignment

The road surface condition or roughness is another important factor that directly affects the wheel alignment and lifespan of the tires. The effect of road conditions on wheel alignment is shown in Table 5.3. This study considers a Toyota Corolla-2E-86 vehicle to analyze the wheel alignment under two different road pavement conditions with a constant vehicle load. The test results show that the vehicle becomes misaligned more quickly when it runs on very poor road conditions. Table 5.3 shows that the wheel toe angles of 00.06° (left toe-in) and -00.18° (right toe-out) are observed when the vehicle is running on a fair pavement with a travel distance of 400 km; however, the toe angle changed to 00.22° (left toe-in angle) and 00.34° (right toe-in angle) when it ran on very poor pavement. It is due to the cars extreme tilting, bumping, and pressure on tire-road surface contact.

Table 5.3 Test result of road condition on wheel alignment

Vehicle Model	Travelling Distance (km)	Vehicle Load (kg)	Road Pavement Condition	Change of Front Wheel Toe Angle (Misalignment)
Toyota Corolla-2E-86	400	900	Fair	00.06°(Left toe-in) and -00.18°(Right toe-out)
Toyota Corolla-2E-86	403	900	Very poor	0.22° (Left toe-in) and 00.34° (Right toe-in)

5.2.4 Effect of Brake Application on Wheel Alignment

It should be noted that applying the brakes affects vehicle control significantly. The stability and safety of the vehicle, as well as accidents, greatly reduce the application of frequent brakes. The effect of brake application on wheel alignment is shown in Table 5.4. According to the experiment data in Table 5.4, the vehicle's wheel became misaligned as the applied brake increased. It is shown that the vehicle's left toe-in angle of the front wheel and the right toe-out angle are observed to be 00.06° and -00.18° , respectively, while 822 times of braking are applied over a 400 km journey on a fair pavement road condition. However, if the car's brake application increased to 1580 times while maintaining the same other conditions as before, the wheel toe alignment changed to -00.02° (left toe-out angle) and -00.20° (right toe-out angle). This occurs as a result of the wheel skidding and the frictional scuffing that occurred as a result of transferring tractive or braking forces, which led to a fast misalignment of the wheel.

Table 5.4 Test results of brake application on wheel alignment

Vehicle Model	Travelling Distance (km)	Vehicle Load(kg)	Brake Application (times)	Change of Front Wheel Toe Angle (Misalignment)
Toyota Corolla-2E-86	400	900	822	00.06° (Left toe-in) and -00.18° (Right toe-out)
Toyota Corolla-2E-86	402	900	1580	-00.02° (Left toe-out) and -00.20° (Right toe-out)

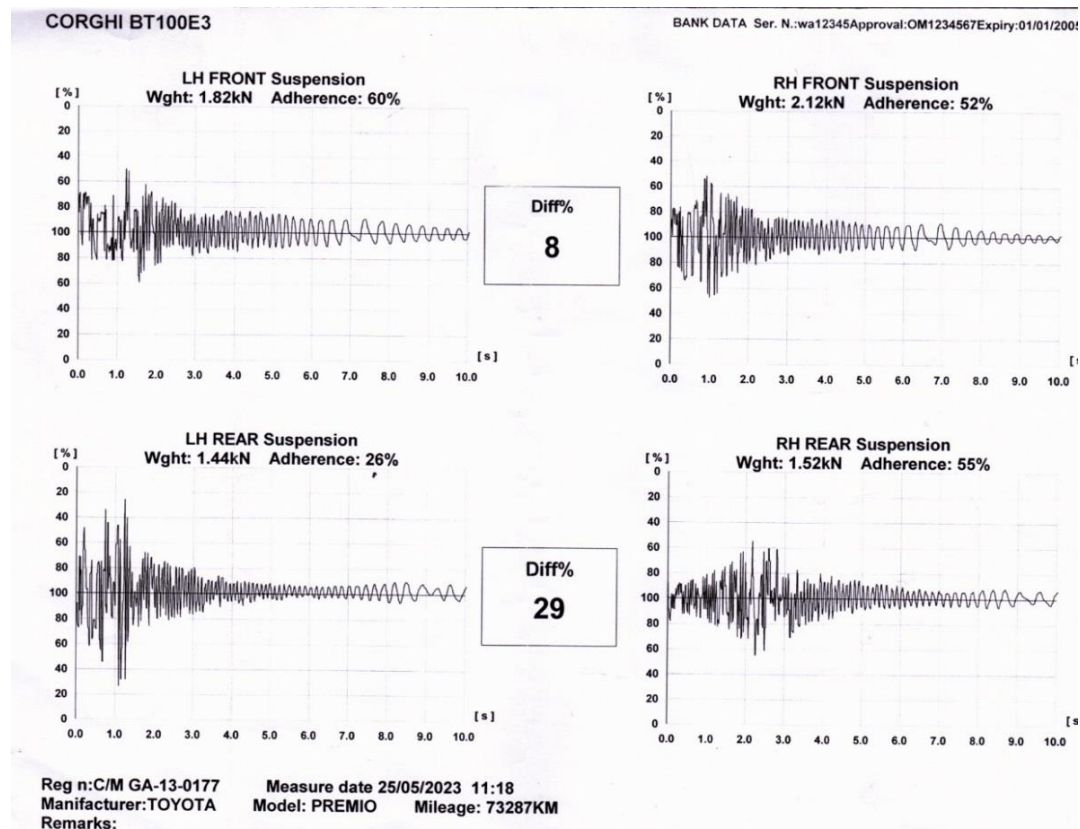
5.2.5 Change of Suspension Condition on Travelling Distance

The shock absorber conditions with the fixed vehicle model as a function of traveling distance are presented in Table 5.5 and graphically in Figure 5.1. Firstly, the suspension conditions of the vehicle are checked with the help of a suspension tester (a CORGHI BT100E3), while the odometer reading is 73287 km. The test results show that the adherence of the front left (L) and right (R) suspensions is 60% and 52%, respectively, and the adherence of the rear left (L) and right (R) suspensions is 26% and 55%, respectively. However, after driving the car for 4645 km with an odometer reading of 77932 km, the adherence of the front left (L) and right (R) suspensions is to be 44% and 50%, respectively. In the rear suspension, adherence is found to be 24% (L) and 30% (R), respectively. It is observed that shock absorber adherence capacity (absorbing capacity) for the front left (L) and right (R) suspensions had dropped to 16% and 2%,

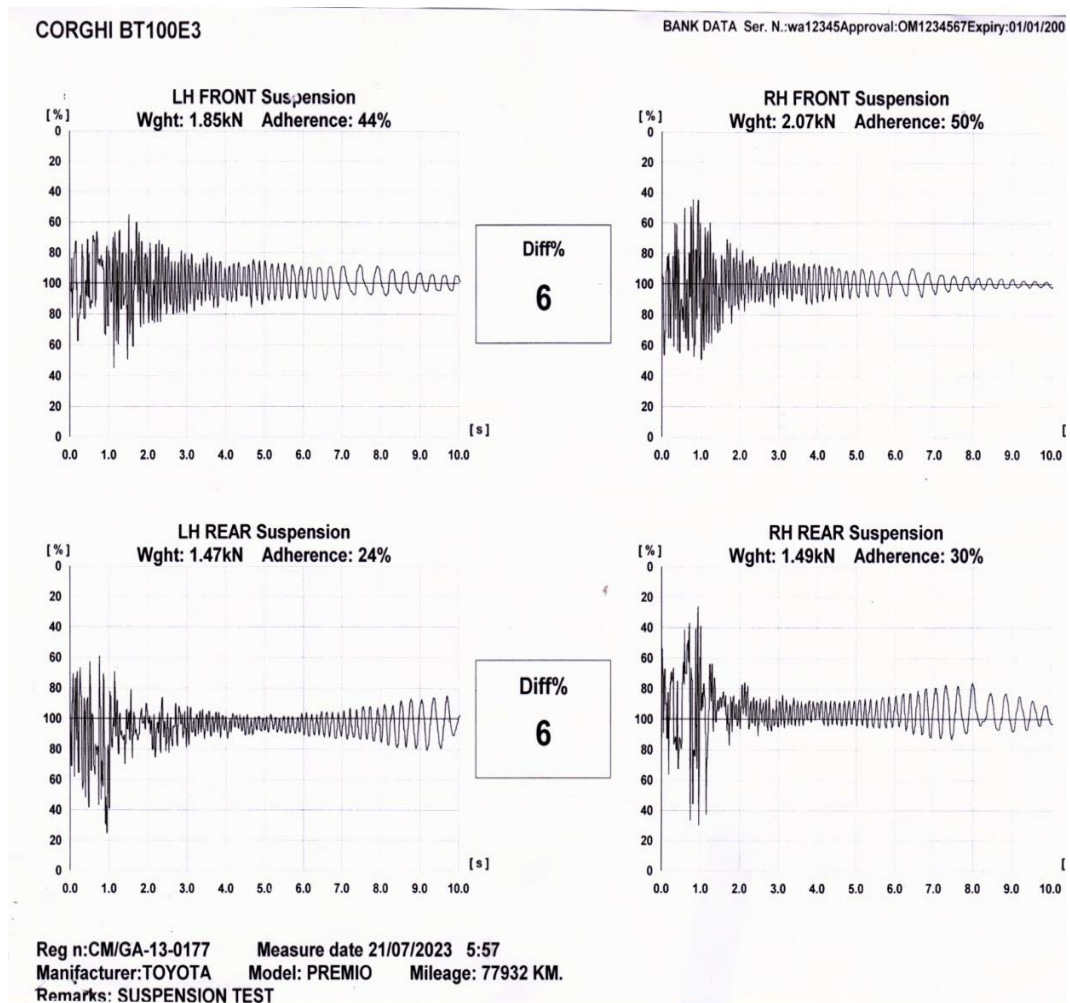
respectively. Whereas in the rear left (L) and right (R) suspensions, the shock absorber adherence capacity reduction is 2% and 25%, respectively. Therefore, it can be seen that shock absorber adherence decreased as the vehicle traveled a particular distance. It should be noted that the vehicle's wheel misalignment is directly related to the suspension system. A destructive (lower absorbing capacity) shock absorber could increase the chance of wheel misalignment and, hence, increase the fuel consumption rate.

Table 5.5 Test results of suspension condition on travelling distance

Vehicle Model	Odometer Reading	Travelling Distance (km)	LH Front Suspension Adherence	RH Front Suspension Adherence	LH Rear Suspension Adherence	RH Rear Suspension Adherence
Toyota Premio-2013	73287 km	-	60%	52%	26%	55%
Toyota Premio-2013	77932 km	4645	44%	50%	24%	30%



a) Before Travelling



(b) After 4645 km Distance of Travel

Figure 5.1 Shock absorber conditions.

5.3 EFFECT OF FRONT WHEEL LEFT TOE-IN ANGLE

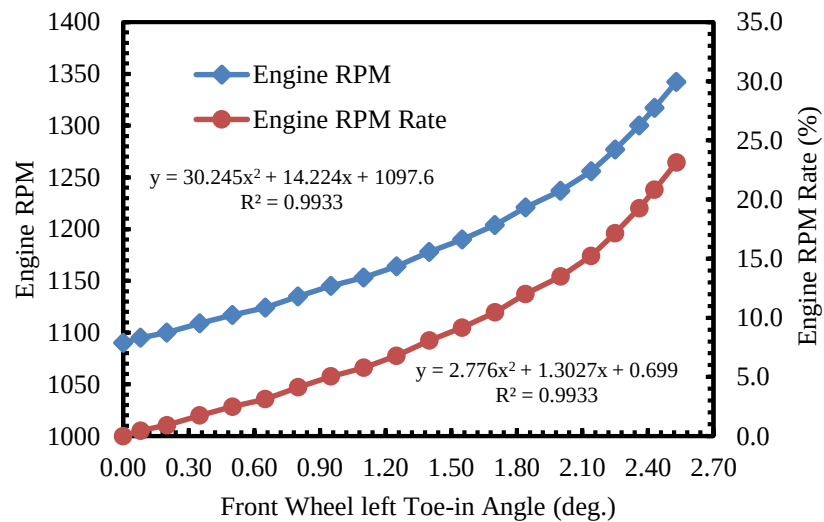
5.3.1 on Fuel Consumption

A Toyota Echo Plus-2ZZ-GE-02 light-duty vehicle is considered to analyze the effect of the front wheel's left toe-in angle on fuel performance. In the first case, the fuel consumption is measured based on the front wheel left toe-in angle on the basis of Table D.1 shown in Appendix D with a tire pressure of 32 psi (the recommended standard tire pressure for a 175/70R14 tire) when the vehicle speed is maintained at 30 km/hr. The fuel consumption rate is determined after a running distance of 6 km. Other parameters such as weather temperature (20.06 °C to 37.63 °C), humidity (29.0% to 85.67%), and engine surface temperature (77.97 °C to 82.73 °C) are also recorded during the tests. Each test has been performed three times over a period of time to determine the average

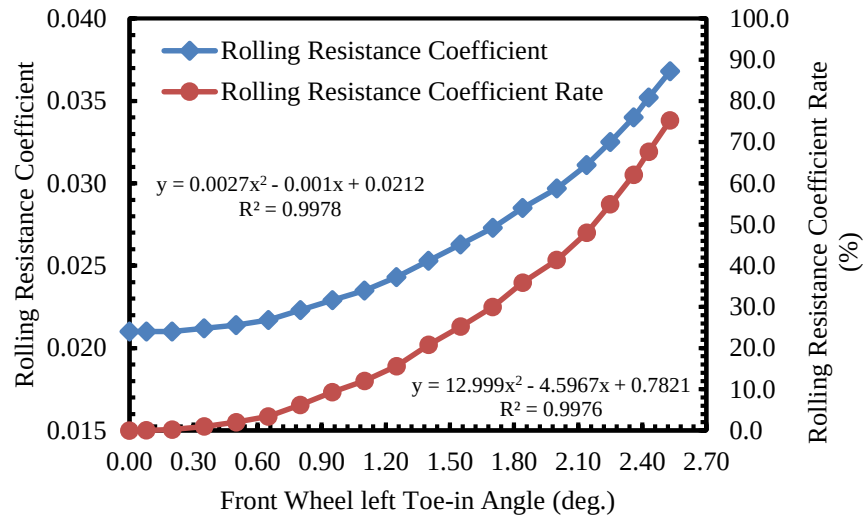
and minimize the error in the results. The same light-duty vehicle weighing 1090 kg, including the driver and one passenger, is used for all the tests. A total of 20 tests were conducted in this experiment.

The experimental test results of engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, fuel consumption, and KPL (kilometer per liter) of the light vehicle with respect to different front wheel left toe-in angles are shown in Table D.1 of Appendix D and graphically illustrated in Figures 5.2(a)–(f). The results of this experimental study show that engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, fuel consumption, and their rates increased under the steady test conditions (constant load, speed, and tire pressure) as the front wheel left toe-in angle increased.

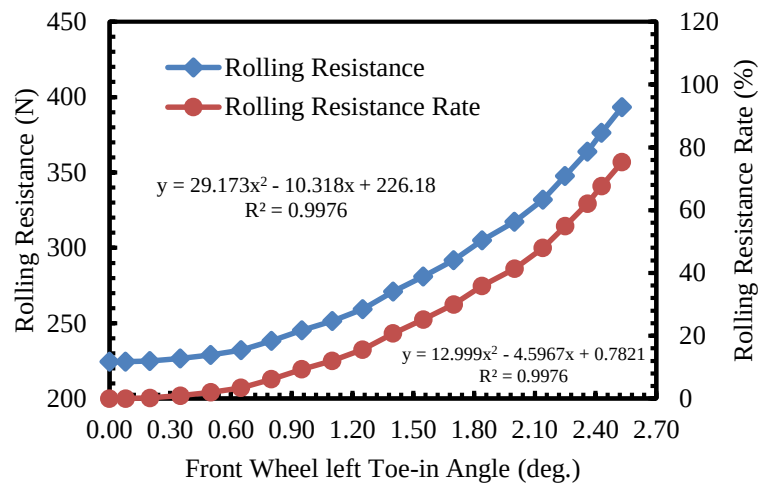
Figures 5.2(a)–(c) show that the engine rpm, rolling resistance coefficient, and rolling resistance all went up from 1090 rev/min to 1342 rev/min, 0.0210 to 0.0368, and 224.42 N to 393.39 N, respectively, when the front wheel left toe-in angle went from 0.00° to 2.53°. For the same condition (0.00° to 2.53°), the rates of engine rpm, rolling resistance coefficient, and rolling resistance are increased by up to 23.12%, 75.29%, and 75.29%, respectively, with respect to the 0.00° alignment condition. The increasing rolling resistance trend is in good agreement with the previously reported results [38,144].



(a)



(b)



(c)

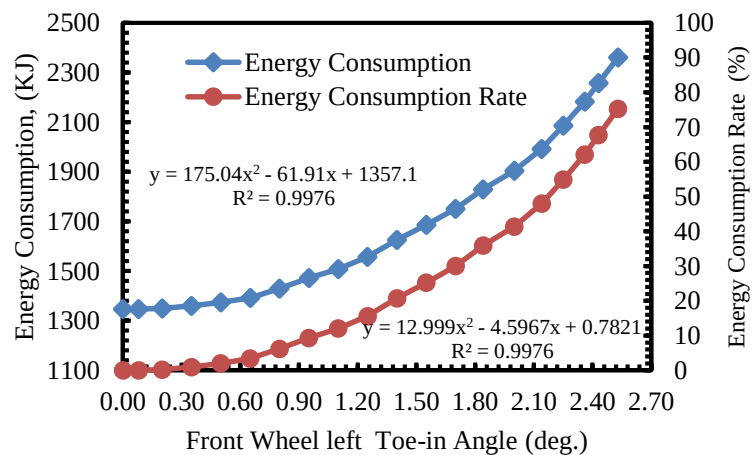
Figure 5.2(a) Variation of engine RPM and engine RPM Rate (%) with respect to front wheel left toe-in angle (deg.); (b) Change in rolling resistance coefficient and its rate (%) with respect to front wheel left toe-in angle (deg.); (c) Rolling resistance (N) and its rate (%) change with respect to front wheel left toe-in angle (deg.).

It is well known that increases in the toe-in angle cause the rolling resistance to increase more rapidly (Figure 5.2(c)). This is probably due to the lateral acceleration over the curve and the local stiffness of the surface. However, the rolling resistance generated is also dependent on a variety of factors, such as vehicle type, tire properties, tire materials, operating conditions, etc. Note that the engine speed would have to be increased to overcome the excessive rolling resistance generated due to the misalignment of the wheels (toe-in angle) to maintain the vehicle's steady speed. Hence,

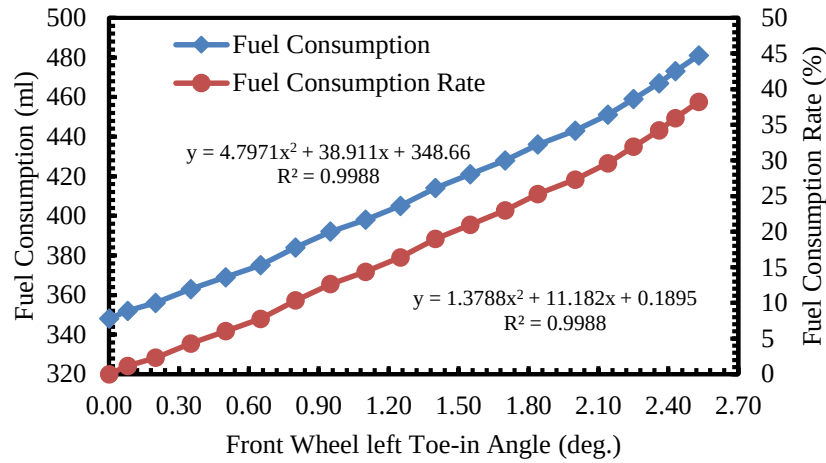
the energy consumption would increase as more fuel was burned (Figure 5.2(d)). Therefore, the rate of CO₂ emissions increases. Also, it is pointed out that low-rolling-resistance tires may help reduce CO₂ and NO emissions, as well as local pollutants like CO emissions from gasoline engines. Research shows that a few degrees of toe angle, equivalent to moderate-high cornering accelerations, can nearly double rolling resistance [37]. Figures 5.2(d) and (e) show energy consumption and fuel consumption varied with respect to the front wheel's left toe-in angle. Figure 5.2(d) shows that energy consumption increased non-linearly with front wheel left toe-in angles.

Conversely, the fuel consumption (Figure 5.2(e)) due to the front wheel's left toe-in angle is quite linear. According to Figures 5.2(d) and (e), the front wheel's left toe-in angle increased from 0.00° to 2.53°, which resulted in an increase in energy and fuel consumption from 1346.53 kJ to 2360.37 kJ and 348 ml to 481 ml, respectively. With the same alignment condition (0.00° to 2.53°), energy consumption and fuel consumption rates increased by up to 75.29% and 38.22%, respectively.

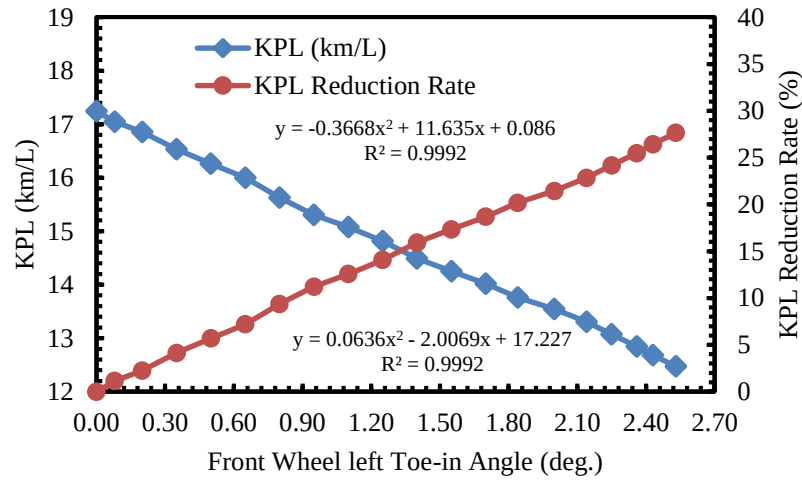
In Figure 5.2(f), the KPL change in ideal condition (constant vehicle speed, constant load, and fair road pavement) is presented with respect to the front wheel's left toe-in angle. Results show that the KPL decreased considerably with the increase in the front wheel's left toe-in angle. At a 0.00° front wheel left toe-in angle, the car's (KPL)_{ideal} is about 17.24 km/L, and the KPL reduction rate is 0%. When the front wheel's left toe-in angle is increased to 2.53°, the car's (KPL)_{ideal} is reduced to 12.47 km/L, and the KPL reduction rate is about 27.65%. Therefore, fuel consumption increased by 38.22%. It is seen that due to the misalignment of the front wheel's left toe-in angle (from 0.00° to 2.53°), the car traveled 4.77 km/L less (17.24 km/L from 12.47 km/L).



(d)



(e)



(f)

Figure 5.2(d) Change in energy consumption (Kj) and its rate (%) with respect to the front wheel left toe-in angle (deg.); (e) Variation of fuel consumption (ml) and its rate (%) with respect to front wheel left toe-in angle (deg.); (f) Change of KPL (Km/L) and its rate (%) with respect to front wheel left toe-in angle (deg.).

5.3.2 on Travelling Cost

The traveling cost per km in USD and BDT of the light vehicle as a function of the front wheel's left toe-in angle is presented in Table 5.6 and graphically in Figure 5.3. From Table 5.6 and Figure 5.3, it is shown that the traveling cost per km increased (0.0645\$ to 0.0892\$, or 5.1620Tk to 7.1348Tk) with the changes in the front wheel's left toe-in angle (0.00° to 2.53°), and the rate of traveling cost per km increased up to 38.22%. It is obvious that an increase in the front wheel's left toe-in angle can increase the fuel consumption of a light vehicle, as discussed in the above section, which results in increased travel costs.

Table 5.6 Experimental of different front wheel left toe-in angles and travelling costs

Front Wheel Left Toe-in Angle (deg.)	Travelling Cost Per km $T_{C/km} = f_c \times f_p$		Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km (min)}}{T_{C/km (min)}}$
	USD	BDT	
0.00	0.0645	5.1620	0.00
0.08	0.0653	5.2213	1.15
0.20	0.0660	5.2807	2.30
0.35	0.0673	5.3845	4.31
0.50	0.0684	5.4735	6.03
0.65	0.0695	5.5625	7.76
0.80	0.0712	5.6960	10.34
0.95	0.0727	5.8147	12.64
1.10	0.0738	5.9037	14.37
1.25	0.0751	6.0075	16.38
1.40	0.0768	6.1410	18.97
1.55	0.0781	6.2448	20.98
1.70	0.0794	6.3487	22.99
1.84	0.0808	6.4673	25.29
2.00	0.0821	6.5712	27.30
2.14	0.0836	6.6898	29.60
2.25	0.0851	6.8085	31.90
2.36	0.0866	6.9272	34.20
2.43	0.0877	7.0162	35.92
2.53	0.0892	7.1348	38.22

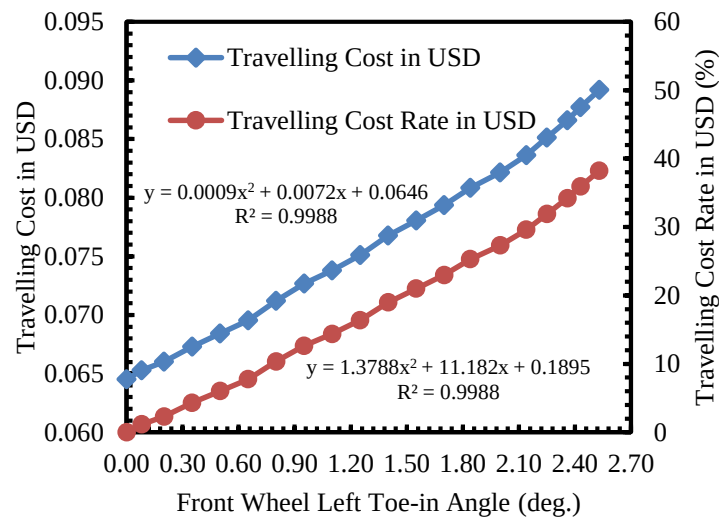


Figure 5.3 Change of travelling cost per Km in USD and its rate (%) with respect to front wheel left toe-in angle (deg.).

The experimental result is further verified by Pearson's correlation coefficient, presented in equation 3.14 of Section 3.7. The Pearson's correlation coefficient relationship between the front wheel's left toe-in angle and fuel consumption is presented in Table E.1 (Appendix E). The correlation coefficient $r_{\alpha_{fc}}$, is found to be 0.99 for the front wheel left toe-in angle and fuel consumption, which is within the range of $0.75 < r_{\alpha_{fc}} < 1$ [201]. It proves a very strong positive correlation between the front wheel's left toe-in angle and the fuel consumption of a light vehicle.

5.3.3 Correlation Development

A regression analysis is carried out to determine the best prediction equation describing the fuel consumption with regard to the front wheel's left toe-in angle. Regression models are chosen according to their best R-square and their level of statistical significance. It has been found that the proposed regression model has maintained a good correlation between the dependent variable (fuel consumption) and the independent variables (front wheel left toe-in angle, engine RPM, rolling resistance coefficient, rolling resistance, and energy consumption) with the experimental data. Based on the experimental data presented in Table 5.7 from Appendix Tables G.1(a), G.1(b), and G.1(c), ten different correlation models are developed using the regression method, which are: i) model 1- correlation between front wheel left toe-in angle and fuel consumption; ii) model 2- correlation between engine RPM and fuel consumption; iii) model 3- correlation between rolling resistance coefficient and fuel consumption; iv) model 4- correlation between rolling resistance and fuel consumption; v) model 5- correlation between energy consumption and fuel consumption; vi) model 6- correlation among front wheel left toe-in angle, engine RPM and fuel consumption; vii) model 7- correlation among front wheel left toe-in angle, rolling resistance coefficient and fuel consumption; viii) model 8- correlation among front wheel left toe-in angle, rolling resistance and fuel consumption; ix) model 9- correlation among front wheel left toe-in angle, energy consumption and fuel consumption, and x) model 10- correlation among front wheel left toe-in angle, engine RPM, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption.

Table 5.7 Different simple and multiple correlation regression models for front wheel left toe-in angle (deg.)

Model No.	Form of Equation	Equation constants		Multiple R	R Square	Adjusted R Square	Stand. Error	P-value
		m_i	k_1					
1	$(F_c)_{1,1} = m_1\alpha + k_1$	$m_1 = 51.1339$	344.0713	0.9974	0.9948	0.9946	3.1520	4.45615E-22
2	$(F_c)_{2,1} = m_1R/M + k_1$	$m_1 = 0.5389$	-229.3128	0.9863	0.9729	0.9714	7.2532	1.4727E-15
3	$(F_c)_{3,1} = m_1\mu_{RRC} + k_1$	$m_1 = 7949.3915$	201.2835	0.9726	0.9460	0.9430	10.2380	7.37759E-13
4	$(F_c)_{4,1} = m_1F_R + k_1$	$m_1 = 0.7440$	201.1225	0.9726	0.9461	0.9431	10.2302	7.27595E-13
5	$(F_c)_{5,1} = m_1E_c + k_1$	$m_1 = 0.1240$	201.1225	0.9726	0.9461	0.9431	10.2302	7.27596E-13
6	$(F_c)_{6,1} = m_1\alpha + m_2^*R/M + k_1$	$m_1 = 36.2548;$ $m_2 = 0.1629$	169.8920	0.9998	0.9996	0.9995	0.8601	3.12419E-11
7	$(F_c)_{7,1} = m_1\alpha + m_2^*\mu_{RRC} + k_1$	$m_1 = 40.0199;$ $m_2 = 1854.9158$	309.6869	0.9996	0.9993	0.9993	1.1155	2.66028E-09
8	$(F_c)_{8,1} = m_1\alpha + m_2^*F_R + k_1$	$m_1 = 39.9938;$ $m_2 = 0.1740$	309.5702	0.9997	0.9994	0.9993	1.0950	1.93683E-09
9	$(F_c)_{9,1} = m_1\alpha + m_2^*E_c + k_1$	$m_1 = 39.9938;$ $m_2 = 0.0290$	309.5702	0.9997	0.9994	0.9993	1.0950	1.93676E-09
10	$(F_c)_{10,1} = m_1\alpha + m_2R/M + m_3\mu_{RRC} + m_4F_R + m_5E_c + k_1$	$m_1 = 36.1101;$ $m_2 = 0.1751;$ $m_3 = -3072.7057;$ $m_4 = -12859.1472;$ $m_5 = 2143.2363$	160.2802	0.9998	0.9997	0.9996	0.7683	0.04183776
	$F_c = 160.2802329 + 36.11015186^*(\alpha) + 0.175167315^*(R/M) - 3072.705761^*(\mu_{RRC}) - 12859.14726^*(F_R) + 2143.23636^*(E_c)$							

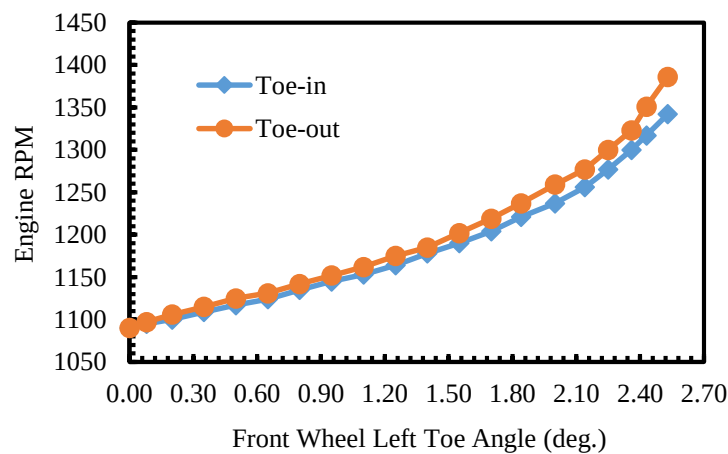
A total of 20 tests are performed per regression model. The R square, adjusted R square, and p -values of the corresponding regression model are given in Table 5.7. In model 1, it is seen that the values of R square and adjusted R^2 are 0.9948 and 0.9946, respectively; these values are very close, and it is also seen that the p -values are less than 0.05, which can be claimed to be a perfect relationship between the front wheel left toe-in angle and fuel consumption. Similarly, models 2–10 show a perfect relationship between the dependent and independent variables because the correlation R -square values for models tend towards 1. Based on these results, it can be said that the proposed regression models have kept a good correlation between the dependent variable (fuel consumption) and the independent variables (front wheel left toe-in angle, engine RPM, rolling resistance, rolling resistance coefficient, and energy consumption) from the experimental data.

5.4 COMPARISON RESULT FOR THE EFFECT OF FRONT WHEEL LEFT TOE-IN AND LEFT TOE-OUT ANGLE

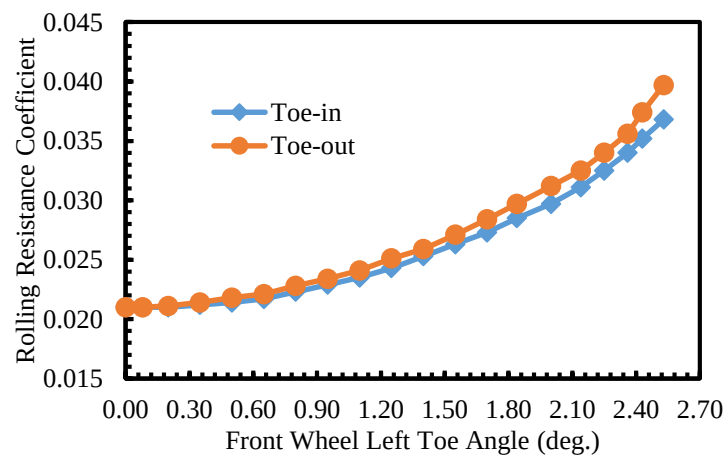
In the second case, similar test conditions are considered for the front wheel's left toe-out angle, as mentioned in Section 5.3. During the experiment, other parameters such as the weather temperature range from 21.70 °C to 32.27 °C, the humidity range from 52.0% to 86.67%, and the engine surface temperature range from 79.83 °C to 82.47 °C were also observed. Each test has been performed three times over a period of time to determine the average and minimize the error in the results. In the case of wheel toe-out angles, the fuel efficiency, KPL, and KPL rate decreased considerably as the front wheel toe-out angles (left) increased. It is observed that for the change in front wheel left toe-out angle from 0.00° to -2.53°, the rates of energy consumption and fuel consumption increased by up to 89.04% and 42.24%, respectively, as shown in Table D.2 in Appendix D. At a 0.00° front wheel left toe-out angle, the car's KPL is found to be about 17.24 km/L, and the KPL reduction rate is considered 0%. However, when the front wheel's left toe-out angle is increased to -2.53°, the car's KPL is reduced to 12.12 km/L, and the KPL reduction rate is about 29.70% (as shown in Table D.2 in Appendix D). Therefore, it is found that due to the misalignment of the front wheel's left toe-out angle (from 0.00° to -2.53°), the car travelled 5.12 km/h less.

According to the comparison results shown in Figures 5.4(a)–(e), the engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, and fuel

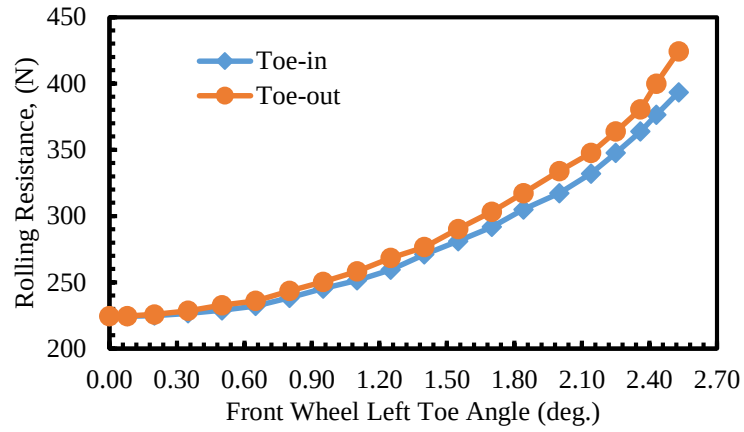
consumption somewhat depend on the front wheel's left toe angle. According to the data, rolling resistance grows as a symmetric second-order polynomial for toe-in/toe-out angles (Figure 5.4(c)). Wheel skid seems to become an increasingly significant effect at high toe-in/out misalignment angles. However, the result shows that the front wheel's left toe-out produces more rolling resistance compared to the front wheel's left toe-in angle. In general, the wheel toe-out angle can mitigate tire contact with the surface of the road, which will impact traction and braking performance. Therefore, the fuel consumption rate is slightly higher for the front wheel's left toe-out angle, as shown in Figure 5.4(e). In comparison to the front wheel's left toe-in angle, the fuel consumption rate is 4.02% higher at the front wheel's left toe-out angle. The deviation is most likely due to increased pressure on the rubber through the increased grip friction of the driven tires. Additionally, it is discovered that the KPL rate is 2.05% higher at the front wheel's left toe-in angle compared to the front wheel's left toe-out angle.



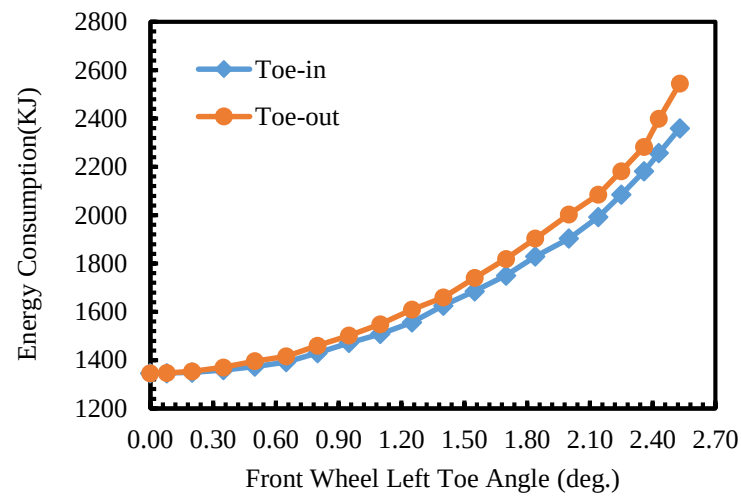
(a)



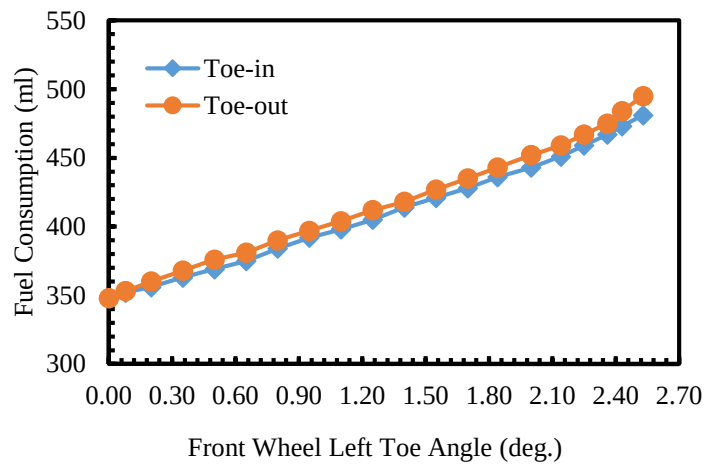
(b)



(c)



(d)



(e)

Figure 5.4 Comparison of (a) Engine RPM, (b) Rolling resistance coefficient, (c) Rolling resistance (N), (d) Energy consumption (Kj), and (e) Fuel consumption (ml) with respect to front wheel left toe-in and toe-out angles.

A comparison of the traveling cost per km in USD of the light vehicle as a function of the front wheel's left toe-in and toe-out angles is presented in Figure 5.5 (also listed in Table 5.6 and Appendix F.1). The front wheel's left toe-out angle is found to have a somewhat higher rate of travel. This is because the fuel consumption at the front wheel's left toe-out angle is higher compared to the front wheel's left toe-in angle.

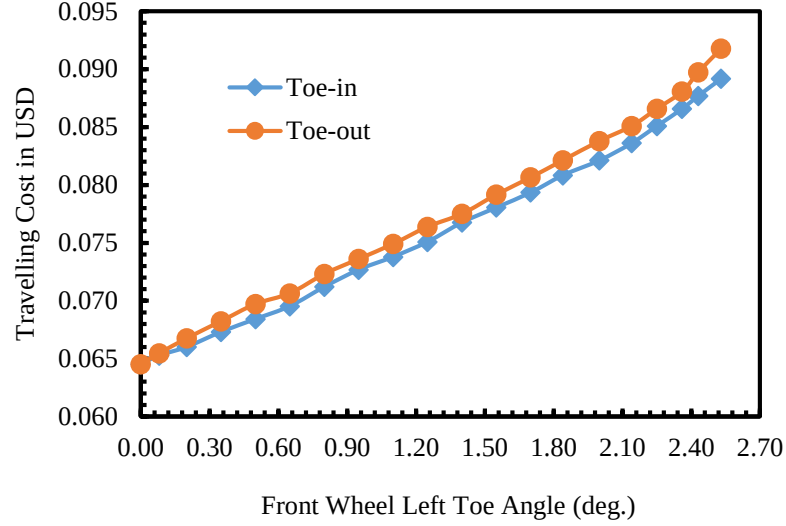


Figure 5.5 Comparison of Travelling Cost in USD with respect to front wheel left toe-in and toe-out angle.

Similarly, the experimental result of the front wheel's left toe-out angle is further verified by Pearson's correlation coefficient presented in Equation 3.14. The Pearson's correlation coefficient relationship between the front wheel's left toe-out angle and fuel consumption is presented in Table E.2 (Appendix E). The correlation coefficient $r_{\alpha_{fc}}$, absolute value is found to be 0.99 for the front wheel left toe-out angle and fuel consumption, which is within the range of $0.75 < r_{\alpha_{fc}} < 1$ [201], providing a very strong positive correlation between the front wheel left toe-out angle and the fuel consumption of a light vehicle.

5.5 EFFECT OF FRONT WHEEL TOTAL TOE-IN ANGLE

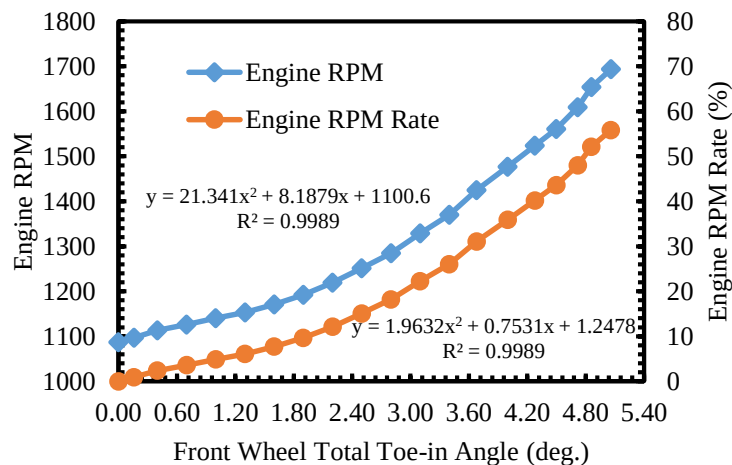
5.5.1 on Fuel Consumption

In the third case, the fuel consumption is measured based on front wheel total toe-in angle data based on Table D.3 shown in Appendix D with a tire pressure of 32 psi (the recommended standard tire pressure for 175/70R14 tires), whereas the vehicle speed is maintained at 30 km/hr. The fuel consumption rate is determined after a running

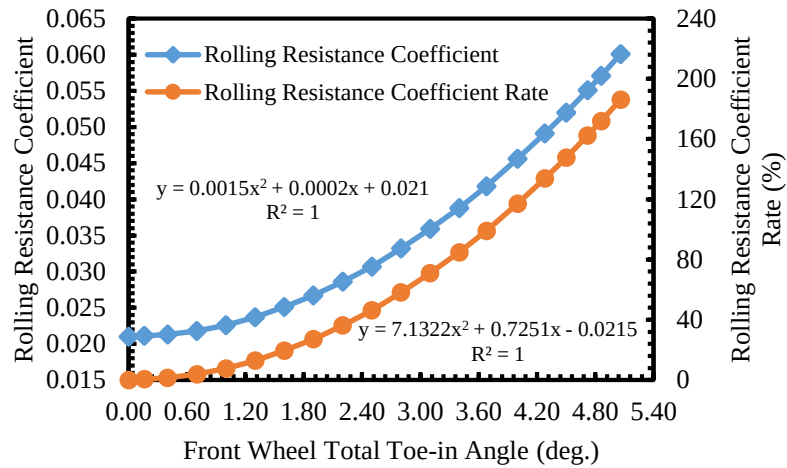
distance of 6 km. Other parameters remain the same as discussed in the previous experiment. Each test has been performed three times over a period of time to determine the average and minimize the error in the results. The same number of tests were conducted in this case, and the vehicle weighed 1090 kg (including the driver and one passenger).

The experimental test results of engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, fuel consumption, and KPL of the light vehicle for different front wheel total toe-in angles are graphically illustrated in Figures 5.6(a)–(f) on the basis of Table D.3 shown in Appendix D. The results show a similar pattern as discussed in previous sections: engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, fuel consumption, and their rates increased as the front wheel total toe-in angle increased.

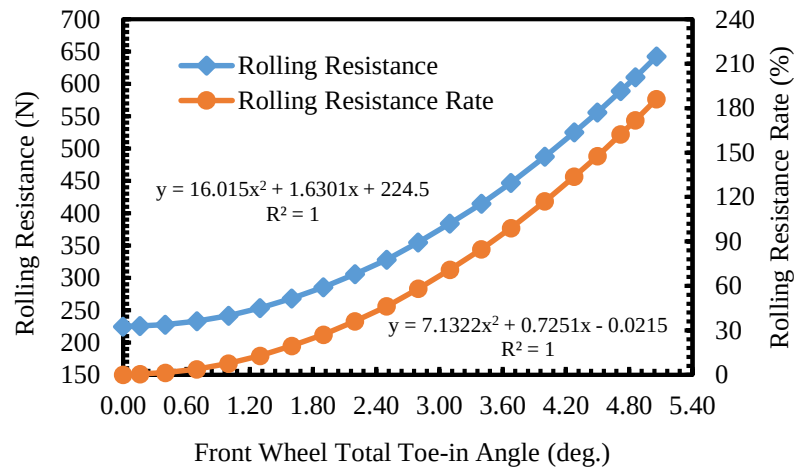
Figures 5.6(a)–5.6(c) show that when the front wheel total toe-in angle changes (from 0.00° to 5.06°), the engine rpm, rolling resistance coefficient, and rolling resistance also change (from 1087 rev/min to 1694 rev/min, from 0.0210 to 0.0601, and from 224.55 N to 642.64 N, respectively). The rates of engine rpm, rolling resistance coefficient, and rolling resistance are raised by up to 55.84%, 186.19%, and 186.19%, respectively, at 5.06° compared to a 0.00° total toe-in angle. The results obtained for the front wheel's total toe-in angle are like those obtained for the front wheel's left toe-in angle. The increasing trend or pattern of rolling resistance and its rates are in good agreement with the previously reported result [38,144].



(a)



(b)

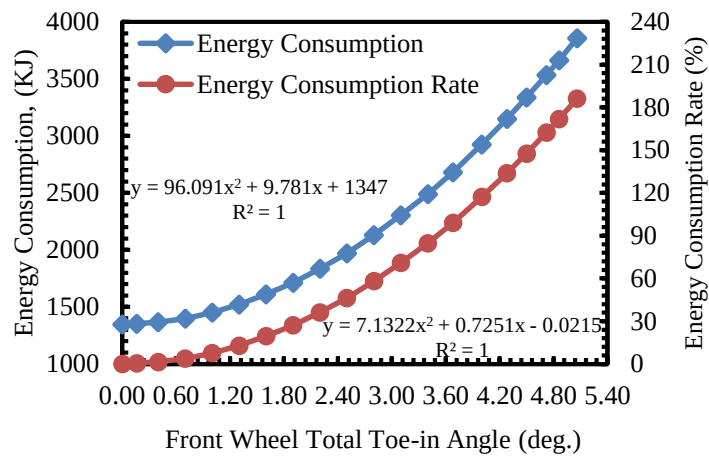


(c)

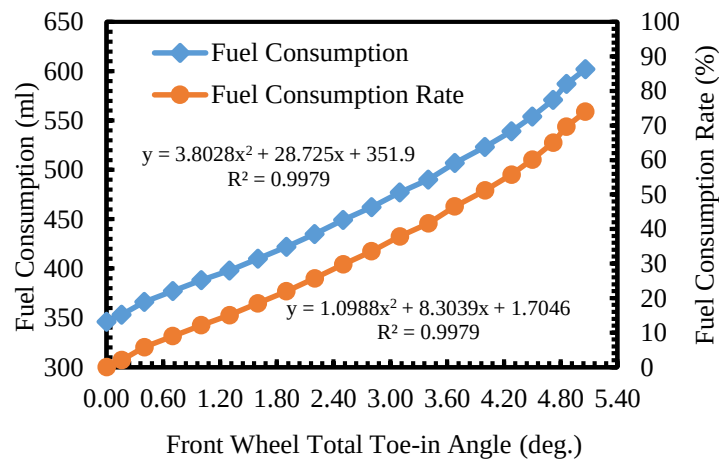
Figure 5.6 (a) Variation of engine RPM and its rate (%) with respect to front wheel total toe-in angle (deg.); (b) Change in rolling resistance coefficient and its rate (%) with respect to front wheel total toe-in angle (deg.); (c) Rolling resistance (N) and its rate (%) change with respect to front wheel total toe-in angle (deg.).

It is widely understood that when the toe-in angle increases, the rolling resistance also rises more rapidly (Figure 5.6(c)). This is most likely caused by the surface's local stiffness and lateral acceleration over the curve. In addition, the rolling resistance generated is also dependent on a variety of factors, such as vehicle type, tire properties, tire materials, operating conditions, etc. It should be noted that in order to maintain the vehicle's steady speed, the engine speed would need to be increased to overcome the extra rolling resistance produced by the increased toe-in angle of the wheels. Hence, energy consumption would increase as more fuel was burned (Figure 5.6(d)). Figures 5.6(d) and 5.6(e) show that energy consumption and fuel consumption varied with

respect to the front wheel's total toe-in angle. As shown in Figure 5.6(d), energy consumption increased non-linearly with front wheel total toe-in angles, whereas fuel consumption (Figure 5.6(e)) is quite linear. According to Figures 5.6(d) and (e), the adjustments in the front wheel's total toe-in angle from 0.00° to 5.06° caused an increase in energy and fuel consumption from 1347.30 kJ to 3855.85 kJ and 346 ml to 602 ml, respectively. Therefore, the energy and fuel consumption rates increased by up to 186.19% and 73.99%, respectively, when the front wheel's total toe-in angle increased from 0.00° to 5.06°. Compared to the toe-in angle of one side wheel (the left front wheel), the energy and fuel consumption rates are almost doubled for the total toe-in angle of 5.06°. This is because the misalignment of the total toe-in angle caused rolling resistances to be generated in both wheels.



(d)



(e)

Figure 5.6(d) Change in energy consumption (Kj) and Its Rate (%) with respect to the front wheel total toe-in angle (deg.); **(e)** Variation of fuel consumption (ml) and its rate (%) with respect to front wheel total toe-in angle (deg.).

In Figure 5.6(f), the KPL change is presented with respect to the front wheel's total toe-in angle. Figure 5.6(f) shows that the KPL decreased considerably with the front wheel's total toe-in angle increase. At a 0.00° front wheel total toe-in angle, the car's KPL is about 17.34 km/L, and the KPL reduction rate is 0%. When the front wheel's total toe-in angle is increased to 5.06°, the car's KPL is reduced to 9.96 km/L, and the KPL reduction rate is about 42.53%. Therefore, fuel consumption increased by 73.99%. It is seen that due to the misalignment of the front wheel's total toe-in angle (from 0.00° to 5.06°), the car traveled 7.37 km/L less (17.34 km/L from 9.96 km/L).

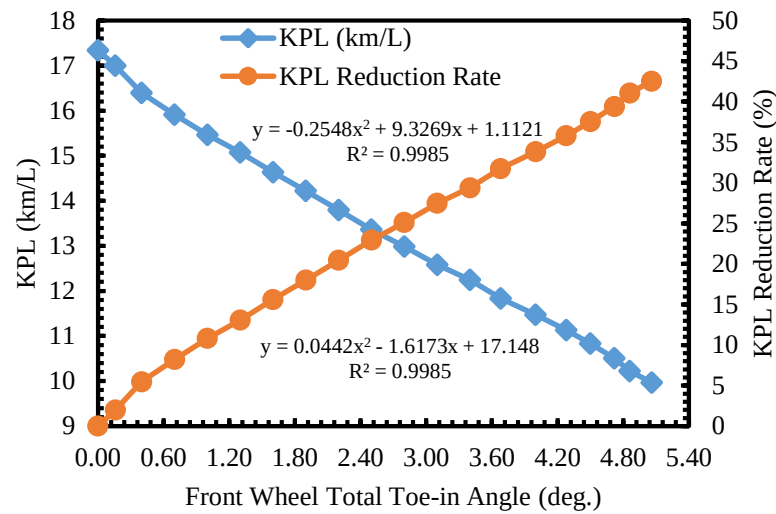


Figure 5.6(f) Change of KPL (km/L) and its rate (%) with respect to front wheel total toe-in angle (deg.).

5.5.2 on Travelling Cost

The traveling cost per km in USD and BDT of the light vehicle as a function of the front wheel total toe-in angle is presented in Table 5.8 and illustrated in Figure 5.7. Table 5.8 shows that as the front wheel total toe-in angle changed from 0.00° to 5.06°, the traveling cost per km increased from 0.0642\$ to 0.1116\$, or from 5.1323Tk to 8.9297Tk, at a rate of up to 73.99%. An increase in front wheel total toe-in angle can increase the fuel consumption of a light vehicle, as discussed in the above section, which results in increased travel costs.

Table 5.8 Experimental data for 20 test runs for different front wheel total toe-in angles and travelling costs

Front Wheel Total Toe-in Angle (deg.)	Travelling Cost Per km $T_{C/km} = f_c \times f_p$		Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km (min)}}{T_{C/km (min)}}$
	USD	BDT	
0.00	0.0642	5.1323	0.00
0.16	0.0655	5.2362	2.02
0.40	0.0679	5.4290	5.78
0.70	0.0699	5.5922	8.96
1.00	0.0719	5.7553	12.14
1.30	0.0738	5.9037	15.03
1.60	0.0760	6.0817	18.50
1.90	0.0782	6.2597	21.97
2.20	0.0807	6.4525	25.72
2.50	0.0833	6.6602	29.77
2.80	0.0857	6.8530	33.53
3.10	0.0884	7.0755	37.86
3.40	0.0909	7.2683	41.62
3.68	0.0940	7.5205	46.53
4.00	0.0970	7.7578	51.16
4.28	0.0999	7.9952	55.78
4.50	0.1027	8.2177	60.12
4.72	0.1059	8.4698	65.03
4.86	0.1088	8.7072	69.65
5.06	0.1116	8.9297	73.99

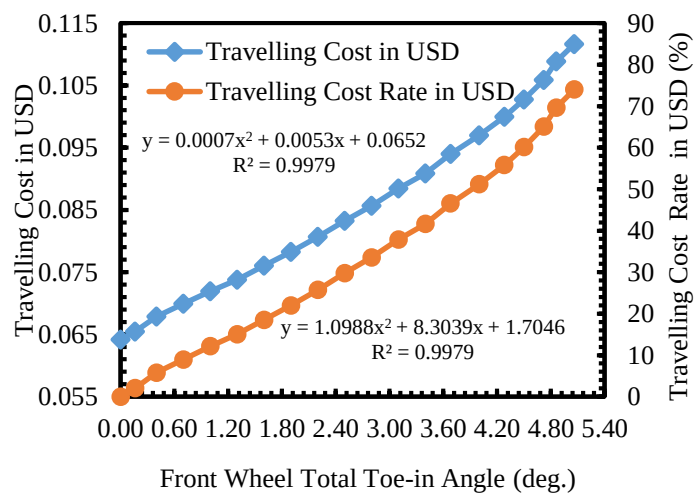


Figure 5.7 Change of travelling cost per Kilometer in USD and its rate in USD (%) with respect to front wheel total toe-in angle (deg.).

Once again, the experimental result is verified by the Pearson's correlation coefficient presented in equation 3.14 of Section 3.7. The Pearson's correlation coefficient relationship between the front wheel's total toe-in angle and fuel consumption is presented in Table E.3 (Appendix E). The correlation coefficient is found to be 0.99 for the front wheel total toe-in angle and fuel consumption, which is within the range of $0.75 < r_{\alpha f_c} < 1$ [201], providing a very strong positive correlation between the front wheel total toe-in angle and the fuel consumption of a light vehicle.

5.5.3 Correlation Development

Based on experimental data as shown in Table 5.9 from Appendix Tables G.2(a), G.2(b), and G.2(c), ten different correlation models are developed using the regression method, which are: i) model 1- correlation between front wheel total toe-in angle and fuel consumption; ii) model 2- correlation between engine RPM and fuel consumption; iii) model 3- correlation between rolling resistance coefficient and fuel consumption; iv) model 4- correlation between rolling resistance and fuel consumption; v) model 5- correlation between energy consumption and fuel consumption; vi) model 6- correlation among front wheel total toe-in angle, engine RPM, and fuel consumption; vii) model 7- correlation among front wheel total toe-in angle, rolling resistance coefficient, and fuel consumption; viii) model 8- correlation among front wheel total toe-in angle, rolling resistance, and fuel consumption; ix) model 9- correlation among front wheel total toe-in angle, energy consumption, and fuel consumption; and x) model 10- correlation among front wheel total toe-in angle, engine RPM, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption. The proposed regression model has been found to maintain a strong correlation with the experimental data between the independent variables (front wheel total toe-in angle, engine RPM, rolling resistance coefficient, rolling resistance, and energy consumption) and the dependent variable (fuel consumption).

A total of 20 tests are performed per regression model. The R square, adjusted R square, and p -values of the corresponding regression model are displayed in Table 5.9. In model 1, the R^2 value is found to be 0.9867. The values of R square and adjusted R^2 are 0.9867 and 0.9860, respectively; these values are pretty close, and it is also seen that the p -values are less than 0.05, indicating that the front wheel total toe-in angle and fuel consumption had a perfect correlation. Similarly, models 2–10 show a perfect

relationship between the dependent and independent variables because the correlation R -square values for models tend towards 1.

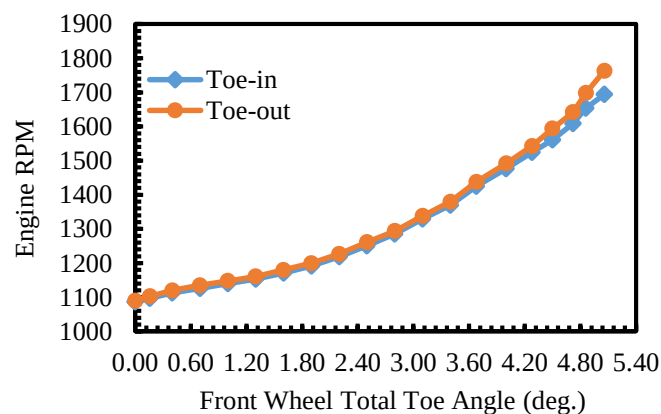
Table 5.9 Different simple and multiple correlation regression models for front wheel total toe-in angle (deg.)

Model No.	Form of Equation	Equation constants		Multiple R	R Square	Adjusted R Square	Stand. Error	P-value
		m_i	k_i					
1	$(F_c)_{1,1} = m_1\alpha + k_1$	$m_1 = 48.1045$	337.3432	0.9933	0.9867	0.9860	9.5789	2.32462E-18
2	$(F_c)_{2,1} = m_1R/M + k_1$	$m_1 = 0.3981$	-64.3301	0.9904	0.9810	0.9799	11.4686	5.95601E-17
3	$(F_c)_{3,1} = m_1\mu_{RRC} + k_1$	$m_1 = 5951.6981$	251.1278	0.9881	0.9764	0.9751	12.7869	4.2316E-16
4	$(F_c)_{4,1} = m_1F_R + k_1$	$m_1 = 0.5566$	251.1278	0.9881	0.9764	0.9751	12.7870	4.23163E-16
5	$(F_c)_{5,1} = m_1E_c + k_1$	$m_1 = 0.0927$	251.1278	0.9881	0.9764	0.9751	12.7869	4.23159E-16
6	$(F_c)_{6,1} = m_1\alpha + m_2^*R/M + k_1$	$m_1 = 26.7725;$ $m_2 = 0.1824$	151.4921	0.9993	0.9986	0.9984	3.2005	9.9381E-10
7	$(F_c)_{7,1} = m_1\alpha + m_2^*\mu_{RRC} + k_1$	$m_1 = 28.3642;$ $m_2 = 2535.6604$	298.6453	0.9989	0.9978	0.9975	3.9957	4.45299E-08
8	$(F_c)_{8,1} = m_1\alpha + m_2^*F_R + k_1$	$m_1 = 28.3642;$ $m_2 = 0.2371$	298.6453	0.9989	0.9978	0.9975	3.9957	4.45299E-08
9	$(F_c)_{9,1} = m_1\alpha + m_2^*E_c + k_1$	$m_1 = 28.3642;$ $m_2 = 0.0395$	298.6453	0.9989	0.9978	0.9975	3.9957	4.45299E-08
10	$(F_c)_{10,1} = m_1\alpha + m_2R/M + m_3\mu_{RRC} + m_4F_R + m_5E_c + k_1$	$m_1 = 25.9333;$ $m_2 = 0.3997;$ $m_3 = 0;$ $m_4 = 0;$ $m_5 = -0.0492$	-21.7483	0.9994	0.9989	0.8737	2.8739	#NUM!
$F_c = -21.74833029 + 25.93335352^*(\alpha) + 0.399728948^*(R/M) + 0^*(\mu_{RRC}) + 0^*(F_R) - 0.049201987^*(E_c)$								

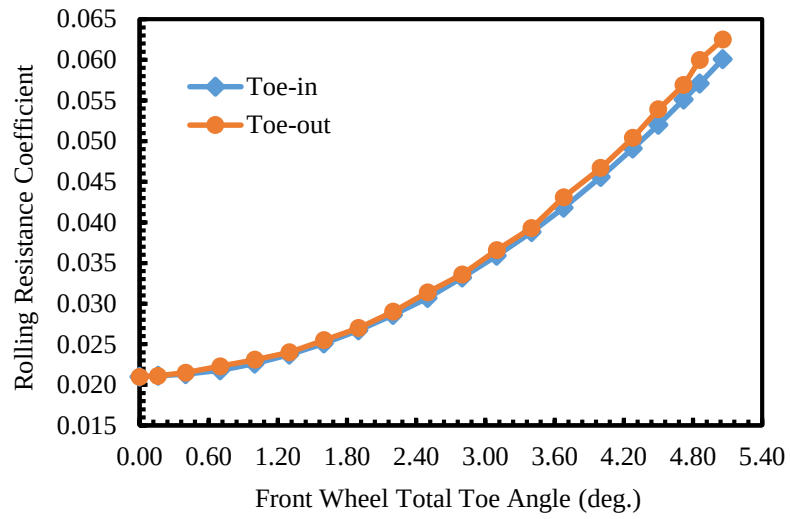
5.6 COMPARISON RESULT OF FRONT WHEEL TOTAL TOE-IN AND TOTAL TOE-OUT ANGLE

In the fourth case, the engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption are measured for the front wheel's total toe-out angle change. Similar test conditions are considered in this case, as mentioned in Section 5.3. In the case of wheel total toe-out angles, the fuel efficiency, KPL, and KPL rate decreased considerably as the front wheel toe-out angles (total) increased. It has been observed that for the front wheel total toe-out angle (0.00° to -5.06°), the rates of energy consumption and fuel consumption increased by up to 197.94% and 79.31%, respectively (as shown in Table D.4 in Appendix D). It is also observed that the car's KPL dropped from 17.24 km/L to 9.61 km/L at a total toe-out angle of -5.06° , with a KPL decrease rate of roughly 44.23%. Thus, it is observed that the car traveled 7.63 km/h less as a result of the front wheel's total toe-out angle being out of alignment (from 0.00° to -5.06°).

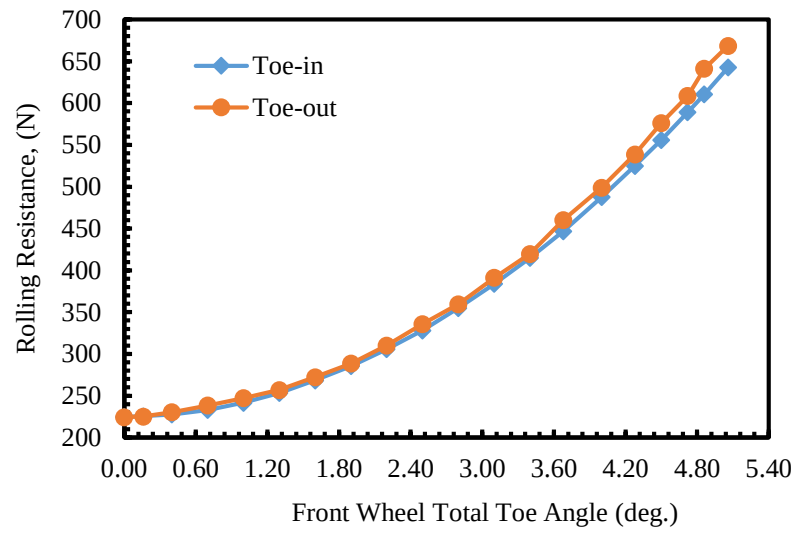
Figures 5.8(a)–(e) show the comparison results of engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption as a function of the front wheel total toe angle. The fuel consumption rate is a little bit higher (5.32%) for the front wheel's total toe-out angle compared to the front wheel's total toe-in angle. Therefore, the KPL rate of 1.70% is higher at the front wheel's total toe-in angle. Again, this is most likely caused by the surface's local stiffness and lateral acceleration over the curve. The result also exhibits that the front wheel's total toe-out produces more rolling resistance compared to the front wheel's total toe-in angle. Moreover, the rolling resistance generated is also dependent on a variety of factors, such as vehicle type, tire properties, tire materials, operating conditions, etc.



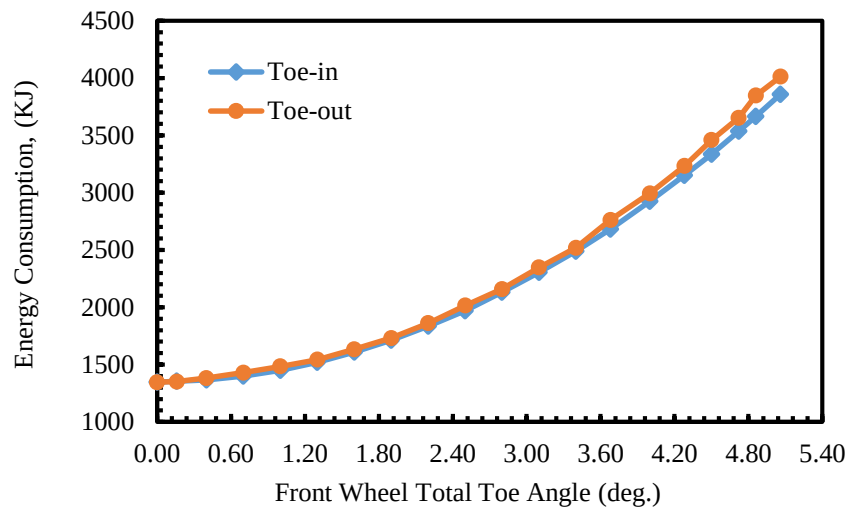
(a)



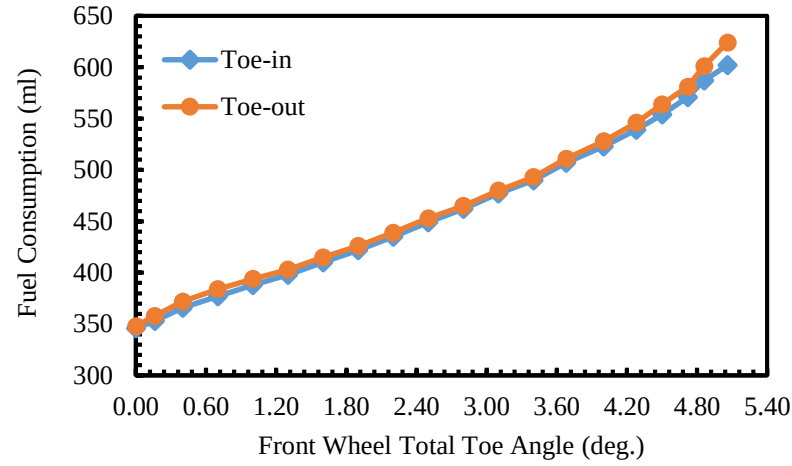
(b)



(c)



(d)



(e)

Figure 5.8 Comparison of (a) Engine RPM, (b) Rolling resistance coefficient, (c) Rolling resistance (N), (d) Energy consumption (Kj), and (e) Fuel consumption (ml) with respect to front wheel total toe-in and toe-out angle.

A comparison of the traveling cost per km in USD of the light vehicle as a function of the front wheel's total toe-in and toe-out angles is presented in Figure 5.9 from Table 5.8 and Appendix F.2. It is observed that the rate of travel is a little bit higher for the front wheel's total toe-out angle. In comparison to the front wheel's total toe-out angle, the traveling cost rate of 5.32% is higher at the front wheel's total toe-in angle. This is because the fuel consumption at the wheel's front total toe-out angle is higher compared to the front total toe-in angle.

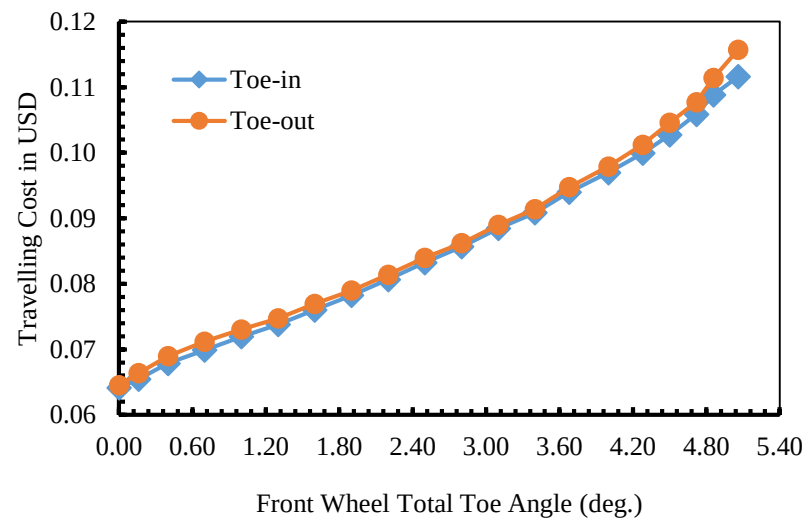


Figure 5.9 Comparison of travelling cost in USD with respect to front wheel total toe-in and toe-out angle.

The Pearson's correlation coefficient relationship between the front wheel's total toe-out angle and fuel consumption is presented in Table E.4 (Appendix E). The correlation coefficient $r_{\alpha_{fc}}$, absolute value is found to be 0.98 for the front wheel total toe-out angle and fuel consumption, which is within the range of $0.75 < r_{\alpha_{fc}} < 1$ [201]. It proves a very strong positive correlation between the front wheel's total toe-out angle and the fuel consumption of a light vehicle.

5.7 EFFECT OF TIRE PRESSURE

5.7.1 on Fuel Consumption (at 0° Front Wheel Total Toe Angle)

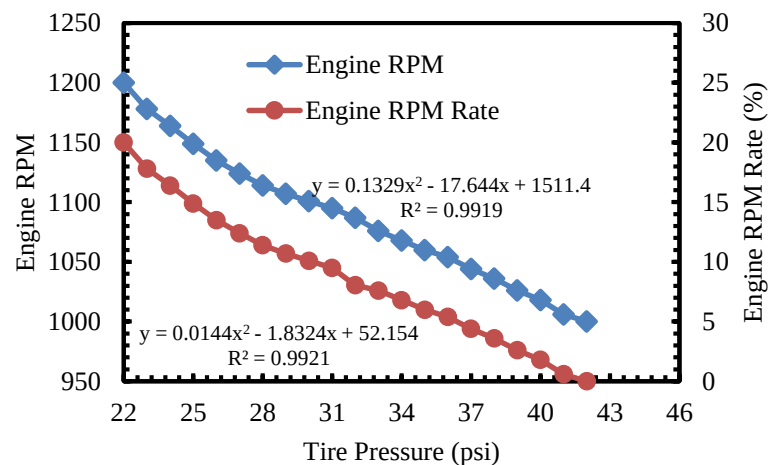
It is widely known that tire pressure plays an important role in determining vehicle handling and passenger comfort. Under- or overinflated tires can cause accidents and lead to excessive tire wear [39]. If the tire is operated at low pressure for an extended period, its life may be significantly shortened. Conversely, an over-inflated tire could cause damage when riding over irregularities in road surfaces, such as potholes and bumps, and reduce comfort marginally. It is well known that rolling resistance is inversely proportional to tire inflation pressure. Fuel consumption rises because of the greater rolling losses caused by the lower tire inflation pressure.

The effect of tire inflation pressure on fuel performance at front wheel total toe angles of 0.00°, 1.44° (toe-in), and -1.44° (toe-out) is investigated experimentally. The experiment used a Toyota Echo Plus-2ZZ-GE-02 light-duty vehicle. In the first case, fuel consumption is determined at the wheel front total toe angle of 0.00° with the variation of tire pressure from 42 psi to 22 psi on the basis of the data presented in Table D.5 of Appendix D when vehicle speed is maintained at 30 km/hr. The fuel consumption rate is calculated after 6 km of running. Other information like weather temperature (19.47 °C to 28.03 °C), humidity (64.0 % to 85.0 %), and engine surface temperature (77.17 °C to 82.17 °C) were also monitored. Like before, each test is performed three times over a period of time in order to determine an average value with the minimum error. In this case, a total of 21 tests are conducted, with the vehicle weighing 1090 kg for all testing (including the driver and one passenger).

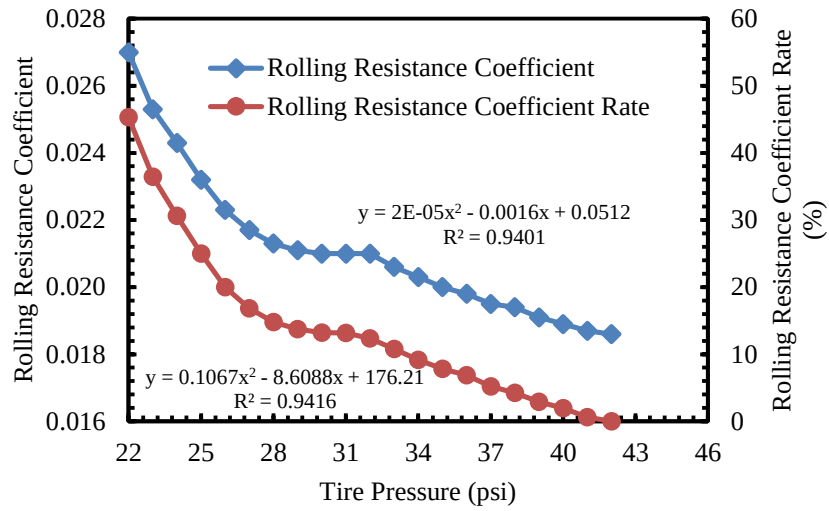
The effects of tire inflation pressure on engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, fuel consumption, and KPL at a 0° total toe angle are graphically illustrated in Figures 5.10(a)–(f). It has been observed that the rolling resistance varies non-linearly with the tire inflation pressure under steady-state

conditions (i.e., constant load, speed, and toe angle). In contrast, fuel consumption varies almost linearly. Interestingly, energy consumption increases almost non-linearly with tire inflation pressure (Figure 5.10(d)). The experimental results of the non-linear relationship agreed well with previously published results [117,205]. The testing findings show that decreasing tire pressure from the optimum inflation pressure (32 psi) led to an increase in rolling resistance and fuel consumption (Figures 5.10(c) and 5.10(e)). The rates of engine rpm, rolling resistance coefficient, and rolling resistance are increased up to 20.0%, 45.31%, and 45.31%, respectively, and the rates of energy consumption and fuel consumption are increased up to 45.31% and 39.22%, respectively. The rate of increase in rolling resistance is about 45.31%, and the fuel consumption rate is nearly 39.22% for the car. In contrast, the rolling resistance and fuel consumption are considerably reduced as the tire pressure increases from 33 to 42 psi (over-inflation). This is brought on by the reduction in the amount of road contact surface, which lowers rolling resistance.

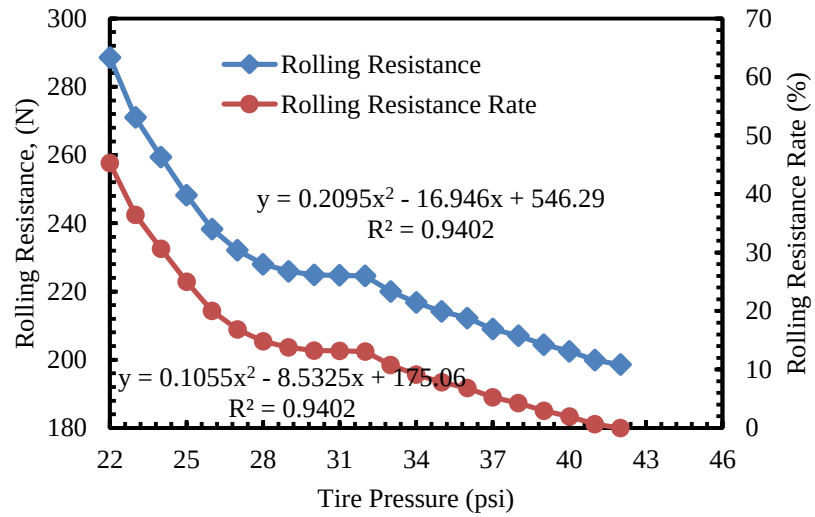
In Figure 5.10(f), it is shown that the KPL decreased considerably with the decrease in tire pressure. At 42 psi (tire over-inflation pressure), the car's KPL is about 19.60 km/L, and the KPL reduction rate is considered to be 0%. When the tire pressure is decreased to 22 psi, the car's KPL is reduced to 14.08 km/L, and the KPL reduction rate is about 28.17%. Therefore, fuel consumption increased by 39.22%. It is clear that the decrement in tire pressure from the standard value causes more fuel consumption, and the car traveled 5.52 km/L less at a front wheel total toe angle of 0.00 °.



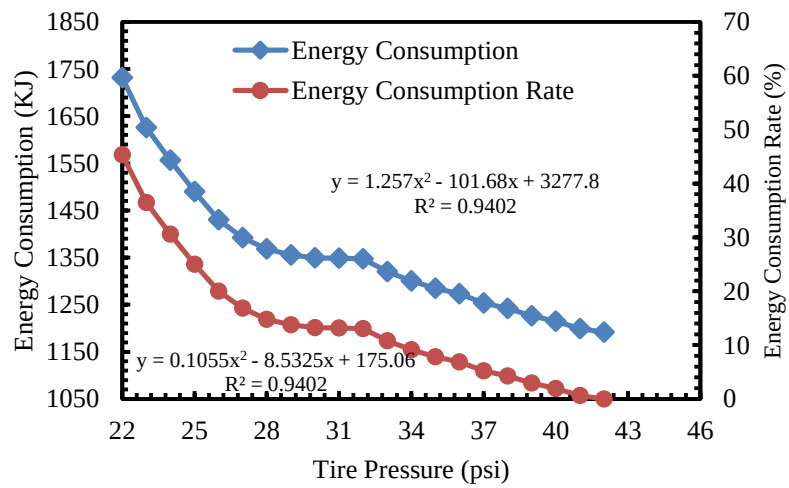
(a)



(b)



(c)



(d)

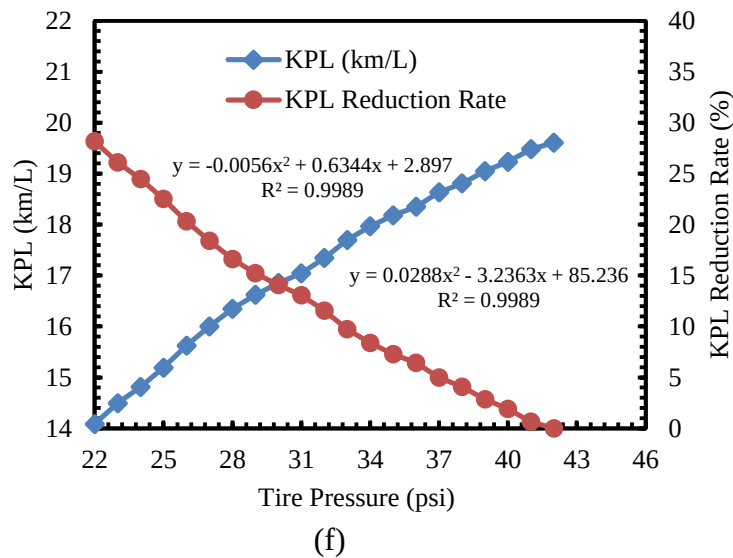
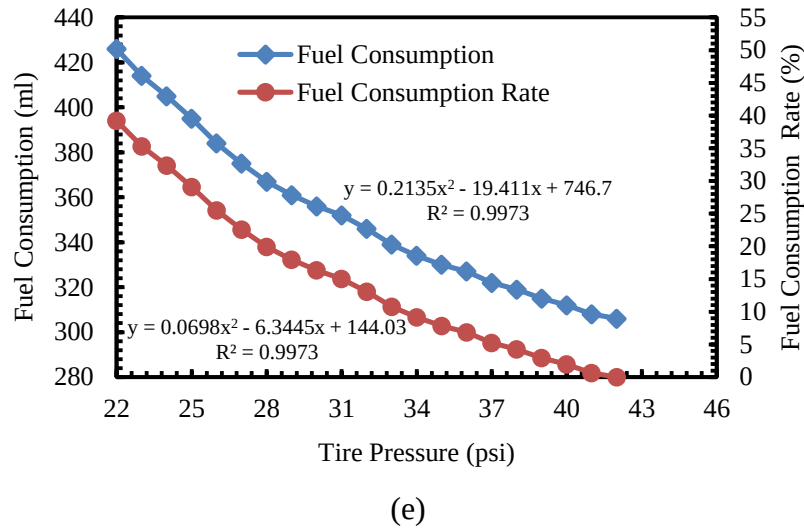


Figure 5.10 Variation of (a) Engine RPM and its rate (%), (b) Rolling resistance coefficient and its rate (%), (c) Rolling resistance (N) and its rate (%), (d) Energy consumption (Kj) and its rate (%), (e) Fuel consumption (ml) and its rate (%), and (f) KPL (Km/L) and KPL reduction rate (%) change with respect to tire pressure at 0.00° of front wheel total toe angle.

5.7.2 on Travelling Cost (at 0° Front Wheel Total Toe Angle)

The traveling cost per km in USD and BDT of the light vehicle as a function of the tire inflation pressure is presented in Table 5.10 and graphically shown in Figure 5.11. In Figure 5.11, it is shown that the traveling cost per km of the vehicle increased (from 0.0567\$ to 0.0790\$, or 4.5390Tk to 6.3190Tk) with the changes in tire pressure (from 42 psi to 22 psi), and the rate of traveling cost per km increased up to 39.22%. It is clear from the observations that a drop in tire pressure may cause a light vehicle's fuel consumption to rise, leading to an increase in the cost of traveling.

Table 5.10 Experimental data for 21 test runs for different tire pressure at 0.00° of front wheel total toe angles and travelling costs

Tire Pressure (psi)	Travelling Cost Per km $T_{C/km} = f_c \times f_p$		Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km (min)}}{T_{C/km (min)}}$
	USD	BDT	
22	0.0790	6.3190	39.22
23	0.0768	6.1410	35.29
24	0.0751	6.0075	32.35
25	0.0732	5.8592	29.08
26	0.0712	5.6960	25.49
27	0.0695	5.5625	22.55
28	0.0680	5.4438	19.93
29	0.0669	5.3548	17.97
30	0.0660	5.2807	16.34
31	0.0653	5.2213	15.03
32	0.0642	5.1323	13.07
33	0.0629	5.0285	10.78
34	0.0619	4.9543	9.15
35	0.0612	4.8950	7.84
36	0.0606	4.8505	6.86
37	0.0597	4.7763	5.23
38	0.0591	4.7318	4.25
39	0.0584	4.6725	2.94
40	0.0579	4.6280	1.96
41	0.0571	4.5687	0.65
42	0.0567	4.5390	0.00

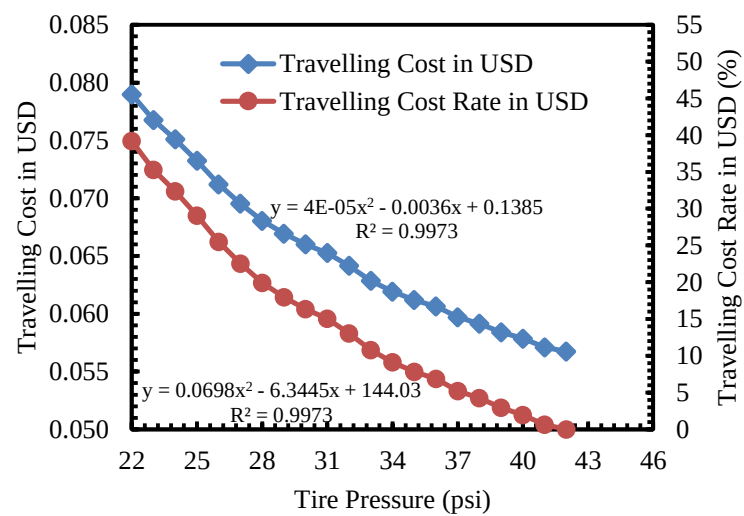


Figure 5.11 Change of travelling cost per Kilometer in USD and its rate in USD (%) with respect to tire pressure at 0.00° of front wheel total toe angle.

The correlation coefficient, $r_{t_{pfc}}$ is found to be -0.76 for tire pressure and fuel consumption, which is within the range of $0.75 < r_{t_{pfc}} < 1$ [201] (Table E.5 (Appendix E)). There is a very strong negative correlation between the tire pressure at 0.00° of the front wheel total toe angle and the fuel consumption of a light vehicle.

5.7.3 Correlation Develops Between Tire Pressure and Fuel Consumption at 0.00° of Front Wheel Total Toe Angle

The regression method has been used to create ten different correlation models based on experimental data. These models are shown in Table -5.11 from the Appendix in Tables G.3(a), G.3(b), and G.3(c). They are as follows: i) model 1- correlation between tire pressure at 0.00° of front wheel total toe angle and fuel consumption; ii) model 2- correlation between engine RPM and fuel consumption; iii) model 3- correlation between rolling resistance coefficient and fuel consumption; iv) model 4- correlation between rolling resistance and fuel consumption; v) model 5- correlation between energy consumption and fuel consumption; vi) model 6- correlation among tire pressure at 0.00° of front wheel total toe angle, engine RPM, and fuel consumption; vii) model 7- correlation among tire pressure at 0.00° of front wheel total toe angle, rolling resistance coefficient, and fuel consumption; viii) model 8- correlation among tire pressure at 0.00° of front wheel total toe angle, rolling resistance, and fuel consumption; ix) model 9- correlation among tire pressure at 0.00° of front wheel total toe angle, energy consumption and fuel consumption; x) model 10- correlation among tire pressure at 0.00° of front wheel total toe angle, engine RPM, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption.

The experimental data show that the proposed regression model has maintained a good correlation between the dependent variable (fuel consumption) and the independent variables (tire pressure at 0.00° of front wheel total toe angle, engine RPM, rolling resistance coefficient, rolling resistance, and energy consumption). Using the experimental data, it is found that the proposed regression model maintained a good correlation between the dependent variable (fuel consumption) and the independent variables (tire pressure at 0.00° of front wheel total toe angle, engine RPM, rolling resistance coefficient, rolling resistance, and energy consumption).

Table 5.11 Different simple and multiple correlation regression models for tire pressure variation at 0.00° of front wheel total toe angle.

Model No.	Form of Equation	Equation constants		Multiple R	R Square	Adjusted R Square	Stand. Error	P-value
		m _i	k _i					
1	$(F_c)_{1,1} = m_1 t_p + k_1$	m ₁ = -5.7441	535.8606	0.9791	0.9586	0.9565	7.5899	1.30883E-14
2	$(F_c)_{2,1} = m_1 R/M + k_1$	m ₁ = 0.6321	-335.5370	0.9918	0.9836	0.9828	4.7721	1.91796E-18
3	$(F_c)_{3,1} = m_1 \mu_{RRC} + k_1$	m ₁ = 15922.7802	15.3187	0.9732	0.9472	0.9444	8.5773	1.34394E-13
4	$(F_c)_{4,1} = m_1 F_R + k_1$	m ₁ = 1.4909	14.7859	0.9736	0.9479	0.9452	8.5172	1.17552E-13
5	$(F_c)_{5,1} = m_1 E_c + k_1$	m ₁ = 0.2484	14.7859	0.9736	0.9479	0.9452	8.5173	1.17556E-13
6	$(F_c)_{6,1} = m_1 t_p + m_2^* R/M + k_1$	m ₁ = 2.3207; m ₂ = 0.8824	-682.1026	0.9929	0.9858	0.9843	4.5574	1.41141E-05
7	$(F_c)_{7,1} = m_1 t_p + m_2^* \mu_{RRC} + k_1$	m ₁ = -3.2011; m ₂ = 7684.7951	291.9704	0.9957	0.9914	0.9904	3.5482	1.43774E-07
8	$(F_c)_{8,1} = m_1 t_p + m_2^* F_R + k_1$	m ₁ = -3.1883; m ₂ = 0.7222	290.6916	0.9956	0.9913	0.9904	3.5619	1.54253E-07
9	$(F_c)_{9,1} = m_1 t_p + m_2^* E_c + k_1$	m ₁ = -3.1883; m ₂ = 0.1203	290.6920	0.9956	0.9913	0.9904	3.5619	1.54258E-07
10	$(F_c)_{10,1} = m_1 t_p + m_2 R/M + m_3 \mu_{RRC} + m_4 F_R + m_5 E_c + k_1$	m ₁ = -12.0340; m ₂ = -1.3896; m ₃ = -19361.8167; m ₄ = 66404.1320; m ₅ = -11066.7526	1839.9097	0.9978	0.9956	0.9942	2.7718	0.061171545
$F_c = 1839.909726 - 12.03407166^*(t_p) - 1.389646154^*(R/M) - 19361.81679^*(\mu_{RRC}) + 66404.13208^*(F_R) - 11066.75262^*(E_c)$								

In this case, 21 tests are performed per regression model. The R square, adjusted R square, and p -values of the corresponding regression model are presented in Table 5.11. In model 1, it is seen that the values of R square and adjusted R^2 are found to be 0.9586 and 0.9565, respectively; these values are very close, and it is also seen that the p -values are less than 0.05, by which it can be claimed that there is a perfect relationship between tire pressure and fuel consumption.

Similarly, models 2–10 show a perfect relationship between the dependent and independent variables because the correlation R -square values for models tend towards 1, as shown in Table 5.11.

5.8 EFFECT OF TIRE PRESSURE

5.8.1 on Fuel Consumption (at Front Wheel Total Toe-in Angle, 1.44°)

In this case, the fuel consumption is measured using a change of tire pressure (from 42 psi to 22 psi) at a front wheel total toe-in angle of 1.44° (as shown in Table D.6 in Appendix D). The experimental results show a similar trend as graphically discussed in the preceding section, that is, the engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption increased as the tire pressure decreased from over inflation to under inflation under constant vehicle load and speed. Due to a similar pattern, the graphical description of rolling resistance, fuel consumption, and other associated characteristics is therefore omitted from this presentation; instead, the results are described with the help of Tabulated data. Once again, the vehicle speed is maintained at 30 km/h. The fuel consumption rate is determined after a running distance of 6 km. The testing procedure, number of tests, and vehicle's weight remained the same as in Section 5.7.

In Table D.6, it is found that the engine rpm, rolling resistance coefficient, and rolling resistance changed from 1093 rev/min to 1373 rev/min, 0.0210 to 0.0388, and 224.38 N to 415.34 N, respectively, with the changes in tire pressure from 42 psi to 22 psi. The rates of engine rpm, rolling resistance coefficient, and rolling resistance increased up to 25.62%, 85.10%, and 85.10%, respectively, when tire pressure dropped from 42 psi to 22 psi at a 1.44° front wheel total toe-in angle.

It is well known that rolling resistance is inversely proportional to tire inflation pressure. The decreased tire inflation pressure increases rolling loss, resulting in

increased fuel consumption. Changing the tire pressure from 42 psi to 22 psi at a front wheel total toe-in angle of 1.44° changed the amount of energy and fuel used from 1346.31 KJ to 2492.04 KJ and 350 ml to 491 ml, respectively. The rates of energy consumption and fuel consumption increased by up to 85.10% and 40.29%, respectively.

In this case, the KPL decreased considerably with the decrease in tire pressure at 1.44° of the front wheel's total toe-in angle. At 42 psi (tire over-inflation pressure), the car's KPL is about 17.14 km/L, and the KPL reduction rate is considered to be 0%. When the tire pressure decreases to 22 psi (under inflation), the car's KPL is reduced to 12.21 km/L, and the KPL reduction rate is about 28.72%. Therefore, fuel consumption increased by 40.29%. It is found that due to the decrease in tire pressure variation from over-inflation to under-inflation, the car travels 4.93 km/L less (from 17.14 km/L to 12.21 km/L).

5.8.2 on Travelling Cost (at Front Wheel Total Toe-in Angle, 1.44°)

The traveling cost per km in USD and BDT of the light vehicle as a function of the tire pressure at 1.44° of front wheel total toe-in angle is presented in Table 5.12 and Figure 5.12. From Table 5.12 and Figure 5.12, it is shown that the traveling cost per km increased from 0.0649\$ to 0.0910\$, or 5.1917Tk to 7.2832Tk, with the changes in tire pressure from 42 psi to 22 psi, and the rate of traveling cost per km increased by up to 40.29%. As was mentioned in the section above, it is evident that a drop in tire pressure might cause a light vehicle's fuel consumption to rise, leading to an increase in the cost of travel.

Table 5.12 Experimental data for 21 test runs for different tire pressure at 1.44° of front wheel total toe-in angles and travelling costs

Tire Pressure (psi)	Travelling Cost Per km $T_{C/km} = f_c \times f_p$		Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km (min)}}{T_{C/km (min)}}$
	USD	BDT	
22	0.0910	7.2832	40.29
23	0.0884	7.0755	36.29
24	0.0866	6.9272	33.43
25	0.0846	6.7640	30.29
26	0.0823	6.5860	26.86
27	0.0805	6.4377	24.00
28	0.0792	6.3338	22.00
29	0.0781	6.2448	20.29
30	0.0769	6.1558	18.57
31	0.0760	6.0817	17.14
32	0.0747	5.9778	15.14
33	0.0732	5.8592	12.86
34	0.0721	5.7702	11.14
35	0.0712	5.6960	9.71
36	0.0705	5.6367	8.57
37	0.0695	5.5625	7.14
38	0.0688	5.5032	6.00
39	0.0680	5.4438	4.86
40	0.0671	5.3697	3.43
41	0.0660	5.2807	1.71
42	0.0649	5.1917	0.00

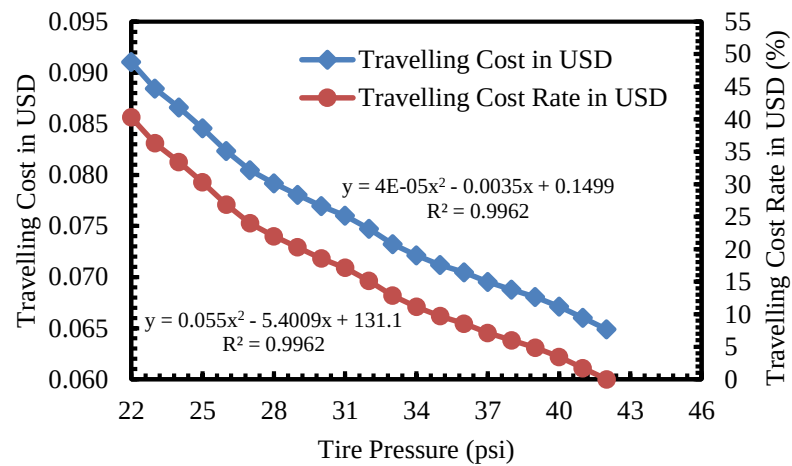


Figure 5.12 Change of travelling cost per Kilometer in USD and its rate in USD (%) with respect to tire pressure at 1.44° of front wheel total toe-in angle.

Similarly, the experimental result has been verified by the Pearson's correlation coefficient presented in Equation 3.14. The Pearson's correlation coefficient relationship between tire pressure and fuel consumption is presented in Table E.6

(Appendix E). The Pearson's correlation coefficient, $r_{t_{pfc}}$ is found to be -0.76 for tire pressure and fuel consumption, which is within the range of $0.75 < r_{t_{pfc}} < 1$ [201], providing a very strong negative correlation between the tire pressure and fuel consumption of a light vehicle at 1.44° of front wheel total toe-in angle.

5.8.3 Correlation Develops Between Tire Pressure and Fuel Consumption at 1.44° of Front Wheel Total Toe-in Angle

Likewise, based on experimental data as shown in Table 5.13 from Appendix Tables G.4(a), G.4(b), and G.4(c), ten different correlation models are developed using the regression method, which are: i) model 1- correlation between tire pressure at 1.44° of front wheel total toe-in angle and fuel consumption; ii) model 2- correlation between engine RPM and fuel consumption; iii) model 3- correlation between rolling resistance coefficient and fuel consumption; iv) model 4- correlation between rolling resistance and fuel consumption; v) model 5- correlation between energy consumption and fuel consumption; vi) model 6- correlation among tire pressure at 1.44° of front wheel total toe-in angle, engine RPM and fuel consumption; vii) model 7- correlation among tire pressure at 1.44° of front wheel total toe-in angle, rolling resistance coefficient and fuel consumption; viii) model 8- correlation among tire pressure at 1.44° of front wheel total toe-in angle, rolling resistance and fuel consumption; ix) model 9- correlation among tire pressure at 1.44° of front wheel total toe-in angle, energy consumption, and fuel consumption; x) model 10- correlation among tire pressure at 1.44° of front wheel total toe-in angle, engine RPM, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption.

The R square, adjusted R square, and p -values of the corresponding regression model are presented in Table 5.13. The values of R square and adjusted R^2 are 0.9719 and 0.9704, respectively; these values are very close, and it is also seen that the p -values are less than 0.05, by which it can be claimed that there is a perfect relationship between tire pressure and fuel consumption.

Similarly, models 2–10 show a perfect relationship between the dependent and independent variables because the correlation R -square values for models tend towards 1. According to these findings, the proposed regression models effectively maintained the positive correlation between the experimentally measured independent variables

(tire pressure, engine RPM, rolling resistance, rolling resistance coefficient, and energy consumption) and the dependent variable (fuel consumption).

Table 5.13 Different simple and multiple correlation regression models for tire pressure variation at 1.44° of front wheel total toe-in angle.

Model No.	Form of Equation	Equation constants		Multiple R	R Square	Adjusted R Square	Stand. Error	P-value
		m _i	k _i					
1	$(F_c)_{1,1} = m_1 t_p + k_1$	m ₁ = -6.5727	618.6129	0.9858	0.9719	0.9704	7.1145	3.34933E-16
2	$(F_c)_{2,1} = m_1 R/M + k_1$	m ₁ = 0.5154	-201.4998	0.9834	0.9671	0.9653	7.6952	1.49069E-15
3	$(F_c)_{3,1} = m_1 \mu_{RRC} + k_1$	m ₁ = 7527.0348	213.1562	0.9733	0.9473	0.9445	9.7418	1.32847E-13
4	$(F_c)_{4,1} = m_1 F_R + k_1$	m ₁ = 0.7035	213.2286	0.9734	0.9475	0.9447	9.7226	1.27946E-13
5	$(F_c)_{5,1} = m_1 E_c + k_1$	m ₁ = 0.1172	213.2286	0.9734	0.9475	0.9447	9.7226	1.27946E-13
6	$(F_c)_{6,1} = m_1 t_p + m_2^* R/M + k_1$	m ₁ = -3.5184; m ₂ = 0.2554	218.6554	0.9998	0.9996	0.9995	0.8708	4.36954E-18
7	$(F_c)_{7,1} = m_1 t_p + m_2^* \mu_{RRC} + k_1$	m ₁ = -3.9355; m ₂ = 3316.9942	448.2326	0.9996	0.9993	0.9993	1.0690	1.75511E-16
8	$(F_c)_{8,1} = m_1 t_p + m_2^* F_R + k_1$	m ₁ = -3.9318; m ₂ = 0.3103	448.0513	0.9997	0.9994	0.9993	1.0521	1.31708E-16
9	$(F_c)_{9,1} = m_1 t_p + m_2^* E_c + k_1$	m ₁ = -3.9318; m ₂ = 0.0517	448.0514	0.9997	0.9994	0.9993	1.0521	1.31709E-16
10	$(F_c)_{10,1} = m_1 t_p + m_2 R/M + m_3 \mu_{RRC} + m_4 F_R + m_5 E_c + k_1$	m ₁ = -3.6282; m ₂ = 0.1840; m ₃ = -330.0107; m ₄ = -6124.3377; m ₅ = 1020.7426	282.4920	0.9998	0.9996	0.9995	0.8385	0.340870857
	$F_c = 282.4920259 - 3.628299445*(t_p) + 0.184007956*(R/M) - 330.0107558*(\mu_{RRC}) - 6124.337761*(F_R) + 1020.742659*(E_c)$							

5.9 EFFECT OF TIRE PRESSURE AT FRONT WHEEL TOTAL TOE-OUT ANGLE, -1.44°

In such a case, the fuel consumption is calculated using a change of tire pressure (from 42 psi to 22 psi) at a vehicle speed of 30 km/hr with the wheel front total toe-out angle at -1.44° on the basis of Table D.7 shown in Appendix D. Following the variation in tire pressure at a -1.44° toe-out angle, the experimental results of engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption revealed a similar trend, as illustrated graphically in the preceding section. Therefore, the graphical description of rolling resistance, fuel consumption, and other associated characteristics is omitted from this presentation; instead, the results are described with the help of Tabulated data. During the experiments, other conditions remained the same as in the previous test.

On condition of the front wheel's total toe-out angle of -1.44°, it is discovered that the changes in tire pressure from 42 psi to 22 psi caused changes in the engine rpm, rolling resistance coefficient, and rolling resistance (from 1097 rev/min to 1428 rev/min, 0.0210 to 0.0424, and 224.51 N to 453.56 N, respectively). The rates of engine rpm, rolling resistance coefficient, and rolling resistance increased by up to 30.17%, 102.02%, and 102.02%, respectively.

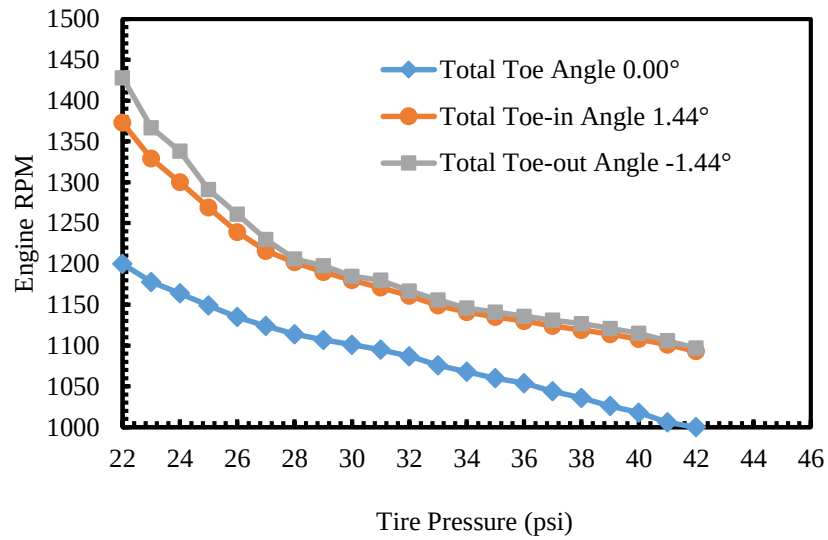
As shown in Table D.7, the energy consumption and fuel consumption increased from 1347.09 kJ to 2721.38 kJ and from 353 ml to 508 ml as the tire pressure dropped from 42 psi to 22 psi at a front wheel total toe-out angle of -1.44°, and the increasing rates are up to 102.02% and 43.91%, respectively. It is also discovered that when the tire pressure is decreased to 22 psi (tire under-inflation pressure), the car's KPL is reduced to 11.81 km/L, and the KPL reduction rate is about 30.51%.

Also, the experimental result is verified by Pearson's correlation coefficient, as presented in Equation 3.14. The Pearson's correlation coefficient relationship between tire pressure and fuel consumption is presented in Table E.7 (Appendix E). The correlation coefficient, $r_{t_{pfc}}$ is found to be -0.76 for tire pressure of the front wheel total toe-out angle at -1.44° and fuel consumption, which is within the range of $0.75 < r_{t_{pfc}} < 1$ [201]. It has been proven to be a very strong negative correlation between the tire pressure at -1.44° of the front wheel's total toe-out angle and the fuel consumption of a light vehicle.

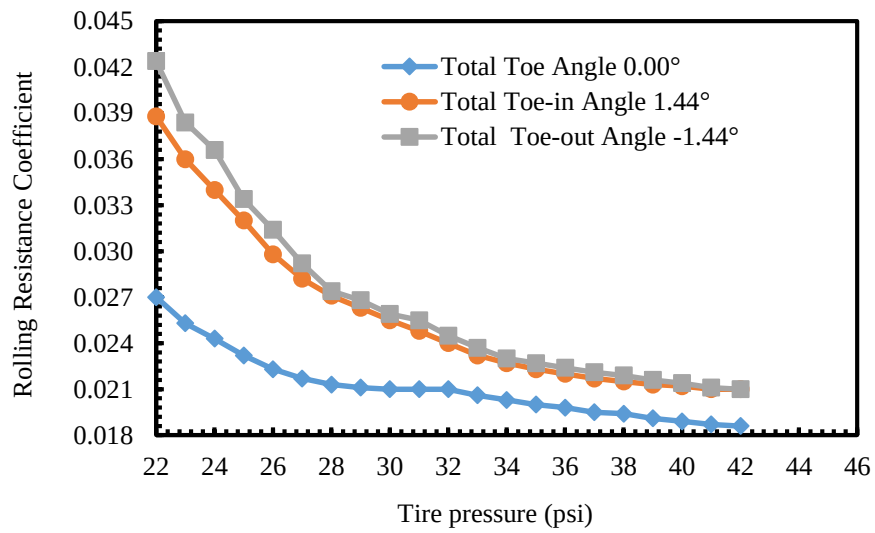
5.10 COMPARATIVE ANALYSIS BASED ON EFFECT OF TIRE PRESSURE AT FRONT WHEEL TOTAL TOE ANGLE OF 0°, 1.44° (TOE-IN), AND -1.44° (TOE-OUT)

The comparison of the engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption in Figures 5.13(a)–5.13(e) revealed some dependence on the tire pressure. The fuel consumption rate is slightly greater for the front wheel total toe-out angle at -1.44° compared to toe-in at 1.44° with a change in tire pressure in Figure 5.13(e). The result also exhibits that the front wheel total toe-out angle at -1.44° produces more rolling resistance compared to the front wheel total toe-in angle at 1.44°. Additionally, the rolling resistance generated by the tire is significantly reduced at 0° of the front wheel's total toe angle. However, the rolling resistance significantly reduces as tire pressure rises from under-inflated to over-inflated.

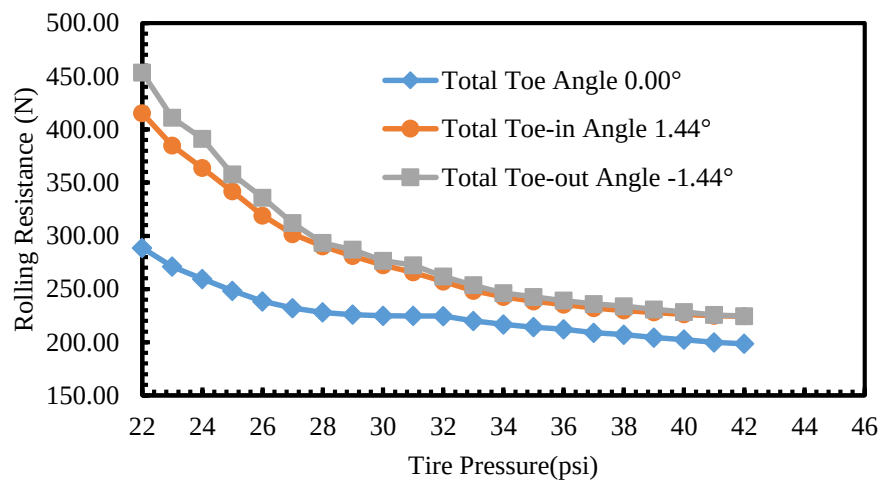
Interestingly, the deviation of rolling resistance at under-inflation is greater compared to over-inflation, as shown in Figure 5.13(c). This is because the tire-road contact surface increases as the wheel toe angle increases (toe-in or toe-out). In comparison to the front wheel total toe-out angle of -1.44°, the fuel consumption rate of 4.69% is higher at the front wheel total toe angle of 0.00°, and 3.62% is higher at the front wheel total toe-in angle of 1.44°. The deviation is most likely due to increased pressure on the rubber through the increased grip friction of the driven tires. In general, a vehicle's excessive toe-in can make the tires scrub or drag on the road surface, which raises rolling resistance. Compared to the front wheel total toe angle at 0.00°, the KPL reduction rate is about 2.34% higher for the front wheel total toe-out angle at -1.44° and 1.79% higher for the front wheel total toe-in angle at 1.44° when the pressure drops from 42 psi to 22 psi. Based on the experimental results, it is concluded that there is no direct combined relationship among tire inflation pressure, wheel toe angle, and fuel consumption. Therefore, it can be concluded that fuel consumption is fairly independent of tire inflation pressure and wheel misalignment (toe angle).



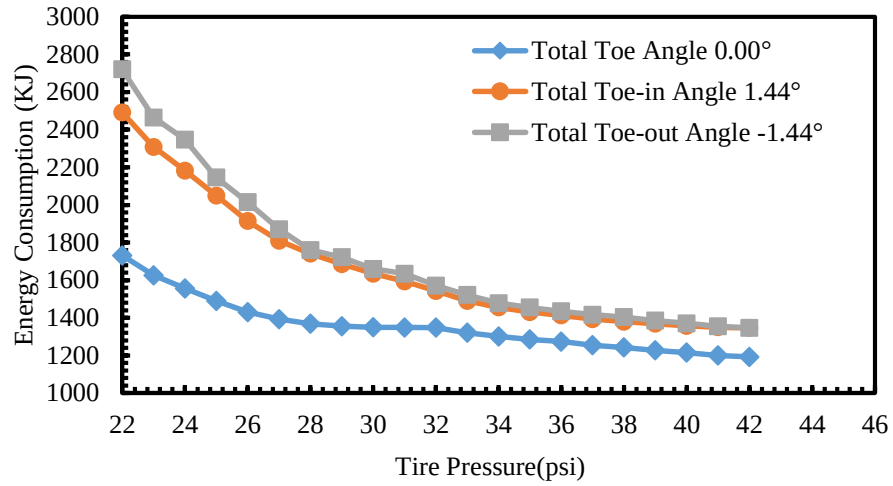
(a)



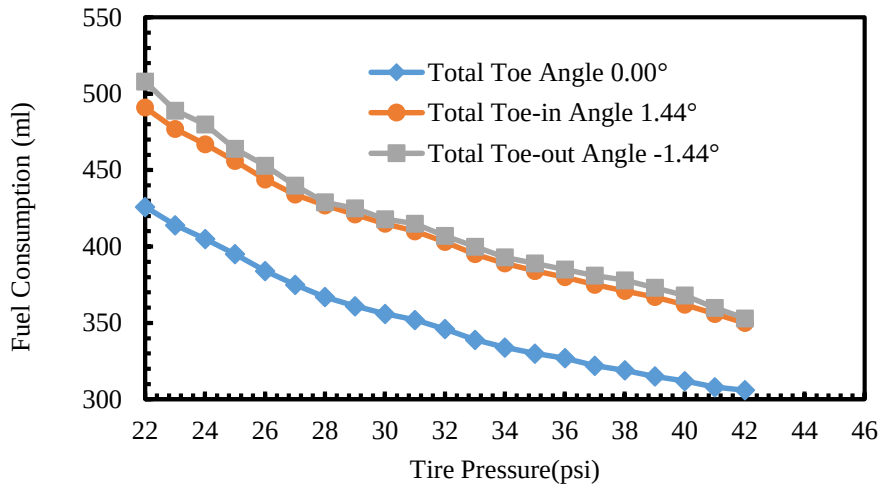
(b)



(c)



(d)



(e)

Figure 5.13 Comparison of (a) Engine RPM, (b) Rolling resistance coefficient, (c) Rolling resistance (N), (d) Energy consumption (Kj), and (e) Fuel consumption (ml) with respect to tire pressure at front wheel total toe angle 0.00°, at front wheel total toe-in angle 1.44°, and at front wheel total toe-out angle -1.44°, respectively.

5.10.1 on Travelling Costs

A comparison of the traveling cost per km in USD of the light vehicle as a function of the tire pressure at the front wheel total toe angle of 0.00°, at the front wheel total toe-in angle of 1.44°, and at the front wheel total toe-out angle of -1.44° is presented in Figure 5.14 (from Tables 5.10 and 5.12 in Appendix F.3). It is observed that the rate of the traveling cost is slightly higher at -1.44° of front wheel total toe-out angle. In comparison to the front wheel total toe angle of 0.00°, the rate of travel cost is 4.69%

higher at the front wheel total toe-out angle of -1.44° and 3.62% higher at the front wheel total toe-in angle of 1.44° .

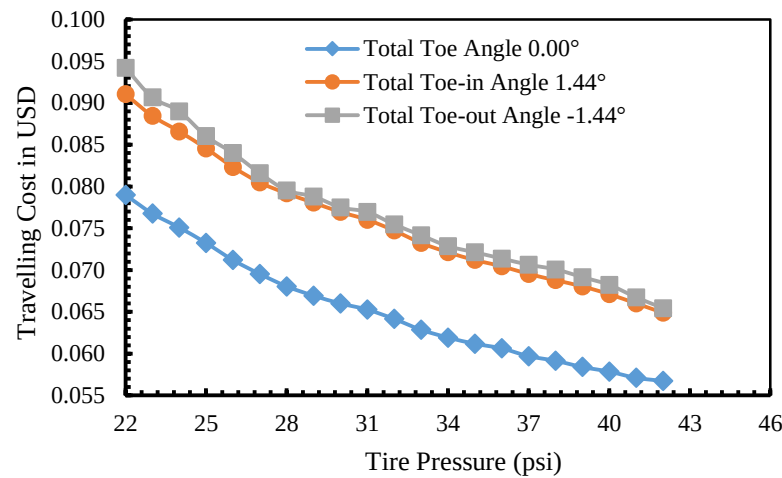


Figure 5.14 Comparison of travelling cost in USD with respect to tire pressure at front wheel total toe angle 0.00° , at front wheel total toe-in angle 1.44° , and at front wheel total toe-out angle -1.44° .

5.11 EFFECT OF VEHICLE LOAD

5.11.1 on Fuel Consumption (at 0.00° of Front Wheel Total Toe Angle)

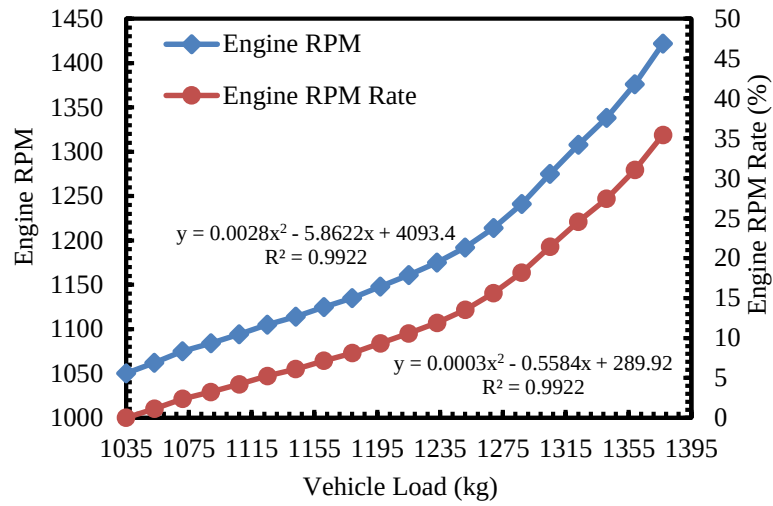
According to Madhusudhana and Nagaraja [40], vehicle load influences vehicle handling and accidents, as well as excessive tire wear and fuel consumption. The tire's life may be significantly prolonged if it is run at a low load for an extended length of time. An over-loaded tire, on the other hand, may cause damage while riding over abnormalities in the road surface, such as potholes and bumps, and comfort is also marginally diminished. The relationship between rolling resistance and vehicle load is widely understood. A higher vehicle load causes a higher rolling loss, which raises fuel consumption.

In such a case, a Toyota Echo Plus-2ZZ-GE-02 light-duty vehicle is used to investigate the effect of vehicle load on fuel performance. In the first case, the fuel consumption is measured based on the variation of vehicle load at 0.00° of front wheel total toe angle, as shown in Table D.8 in Appendix D. During the tests, the tire pressure is kept constant at 32 psi (the recommended standard tire pressure for a 175/70R14 tire), and the vehicle speed is maintained at 30 km/h. For each test run, the fuel consumption rate is calculated after a travel distance of 6 km. Other data associated with the test, such as weather temperature (24.53°C to 35.67°C), humidity (46.67% to 90.00%), and engine surface

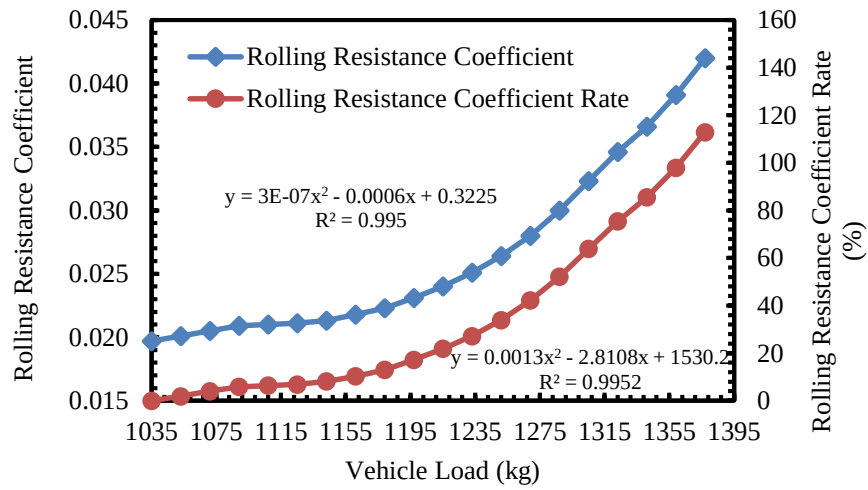
temperature (79.93 °C to 82.37 °C), are also recorded during the experiment. A total of 20 tests are conducted in this case, and each test is repeated three times over a period of time to calculate an average value and minimize error in the results.

The effects of vehicle load on engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, fuel consumption, and KPL at a 0° toe angle are graphically illustrated in Figures 5.15(a)–(f). It has been observed that the rolling resistance varies non-linearly with the vehicle load under steady-state conditions (i.e., constant tire inflation pressure, speed, and toe angle), while the fuel consumption varies almost linearly. Interestingly, energy consumption increases almost non-linearly with vehicle load (Figure 5.15(d)). The experimental results of the non-linear relationship agreed well with previously published results [206]. The experimental findings show that increasing the vehicle load from 1035 kg to 1377 kg led to an increase in rolling resistance and fuel consumption (Figures 5.15(c) and 5.15(e)). The rates of engine rpm, rolling resistance coefficient, and rolling resistance are increased up to 35.43%, 112.88%, and 112.88%, respectively, and the rates of energy consumption and fuel consumption are increased up to 112.88% and 55.69%, respectively. The rate of increase in rolling resistance is about 112.88%, and the fuel consumption rate is nearly 55.69% for the car. In contrast, the rolling resistance and fuel consumption considerably increased as the vehicle load increased from 1035 kg to 1377 kg. This is brought on by the increasing road contact surface, which creates higher rolling resistance.

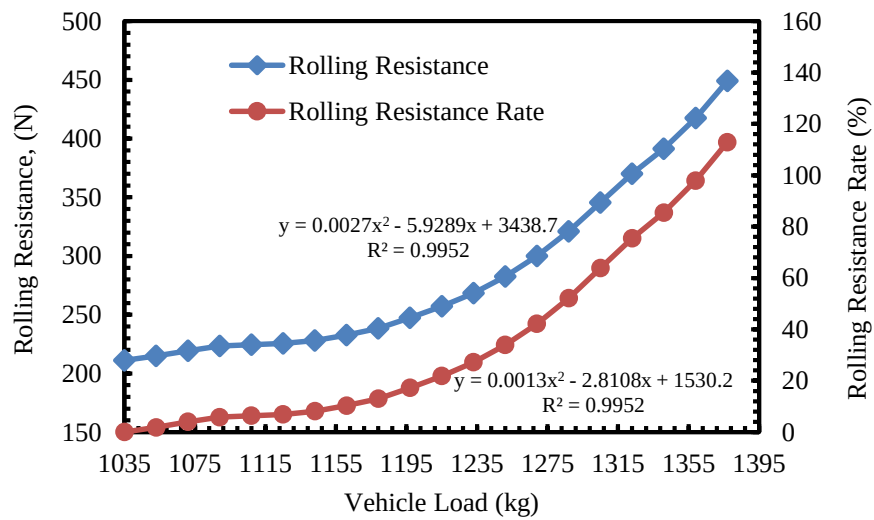
Figure 5.15(f) shows that the KPL decreased considerably with the increase in vehicle load. At 1035 kg vehicle load, the car's KPL is about 18.46 km/L, and the KPL reduction rate is considered to be 0%. When the vehicle load is increased to 1377 kg, the car's KPL is reduced to 11.85 km/L, and the KPL reduction rate is about 35.77%. Therefore, fuel consumption increased by 55.69%. It is clear that the increment in vehicle load from the standard load causes more fuel consumption, and the car traveled 6.61 km/L less at a front wheel total toe angle of 0.00°.



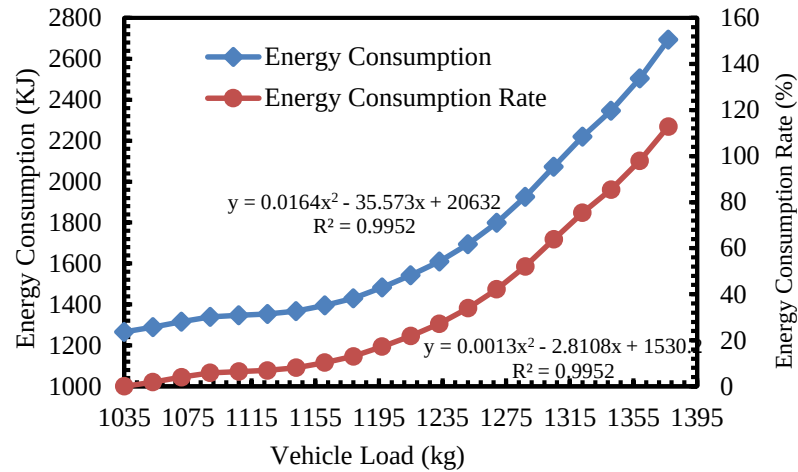
(a)



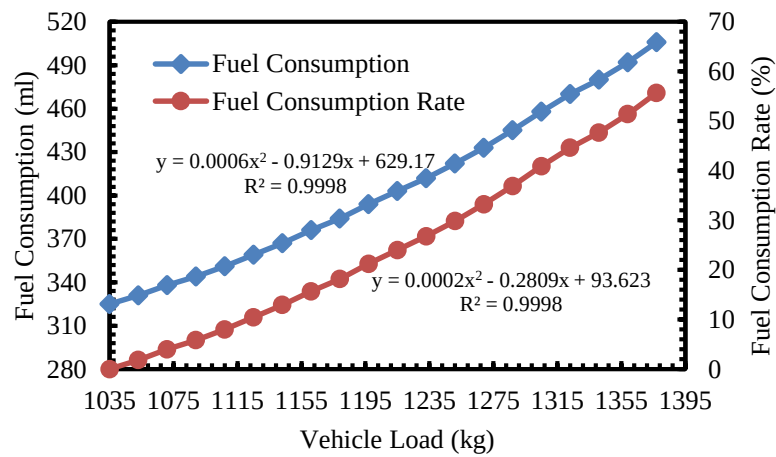
(b)



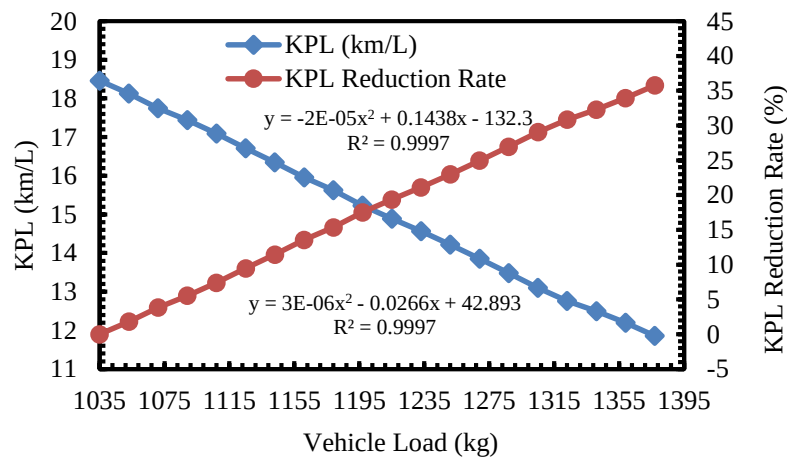
(c)



(d)



(e)



(f)

Figure 5.15 Variation of (a) Engine RPM and its rate (%), (b) Rolling resistance coefficient and its rate (%), (c) Rolling resistance (N) and its rate (%), (d) Energy consumption (Kj) and its rate (%), (e) Fuel consumption (ml) and fuel consumption rate (%), and (f) KPL (Km/L) and KPL reduction rate (%) change with respect to vehicle load at 0.00° of front wheel total toe angle.

5.11.2 on Travelling Cost (at 0.00° of Front Wheel Total Toe Angle)

The traveling cost per km in USD and BDT of the light vehicle as a function of the vehicle load is presented in Table 5.14 and graphically shown in Figure 5.16. As the vehicle's weight changed (1035kg to 1377kg), the traveling cost per kilometer increased (from 0.0603\$ to 0.0938\$, or 4.8208Tk to 7.5057Tk), and the rate of rise reached 55.69%, as shown in Figure 5.16. It is clear that, as stated in the section above, an increase in vehicle load can cause a light vehicle's fuel consumption to rise, leading to a rise in the cost of travel.

Table 5.14 Experimental data for 20 test runs for different vehicle load conditions at 0.00° of front wheel total toe angles and travelling costs

Vehicle Load (kg)	Travelling Cost Per km $T_{C/km} = f_c \times f_p$		Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km (min)}}{T_{C/km (min)}}$
	USD	BDT	
1035	0.0603	4.8208	0.00
1053	0.0614	4.9098	1.85
1071	0.0627	5.0137	4.00
1089	0.0638	5.1027	5.85
1107	0.0651	5.2065	8.00
1125	0.0666	5.3252	10.46
1143	0.0680	5.4438	12.92
1161	0.0697	5.5773	15.69
1179	0.0712	5.6960	18.15
1197	0.0731	5.8443	21.23
1215	0.0747	5.9778	24.00
1233	0.0764	6.1113	26.77
1251	0.0782	6.2597	29.85
1269	0.0803	6.4228	33.23
1287	0.0825	6.6008	36.92
1305	0.0849	6.7937	40.92
1323	0.0871	6.9717	44.62
1341	0.0890	7.1200	47.69
1359	0.0912	7.2980	51.38
1377	0.0938	7.5057	55.69

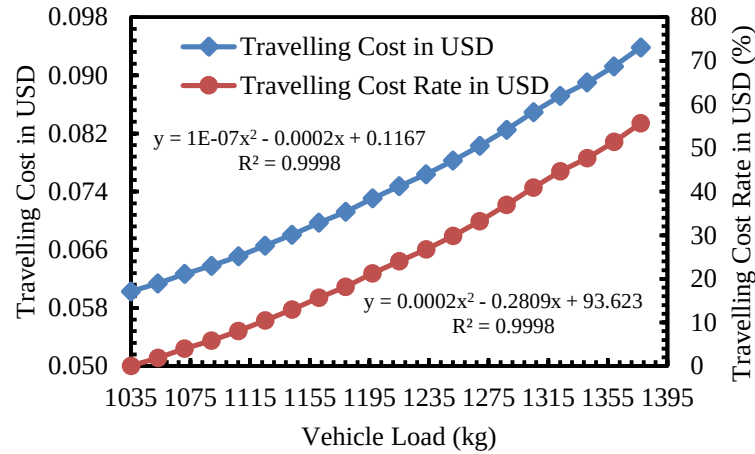


Figure 5.16 Change of travelling cost per Kilometer in USD and its rate in USD (%) with respect to vehicle load at 0.00° of front wheel total toe angle.

The result is verified by Pearson's correlation coefficient, presented in equation 3.14. The Pearson's correlation coefficient relationship between the vehicle load and fuel consumption is presented in Table E.8 (Appendix E). In this case, The Pearson's correlation coefficient is found to be 0.99 for vehicle load and fuel consumption at 0.00° of front wheel total toe angle, which is within the range of $0.75 < r_{vlf_c} < 1$ [201], providing a very strong positive correlation between the vehicle load and fuel consumption of light vehicles at 0.00° of front wheel total toe angle.

5.11.3 Correlation Develops Between Vehicle Load and Fuel Consumption at 0.00° of Front Wheel Total Toe Angle

In the same way, ten different correlation models are developed using the regression method based on the experimental data shown in Table 5.15 from Appendix Tables G.5(a), G.5(b), and G.5(c), which are: i) model 1- correlation between vehicle load at 0.00° of front wheel total toe angle and fuel consumption; ii) model 2- correlation between engine RPM and fuel consumption; iii) model 3- correlation between rolling resistance coefficient and fuel consumption; iv) model 4- correlation between rolling resistance and fuel consumption; v) model 5- correlation between energy consumption and fuel consumption; vi) model 6- correlation among vehicle load at 0.00° of front wheel total toe angle, engine RPM and fuel consumption; vii) model 7- correlation among vehicle load at 0.00° of front wheel total toe angle, rolling resistance coefficient, and fuel consumption; viii) model 8- correlation among vehicle load at 0.00° of front wheel total toe angle, rolling resistance, and fuel consumption; ix) model 9- correlation among vehicle load at 0.00° of front wheel total toe angle, energy consumption, and

fuel consumption; x) model 10- correlation among vehicle load at 0.00° of front wheel total toe angle, engine RPM, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption.

Table 5.15 Different simple and multiple correlation regression models for vehicle load condition at 0.00° of front wheel total toe angle.

Model No.	Form of Equation	Equation constants		Multiple R	R Square	Adjusted R Square	Stand. Error	P-value
		m _i	k _i					
1	$(F_c)_{1,1} = m_1 v_l + k_1$	m ₁ = 0.5296	-234.2669	0.9944	0.9890	0.9884	6.1032	4.29164E-19
2	$(F_c)_{2,1} = m_1 R/M + k_1$	m ₁ = 0.5107	-200.6191	0.9840	0.9683	0.9665	10.3687	6.02217E-15
3	$(F_c)_{3,1} = m_1 \mu_{RRC} + k_1$	m ₁ = 7860.5184	196.2355	0.9619	0.9253	0.9212	15.9187	1.37996E-11
4	$(F_c)_{4,1} = m_1 F_R + k_1$	m ₁ = 0.7358	196.0164	0.9622	0.9259	0.9218	15.8533	1.28104E-11
5	$(F_c)_{5,1} = m_1 E_c + k_1$	m ₁ = 0.1226	196.0164	0.9622	0.9259	0.9218	15.8533	1.28104E-11
6	$(F_c)_{6,1} = m_1 v_l + m_2^* R/M + k_1$	m ₁ = 0.3429; m ₂ = 0.1889	-233.0405	0.9993	0.9987	0.9986	2.1203	1.93663E-09
7	$(F_c)_{7,1} = m_1 v_l + m_2^* \mu_{RRC} + k_1$	m ₁ = 0.3924; m ₂ = 2265.0450	-128.8402	0.9997	0.9995	0.9994	1.3119	5.34884E-13
8	$(F_c)_{8,1} = m_1 v_l + m_2^* F_R + k_1$	m ₁ = 0.3919; m ₂ = 0.2127	-128.4316	0.9997	0.9995	0.9994	1.2807	3.549E-13
9	$(F_c)_{9,1} = m_1 v_l + m_2^* E_c + k_1$	m ₁ = 0.3919; m ₂ = 0.0354	-128.4316	0.9997	0.9995	0.9994	1.2807	3.54892E-13
10	$(F_c)_{10,1} = m_1 v_l + m_2 R/M + m_3 \mu_{RRC} + m_4 F_R + m_5 E_c + k_1$	m ₁ = 0.4283; m ₂ = -0.1217; m ₃ = 0; m ₄ = 0; m ₅ = 0.0570	-64.7813	0.9998	0.9996	0.8746	1.1021	#NUM!
$F_c = -64.78135307 + 0.428341236^*(v_l) - 0.121780069^*(R/M) + 0^*(\mu_{RRC}) + 0^*(F_R) + 0.057045263^*(E_c)$								

Model 1 in Table 5.15 shows that the values of R square and adjusted R^2 are 0.9890 and 0.9884, respectively; these values are very close, and it can also be seen that the p -values are less than 0.05, by which it can be claimed that there is a perfect relationship between the vehicle load and fuel consumption. Correspondingly, the values of R square and adjusted R^2 for model-2-10 are tending towards 1, which shows a perfect relationship between the dependent and independent variables. From these results, it can be said that the proposed regression model has maintained a good correlation between the dependent variable (fuel consumption) and the independent variable (vehicle load, engine RPM, rolling resistance, rolling resistance coefficient, and energy consumption) from the experimental data.

5.12 EFFECT OF VEHICLE LOAD

5.12.1 on Fuel Consumption (at Front Wheel Total Toe-in Angle, 1.44°)

Table D.9 in Appendix D shows the fuel consumption measured with the change of a vehicle load from 1035 kg to 1377 kg at a front wheel total toe-in angle of 1.44°. The experimental results show that the engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption increased as the vehicle load increased from 1035 kg to 1377 kg under constant speed and recommended tire pressure. The graphical representation of rolling resistance, fuel consumption, and other pertinent parameters are therefore omitted from this presentation due to a similar trend as discussed in the earlier section, and the results are instead expressed with tabulated data. Except for vehicle load, the same test conditions are used for the front wheel's total toe angle of 0.00°.

According to Table D.9, with changes in the vehicle load, the engine rpm, rolling resistance coefficient, and rolling resistance increased from 1120 rev/min to 1598 rev/min, 0.0215 to 0.0573, and 230.38 N to 612.27 N, respectively. At 1.44° of front wheel total toe-in angle, the rates of engine rpm, rolling resistance coefficient, and rolling resistance increased up to 42.68%, 165.77%, and 165.77%, respectively, as the vehicle load increased from 1035 kg to 1377 kg.

It should be noted that rolling resistance is proportional to the vehicle load. The increased vehicle load results in increased rolling losses, resulting in increased fuel consumption. The experimental results confirm that energy and fuel consumption varied with respect to the vehicle load. From Table D.9, we can see that the vehicle load changed from 1035 kg to 1377 kg, which caused the energy and fuel consumption to

change from 1382.28 kJ to 3673.67 kJ and 372 ml to 583 ml, respectively, at 1.44° of front wheel total toe-in angle. It is also shown that the car's KPL decreased considerably with the increase in vehicle load. At 1035 kg vehicle load, the car's (KPL) is about 16.12 km/L, and the KPL reduction rate is 0%. When the vehicle load of the front wheel's total toe-in angle at 1.44° was increases to 1377 kg, the car's (KPL) is reduced to 10.29 km/L, and the KPL reduction rate is about 36.20%. Due to the increase in vehicle load from 1035 kg to 1377kg, the car travelled 5.83 km/L (16.12 km/L to 10.29 km/L) less.

5.12.2 on Travelling Cost (at Front Wheel Total Toe-in Angle, 1.44°)

The traveling cost per km in USD and BDT of the light vehicle as a function of the vehicle load at 1.44° of front wheel total toe-in angle is presented in Table 5.16 and Figure 5.17. Table 5.16 and Figure 5.17 show that the cost of traveling per kilometer went up from \$0.0690 to \$0.1081 (or from 5.5180 Tk to 8.6478 Tk) as the vehicle's load changed at 1.44° of front wheel total toe-in angle (1035 kg to 1377 kg), and the rate of increase went up by up to 56.72%.

Table 5.16 Experimental data for 20 test runs for different vehicle load at 1.44° of front wheel total toe-in angles and travelling costs

Vehicle Load (kg)	Travelling Cost Per km $T_{C/km} = f_c \times f_p$		Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km (min)}}{T_{C/km (min)}}$
	USD	BDT	
1035	0.0690	5.5180	0.00
1053	0.0706	5.6515	2.42
1071	0.0725	5.7998	5.11
1089	0.0742	5.9333	7.53
1107	0.0760	6.0817	10.22
1125	0.0781	6.2448	13.17
1143	0.0795	6.3635	15.32
1161	0.0816	6.5267	18.28
1179	0.0833	6.6602	20.70
1197	0.0853	6.8233	23.66
1215	0.0871	6.9717	26.34
1233	0.0892	7.1348	29.30
1251	0.0914	7.3128	32.53
1269	0.0935	7.4760	35.48
1287	0.0959	7.6688	38.98
1305	0.0983	7.8617	42.47
1323	0.1005	8.0397	45.70
1341	0.1025	8.2028	48.66
1359	0.1049	8.3957	52.15
1377	0.1081	8.6478	56.72

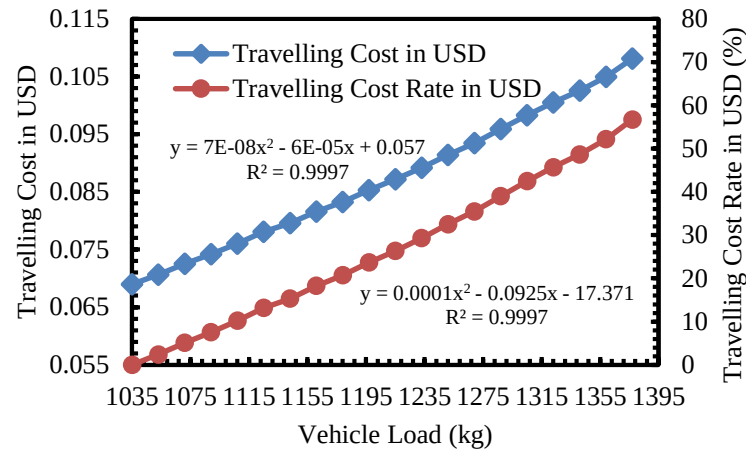


Figure 5.17 Change of travelling cost per Kilometer in USD and its rate in USD (%) with respect to vehicle load at 1.44° of front wheel total toe-in angle

Similarly, the experimental result has been verified by Pearson's correlation coefficient, presented in equation 3.14. The Pearson's correlation coefficient relationship between the vehicle load and fuel consumption is presented in Table E.9 (Appendix E). The correlation of Pearson's coefficient is found to be 0.99 for vehicle load at 1.44° of front wheel total toe-in angle and fuel consumption, which is within the range of $0.75 < r_{vlf_c} < 1$ [201]. There is a very strong positive correlation between the vehicle load at 1.44° of front wheel total toe-in angle and the fuel consumption of a light vehicle.

5.12.3 Correlation Develops Between Vehicle Load and Fuel Consumption at 1.44° of Front Wheel Total Toe-in Angle

Based on experimental data as shown in Table 5.17 from Appendix Tables G.6(a), G.6(b), and G.6(c), ten different correlation models are developed using the regression method, which are: i) model 1- correlation between vehicle load at 1.44° of front wheel total toe-in angle and fuel consumption; ii) model 2- correlation between engine RPM and fuel consumption; iii) model 3- correlation between rolling resistance coefficient and fuel consumption; iv) model 4- correlation between rolling resistance and fuel consumption; v) model 5- correlation between energy consumption and fuel consumption; vi) model 6- correlation among vehicle load at 1.44° of front wheel total toe-in angle, engine RPM and fuel consumption; vii) model 7- correlation among vehicle load at 1.44° of front wheel total toe-in angle, rolling resistance coefficient and fuel consumption; viii) model 8- correlation among vehicle load at 1.44° of front wheel total toe-in angle, rolling resistance, and fuel consumption; ix) model 9- correlation among vehicle load at 1.44° of front wheel total toe-in angle, energy consumption and

fuel consumption; x) model 10- correlation among vehicle load at 1.44° of front wheel total toe-in angle, engine RPM, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption.

Table 5.17 Different simple and multiple correlation regression models for vehicle load condition at 1.44° of front wheel total toe-in angle.

Model No.	Form of Equation	Equation constants		Multiple R	R Square	Adjusted R Square	Stand. Error	P-value
		m_i	k_i					
1	$(F_c)_{1,1} = m_1 v_l + k_l$	$m_1 = 0.6060$	-261.3548	0.9980	0.9961	0.9959	4.1391	3.71206E-23
2	$(F_c)_{2,1} = m_1 R/M + k_l$	$m_1 = 0.3924$	-51.2768	0.9931	0.9863	0.9855	7.7649	3.08733E-18
3	$(F_c)_{3,1} = m_1 \mu_{RRc} + k_l$	$m_1 = 5530.9551$	271.0387	0.9937	0.9875	0.9868	7.4225	1.37038E-18
4	$(F_c)_{4,1} = m_1 F_R + k_l$	$m_1 = 0.5177$	270.8862	0.9937	0.9875	0.9868	7.4279	1.38845E-18
5	$(F_c)_{5,1} = m_1 E_c + k_l$	$m_1 = 0.0862$	270.8862	0.9937	0.9875	0.9868	7.4279	1.38847E-18
6	$(F_c)_{6,1} = m_1 v_l + m_2^* R/M + k_l$	$m_1 = 0.4195;$ $m_2 = 0.1229$	-199.6047	0.9993	0.9986	0.9984	2.5514	3.81751E-05
7	$(F_c)_{7,1} = m_1 v_l + m_2^* \mu_{RRc} + k_l$	$m_1 = 0.3974;$ $m_2 = 1939.9308$	-79.3348	0.9997	0.9995	0.9994	1.4914	3.61825E-09
8	$(F_c)_{8,1} = m_1 v_l + m_2^* F_R + k_l$	$m_1 = 0.3975;$ $m_2 = 0.1814$	-79.5366	0.9997	0.9995	0.9994	1.4985	3.9229E-09
9	$(F_c)_{9,1} = m_1 v_l + m_2^* E_c + k_l$	$m_1 = 0.3975;$ $m_2 = 0.0302$	-79.5367	0.9997	0.9995	0.9994	1.4985	3.92291E-09
10	$(F_c)_{10,1} = m_1 v_l + m_2 R/M + m_3 \mu_{RRc} + m_4 F_R + m_5 E_c + k_l$	$m_1 = 0.4158;$ $m_2 = -0.1288;$ $m_3 = 4875.2990;$ $m_4 = 0;$ $m_5 = -0.0201$	10.4295	0.9999	0.9998	0.9330	1.0107	#NUM!
$F_c = 10.4295272 + 0.415817152^*(v_l) - 0.12884117^*(R/M) + 4875.299079^*(\mu_{RRc}) + 0^*(F_R) - 0.020120122^*(E_c)$								

The R square, adjusted R square, and p -values of the corresponding regression model are shown in Table 5.17. In model 1, the R square and adjusted R^2 values are 0.9961 and 0.9959, respectively. These values are pretty close, and the p -values are less than 0.05, proving a perfect relationship between the vehicle load and fuel consumption.

Similarly, models 2–10 show a perfect relationship between the dependent and independent variables because the correlation R -square values for models tend towards 1.

5.13 EFFECT OF VEHICLE LOAD AT FRONT WHEEL TOTAL TOE-OUT ANGLE, -1.44°

The fuel consumption is calculated using a change in vehicle load (from 1035 kg to 1377 kg) at a vehicle speed of 30 km/hr and standard tire pressure of 32 psi with the wheel front total toe-out angle at -1.44° on the basis of Table D.10 shown in Appendix D. Again, the testing procedure and number of tests remained the same as in Section 5.11. Here, the experimental results at a front wheel total toe-out angle of -1.44° with different vehicle loads are discussed using tabular data instead of a plotted graph, because the plotted graph shows a pattern that is very similar to the experimental results. From Table D.10, it is seen that the engine rpm and rolling resistance coefficient changed (from 1125 rev/min to 1679 rev/min and 0.0218 to 0.0596) with the changes in the vehicle load (from 1035 kg to 1377 kg) at -1.44° of front wheel total toe-out angle. For the same vehicle load variation (1035kg to 1377kg), the rates of engine rpm and rolling resistance coefficient increased by up to 49.24% and 173.68%, respectively. It is also observed that the fuel consumption changed from 376 ml to 598 ml with the changes in vehicle load variation, and the rate of fuel consumption increased up to 59.04%.

As shown in Table D.10, it has been seen that the KPL decreased considerably with the increase in vehicle load at -1.44° of front wheel total toe-out angle. The vehicle's KPL reduction rate is 37.12% when the load increases from 1035 kg to 1377 kg.

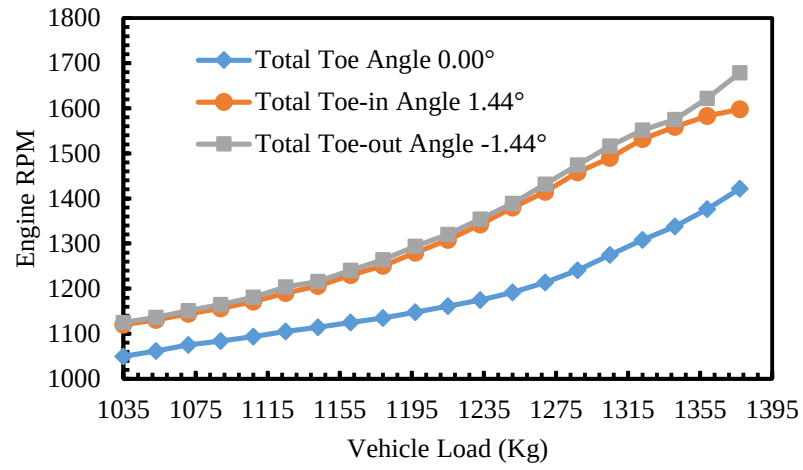
The experimental result has been verified by Pearson's correlation coefficient, presented in Equation 3.14 and tabulated in Table E.10 (Appendix E). The correlation of Pearson's coefficient is found to be 0.99 for vehicle load at -1.44° of front wheel total toe-out angle and fuel consumption, which is within the range of $0.75 < r_{vlf_c} < 1$

[201]. There is a very strong positive correlation between the vehicle load at -1.44° of front wheel total toe-out angle and the fuel consumption of a light vehicles.

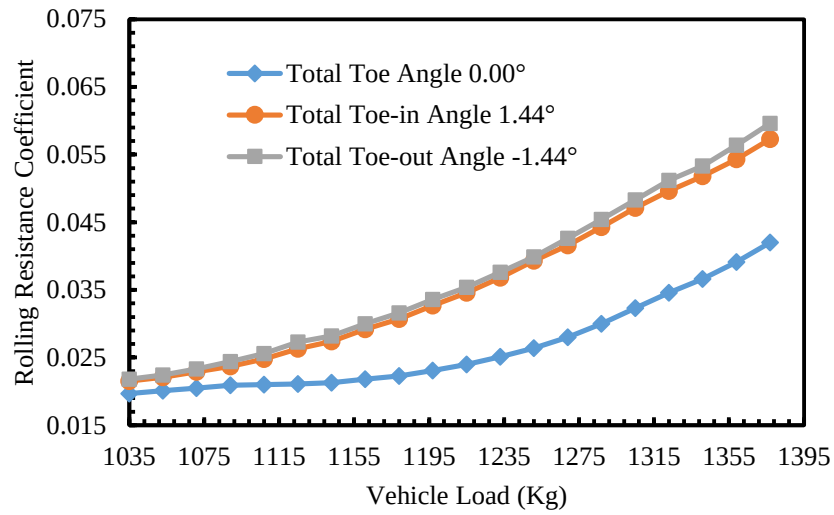
5.14 COMPARISON RESULT BASED ON THE EFFECT OF VEHICLE LOAD AT FRONT WHEEL TOTAL TOE ANGLE, 0° , 1.44° (TOE-IN), AND -1.44° (TOE-OUT)

Figures 5.18(a)–(e) show the comparison results of the engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption as a function of vehicle load. The experimental finding indicates some dependence on the load of the vehicle. In Figure 5.18(e), it is observed that the rate of fuel consumption is a little bit higher for the front wheel total toe-out angle at -1.44° of vehicle load compared to toe angles at 0.00° and 1.44° . The result also exhibits that the front wheel's total toe-out angle produces more rolling resistance compared to the front wheel's total toe-in angle. However, at 0.00° of the front wheel total toe angle, the gradual increase of rolling resistance with vehicle load and the trend are non-linear in nature, whereas the sharp increase is observed at toe-in (1.44°) and toe-out (-1.44°). Interestingly, for lower vehicle loads, changes in the wheel toe angle have relatively little effect on rolling resistance; nevertheless, as the vehicle load rises, the difference becomes more pronounced. This is because when the vehicle's load increased, the wheel tire's deformation increased, increasing the grip friction of the driven tires.

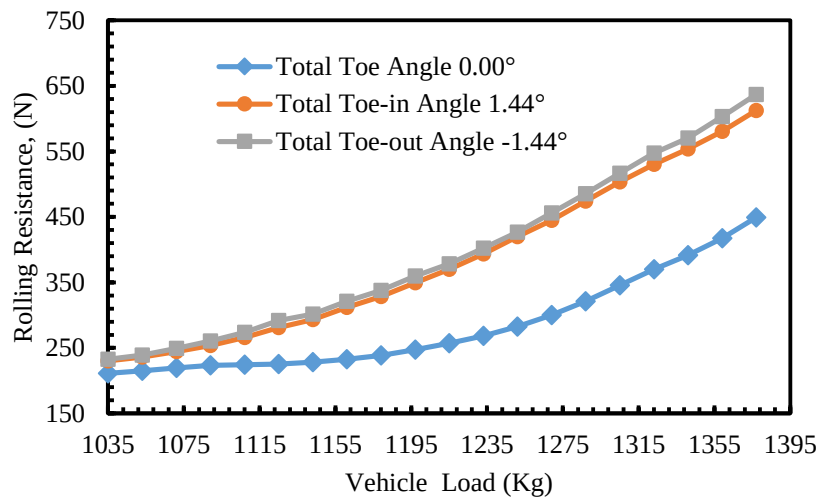
In comparison to the front wheel total toe angle of 0.00° , the fuel consumption rate is 3.35% higher at the front wheel total toe-out angle of -1.44° and 2.32% higher at the front wheel total toe-in angle of 1.44° when the vehicle load increased from 1035 kg to 1377 kg. It is also found that the KPL reduction rate of 1.35% is higher at the front wheel total toe-out angle of -1.44° and 0.92% is higher at the front wheel total toe-in angle of 1.44° compared to 0.00° at the front wheel total toe angle. Based on the experimental results, it is found that there is some dependency between vehicle load, wheel toe angle, and fuel consumption when compared to 0.00° of wheel toe angle. However, almost no difference has been observed for toe-in/ toe-out with variation in vehicle loads.



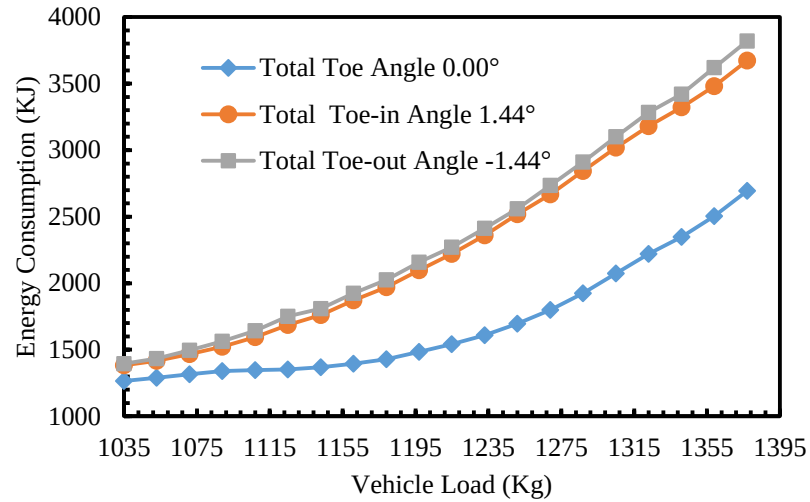
(a)



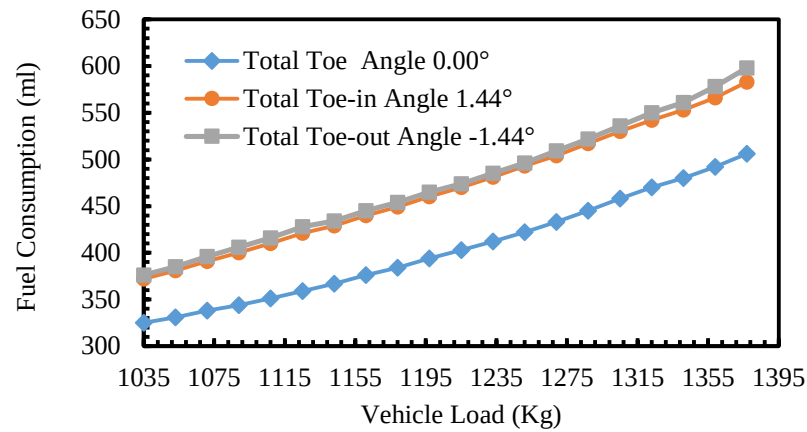
(b)



(c)



(d)



(e)

Figure 5.18 Comparison of (a) Engine RPM, (b) Rolling resistance coefficient, (c) Rolling resistance (N), (d) Energy consumption (Kj), and (e) Fuel consumption (ml) with respect to vehicle load at front wheel total toe angle 0.00°, at front wheel total toe-in angle 1.44°, and at front wheel total toe-out angle -1.44°.

5.14.1 on Travelling Cost

Figure 5.19 (from Tables 5.14 and 5.16 (Appendix F.4)) shows a comparison of the light vehicle's travel cost per kilometer in US dollars based on the vehicle's load at front wheel total toe angles of 0.00°, 1.44° (toe-in angle), and -1.44° (toe-out angle). It is observed that the rate of the traveling cost is a little bit higher for the vehicle load at -1.44° of front wheel total toe-out angle. In comparison to the vehicle load at 0.00° of the front wheel total toe angle, the traveling cost rate of 3.35% is higher at a front wheel total toe-out angle of °-1.44°, and 2.32% is higher at a front wheel total toe-in angle of 1.44°.

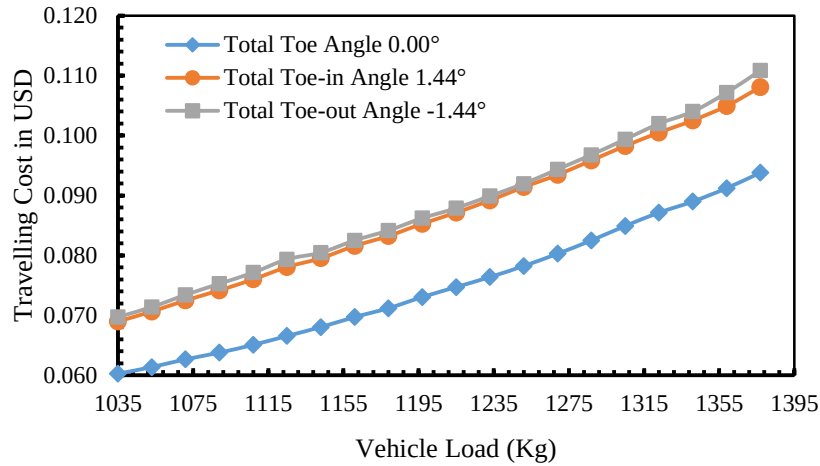


Figure 5.19 Comparison of travelling cost in USD with respect to vehicle load at front wheel total toe angle 0.00° , 1.44° (toe-in) and -1.44° (toe-out).

5.15 EFFECT OF BRAKE APPLICATION

5.15.1 on Fuel Consumption (at 0.00° of Front Wheel Total Toe Angle)

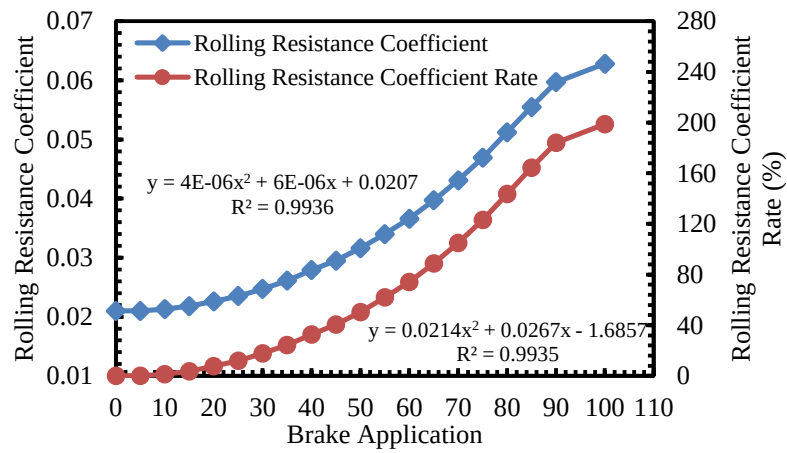
Brake application has a significant impact on vehicle control. The stability and safety of the vehicle, as well as accidents, significantly hinder the use of frequent brakes [41]. The tire life may be greatly prolonged if the tire is operated with less braking application over an extended length of time. On the other side, a tire with a more robust braking application may harm the vehicle and cause rapid tire wear as it travels over road irregularities like potholes and bumps, and comfort may be slightly reduced. Overuse or hard braking can hasten brake pad wear, which may lead to issues with the brake rotors and suspension. The relationship between rolling resistance and brake application is widely understood. More braking results in more rolling loss, which increases fuel consumption. According to experiments conducted in Europe, aggressive driving, which includes frequently accelerating quickly and braking forcefully, can increase fuel consumption by about 40% [207].

A Toyota Echo Plus-2ZZ-GE-02 light-duty vehicle is used to investigate the effect of brake application on fuel performance. The fuel consumption is measured based on the number of brake applications at a 0.00° front wheel total toe angle, as shown in Table D.11 (Appendix D). In this case, a tire pressure of 32 psi (the recommended standard tire pressure for a 175/70R14 tire) and a vehicle speed of 30 km/h are maintained. Like in the previous test, the fuel consumption rate is calculated after a travel distance of 6 km. Other data associated with the test is also monitored. Three sets of repeated data

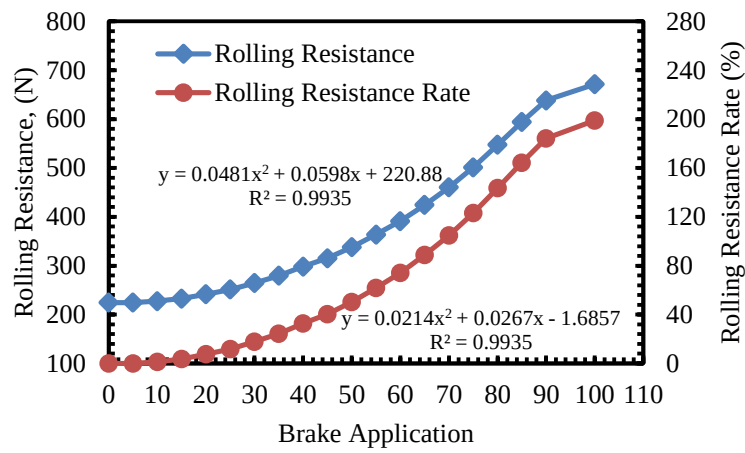
are measured for each test over time to determine an average value and minimize the error in the results. There were a total of 20 tests performed, with the car weighing 1090 kg during all testing (including the driver and one passenger).

The effects of brake application on rolling resistance coefficient, rolling resistance, energy consumption, fuel consumption, and KPL at a 0° toe angle are graphically illustrated in Figures 5.20(a)–(e). It is observed that the rolling resistance varies almost non-linearly (2nd order polynomial shown in Figure 5.20(b)) with the brake application under steady-state conditions (i.e., constant load, speed, and toe angle), while the fuel consumption varies almost linearly (Figure 5.20(d)). Interestingly, energy consumption also increases almost non-linearly with brake application (Figure 5.20(c)). The experimental results of the non-linear relationship agreed well with previously published results [206]. According to the test results, increasing the number of brakes applied (0 to 100 times) resulted in increased rolling resistance and fuel consumption (Figures 5.20(b) and 5.20(d)). The rates of rolling resistance coefficient and rolling resistance increased to 198.79% and 198.79%, respectively, and the rates of energy consumption and fuel consumption increased to 198.79% and 81.74%, respectively. This is due to the wheel skid and the resulting frictional scuffing caused by transferring tractive or braking forces, which show higher rolling resistance [191]. This is also brought on by increasing acceleration, which results in higher fuel consumption.

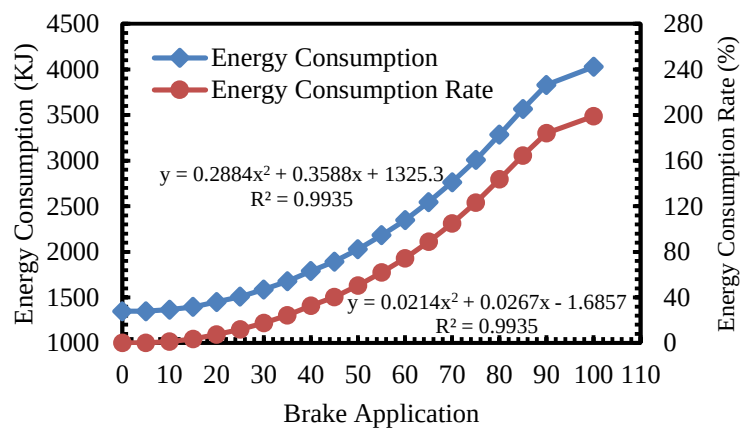
The KPL in Figure 5.20(e) significantly dropped as brake application increased. The car's KPL is approximately 17.39 km/L while the brakes are not being utilized, and the KPL reduction rate is 0%. When the number of applied brakes is increased to 100 times, the car's KPL is reduced to 9.57 km/L, and the KPL reduction rate is about 44.98%. Thus, fuel consumption increased by 81.74%. It is clear that the increment in braking causes more fuel consumption, and the car traveled 7.82 km/L less compared to without braking.



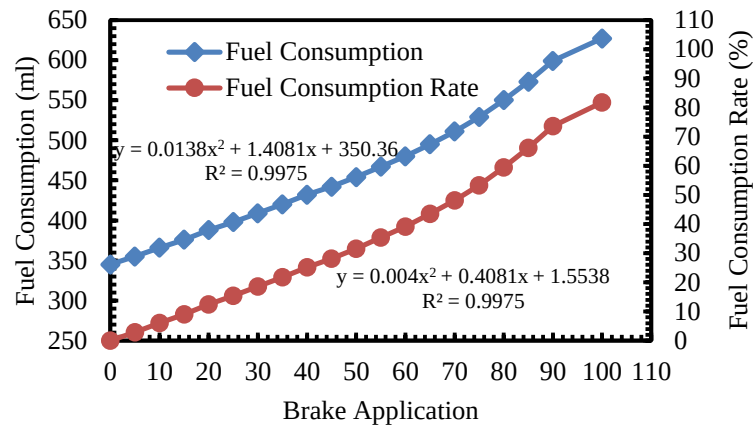
(a)



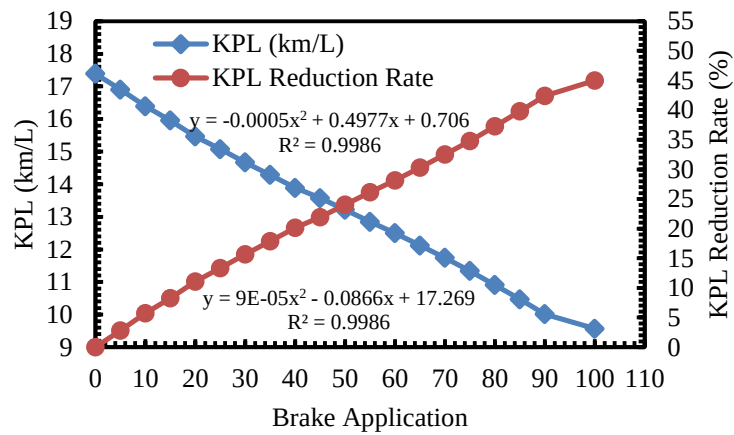
(b)



(c)



(d)



(e)

Figure 5.20 Variation of (a) Rolling resistance coefficient and its rate (%), (b) Rolling resistance (N) and its rate (%), (c) Energy consumption (Kj) and its rate (%), (d) Fuel consumption (ml) and its rate (%), and (e) KPL (Km/L) and KPL reduction rate (%) change with respect to brake application at 0.00° of front wheel total toe angle.

5.15.2 on Travelling Cost (at 0.00° of Front Wheel Total Toe Angle)

The traveling cost per km in USD and BDT of the light vehicle as a function of the brake application is presented in Table 5.18 and graphically shown in Figure 5.21. In Figure 5.21, it is shown that the traveling cost per km of the vehicle increased (from 0.0640\$ to 0.1163\$, or 5.1175Tk to 9.3005Tk) with the changes in the number of braking cycles at 0.00° of the front wheel total toe angle. It shows that the rate of travel costs per km increased to 81.74%.

Table 5.18 Experimental data for 20 test runs for different brake application at 0.00° of front wheel total toe angles and travelling costs

Brake Application (times)	Travelling Cost Per km $T_{C/km} = f_c \times f_p$		Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km(min)}}{T_{C/km(min)}}$
	USD	BDT	
0	0.0640	5.1175	0.00
5	0.0658	5.2658	2.90
10	0.0679	5.4290	6.09
15	0.0697	5.5773	8.99
20	0.0719	5.7553	12.46
25	0.0738	5.9037	15.36
30	0.0758	6.0668	18.55
35	0.0779	6.2300	21.74
40	0.0801	6.4080	25.22
45	0.0820	6.5563	28.12
50	0.0842	6.7343	31.59
55	0.0866	6.9272	35.36
60	0.0890	7.1200	39.13
65	0.0918	7.3425	43.48
70	0.0947	7.5798	48.12
75	0.0981	7.8468	53.33
80	0.1020	8.1583	59.42
85	0.1062	8.4995	66.09
90	0.1111	8.8852	73.62
100	0.1163	9.3005	81.74

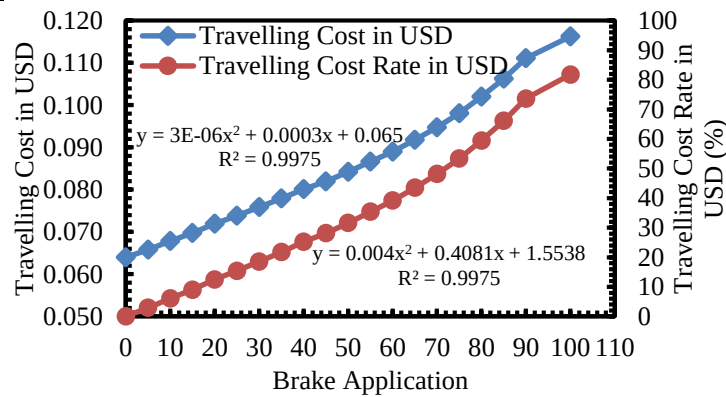


Figure 5.21 Variation of travelling cost per Kilometer in USD and its rate in USD (%) with respect to brake application at 0.00° of front wheel total toe angle.

Furthermore, the result has been verified by Pearson's correlation coefficient, presented in Equation 3.14. The Pearson's correlation coefficient relationship between brake application and fuel consumption is presented in Table E.11 (Appendix E). The correlation coefficient, r_{bafc} is found to be 0.98 between brake application and fuel consumption at 0.00° of front wheel total toe angle, which is within the range of $0.75 < r_{bafc} < 1$ [201], providing a very strong positive correlation between the brake

application and fuel consumption of light vehicles at 0.00° of front wheel total toe angle.

5.15.3 Correlation Develops Between Brake Application and Fuel Consumption at 0.00° of Front Wheel Total Toe Angle

Using the regression method, eight different correlation models are made between the dependent variable (fuel consumption) and the independent variables (brake application at 0.00° of front wheel total toe angle, rolling resistance coefficient, rolling resistance, and energy consumption) based on the experimental data as shown in Table 5.19 from Appendix Tables G.7(a), G.7(b), and G.7(c). These models are: i) model 1- correlation between brake application at 0.00° of front wheel total toe angle and rolling resistance coefficient; ii) model 2- correlation between rolling resistance coefficient and fuel consumption; iii) model 3- correlation between rolling resistance and fuel consumption; iv) model 4- correlation between energy consumption and fuel consumption; v) model 5- correlation among brake application at 0.00° of front wheel total toe angle, rolling resistance coefficient and fuel consumption; vi) model 6- correlation among brake application at 0.00° of front wheel total toe angle, rolling resistance and fuel consumption; vii) model 7- correlation among brake application at 0.00° of front wheel total toe angle, energy consumption, and fuel consumption; viii) model 8- correlation among brake application at 0.00° of front wheel total toe angle, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption.

There are a total of 20 tests run on each regression model. Table 5.19 provides the R square, adjusted R square, and p -values for the associated regression model. In model 1, it is observed that the R square and adjusted R square values are 0.9793 and 0.9781, respectively. The p -values are also found to be less than 0.05, indicating that there is a perfect correlation between the brake application at 0.00° of front total toe angle and fuel consumption. Similarly, models 2–8 show a perfect relationship between the dependent and independent variables because the correlation R -square values for the models are about 1.

Table 5.19 Different simple and multiple correlation regression models for brake application at 0.00° of front wheel total toe angle.

Model No.	Form of Equation	Equation constants		Multiple R	R Square	Adjusted R Square	Stand. Error	P-value
		m _i	k ₁					
1	$(F_c)_{1,I} = m_1 b_a + k_1$	m ₁ = 2.7436	329.7923	0.9896	0.9793	0.9781	12.2912	1.28452E-16
2	$(F_c)_{2,I} = m_1 \mu_{RRC} + k_1$	m ₁ = 5961.2756	252.0063	0.9880	0.9763	0.9749	13.1620	4.40999E-16
3	$(F_c)_{3,I} = m_1 F_R + k_1$	m ₁ = 0.5576	251.9496	0.9880	0.9762	0.9749	13.1652	4.42906E-16
4	$(F_c)_{4,I} = m_1 E_c + k_1$	m ₁ = 0.0929	251.9496	0.9880	0.9762	0.9749	13.1652	4.42905E-16
5	$(F_c)_{5,I} = m_1 b_a + m_2^* \mu_{RRC} + k_1$	m ₁ = 1.4524; m ₂ = 2924.1956	289.0272	0.9986	0.9973	0.9970	4.5276	5.2791E-09
6	$(F_c)_{6,I} = m_1 b_a + m_2^* F_R + k_1$	m ₁ = 1.4525; m ₂ = 0.2735	288.9989	0.9986	0.9973	0.9970	4.5112	4.96038E-09
7	$(F_c)_{7,I} = m_1 b_a + m_2^* E_c + k_1$	m ₁ = 1.4525; m ₂ = 0.0455	288.9989	0.9986	0.9973	0.9970	4.5112	4.96038E-09
8	$(F_c)_{8,I} = m_1 b_a + m_2 \mu_{RRC} + m_3 F_R + m_4 E_c + k_1$	m ₁ = 1.4525; m ₂ = 0; m ₃ = 0; m ₄ = 0.0455	288.9989	0.9986	0.9973	0.8794	4.5112	#NUM!
$F_c = 288.9989074 + 1.452581241*(b_a) + 0*(\mu_{RRC}) + 0*(F_R) + 0.045591208*(E_c)$								

5.16 EFFECT OF BRAKE APPLICATION

5.16.1 on Fuel Consumption (at a Front Wheel Total Toe-in Angle of 1.44°)

Here, the same vehicle is considered as used in the previous section to analyze the effect of brake application on fuel performance. The fuel consumption is measured as a function of the number of brake applications at 1.44° of the front wheel's total toe-in angle (as shown in Table D.12 in Appendix D). The testing procedure and other conditions remain the same as those discussed in Section 5.15.

The experimental results show a similar trend, that is, the engine rpm, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption all increased as the number of times a brake is applied increased from 0 to 100 while maintaining the same tire inflation pressure, vehicle load, and speed, as was graphically discussed in Section 5.15. Therefore, to avoid being annoying, the tabulated data is utilized for discussing the experimental results rather than a graphical representation.

It is widely accepted that rolling resistance increases as the brakes are applied. Increased rolling loss due to more frequent application of the brakes leads to increased fuel consumption. Table D.12 shows that as the number of brake applications increased (from 0 to 100 times), the rolling resistance coefficient and rolling resistance also increased (from 0.0229 to 0.0700 and 244.38 N to 748.25 N, respectively). At 1.44° of front wheel total toe-in angle, the rates of rolling resistance coefficient and rolling resistance are increased up to 206.19% and 206.19%, respectively, when the brake is applied 100 times.

From Table D.12, it is also seen that the energy consumption and fuel consumption increased (from 1466.28 kJ to 4489.55 kJ and 391 ml to 715 ml) with the increased brake application (from 0 to 100 times) at 1.44° of front wheel total toe-in angle. Their rates have increased to 206.19% and 82.86%. The KPL decreased considerably with the increase in the number of brake applications at 1.44° of front wheel total toe-in angle. The car's KPL is about 15.34 km/L, and the KPL reduction rate is considered to be 0% when no brake is applied. The car's KPL is reduced to 8.39 km/L, and the KPL reduction rate is about 45.32%. It is observed that due to the increased brake application of 100 times, the car traveled 6.95 km/L (15.34 km/L–8.39 km/L shown in Table D.12) less compared to without braking.

5.16.2 on Travelling Cost (at Front Wheel Total Toe-in Angle of 1.44°)

Table 5.20 and the graphical illustration in Figure 5.22 show the light vehicle's travelling cost per km in USD and BDT as a function of brake application at 1.44° of front wheel total toe-in angle. According to Table 5.20, it is shown that the traveling cost per km increased from 0.0725\$ to 0.1326\$, or from 5.7998Tk to 10.6058Tk, with the increase in brake applications (0 to 100 times), and the rate of traveling cost per km increased up to 82.86%. Once again, an increase in the number of brake applications can increase the fuel consumption of a light vehicle, as discussed in the above section, which results in increased travel costs.

Table 5.20 Experimental data for 20 test runs for different brake application at 1.44° of front wheel total toe-in angles and travelling costs

Brake Application (times)	Travelling Cost Per km $T_{C/km} = f_c \times f_p$		Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km (min)}}{T_{C/km (min)}}$
	USD	BDT	
0	0.0725	5.7998	0.00
5	0.0747	5.9778	3.07
10	0.0773	6.1855	6.65
15	0.0797	6.3783	9.97
20	0.0823	6.5860	13.55
25	0.0846	6.7640	16.62
30	0.0868	6.9420	19.69
35	0.0892	7.1348	23.02
40	0.0914	7.3128	26.09
45	0.0936	7.4908	29.16
50	0.0959	7.6688	32.23
55	0.0986	7.8913	36.06
60	0.1016	8.1287	40.15
65	0.1046	8.3660	44.25
70	0.1079	8.6330	48.85
75	0.1116	8.9297	53.96
80	0.1161	9.2857	60.10
85	0.1211	9.6862	67.01
90	0.1265	10.1163	74.42
100	0.1326	10.6058	82.86

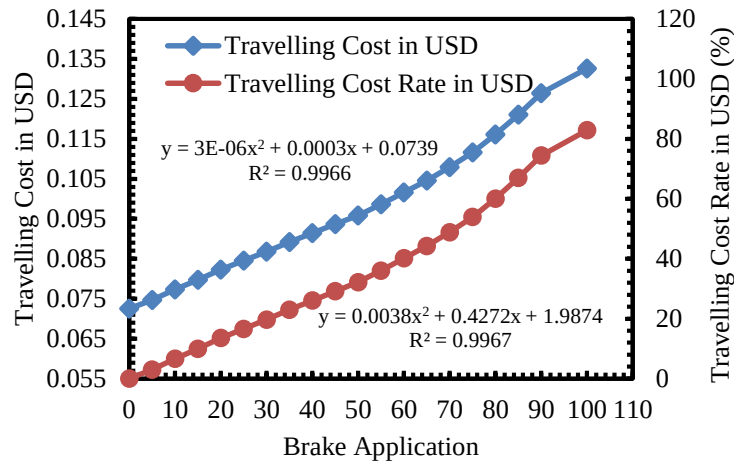


Figure 5.22 Change of travelling cost per Kilometer in USD and its rate in USD (%) with respect to brake application at 1.44° of front wheel total toe-in angle.

Similarly, the experimental result has been verified by the Pearson's correlation coefficient as presented in Table E.12 (Appendix E). The correlation coefficient, r_{bafc} is found to be 0.99 between the brake application at 1.44° of front wheel total toe-in angle and fuel consumption, which is within the range of $0.75 < r_{bafc} < 1$ [201]. Thus, it can be said that there is a very strong positive correlation between the brake application conditions and the fuel consumption of a light vehicle.

5.16.3 Correlation Develops Between Brake Application and Fuel Consumption at 1.44° of Front Wheel Total Toe-in Angle

Eight different correlation models are developed using the regression method based on experimental data, as shown in Table 5.21 from Appendix Tables G.8(a), G.8(b), and G.8(c), which are as follows: i) model 1- correlation between brake application at 1.44° of front wheel total toe-in angle and fuel consumption; ii) model 2- correlation between rolling resistance coefficient and fuel consumption; iii) model 3- correlation between rolling resistance and fuel consumption; iv) model 4- correlation between energy consumption and fuel consumption; v) model 5- correlation among brake application at 1.44° of front wheel total toe-in angle, rolling resistance coefficient and fuel consumption; vi) model 6- correlation among brake application at 1.44° of front wheel total toe-in angle, rolling resistance and fuel consumption; vii) model 7- correlation among brake application at 1.44° of front wheel total toe-in angle, energy consumption and fuel consumption; viii) model 8- correlation among brake application at 1.44° of front wheel total toe-in angle, rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption.

Given experimental data on brake application, engine RPM, rolling resistance, rolling resistance coefficient, and energy consumption, the regression model, as mentioned above, has been assessed to be a good correlation model for predicting fuel consumption. In Table 5.21, the R square and adjusted R square values are found to be 0.9801 and 0.9790, respectively, in model 1; these values are very close, and the p -values are less than 0.05, by which it can be claimed that there is a perfect relationship between the application of brakes and fuel consumption at 1.44° of front wheel total toe-in angle.

Similarly, the values of R square and adjusted R square for models 2–8 tend to be 1 towards a perfect relationship between the dependent and independent variables.

Table 5.21 Different simple and multiple correlation regression models for brake application at 1.44° of front wheel total toe-in angle.

Model No.	Form of Equation	Equation constants		Multiple R	R Square	Adjusted R Square	Stand. Error	P-value
		m _i	k ₁					
1	$(F_c)_{1,1} = m_1 b_a + k_1$	m ₁ = 3.1205	376.4444	0.9900	0.9801	0.9790	13.7121	9.13069E-17
2	$(F_c)_{2,1} = m_1 \mu_{RRC} + k_1$	m ₁ = 6091.1610	254.0583	0.9892	0.9786	0.9775	14.1918	1.69672E-16
3	$(F_c)_{3,1} = m_1 F_R + k_1$	m ₁ = 0.5695	254.0852	0.9893	0.9787	0.9775	14.1712	1.65283E-16
4	$(F_c)_{4,1} = m_1 E_c + k_1$	m ₁ = 0.0949	254.0851	0.9893	0.9787	0.9775	14.1712	1.65283E-16
5	$(F_c)_{5,1} = m_1 b_a + m_2^* \mu_{RRC} + k_1$	m ₁ = 1.9118; m ₂ = 2368.5656	328.6256	0.9904	0.9810	0.9788	13.7675	0.367972435
6	$(F_c)_{6,1} = m_1 b_a + m_2^* F_R + k_1$	m ₁ = 1.8953; m ₂ = 0.2245	327.9837	0.9904	0.9810	0.9788	13.7566	0.360328633
7	$(F_c)_{7,1} = m_1 b_a + m_2^* E_c + k_1$	m ₁ = 1.8953; m ₂ = 0.0374	327.9838	0.9904	0.9810	0.9788	13.7566	0.360328785
8	$(F_c)_{8,1} = m_1 b_a + m_2 \mu_{RRC} + m_3 F_R + m_4 E_c + k_1$	m ₁ = 1.9766; m ₂ = -72587.4160; m ₃ = 0; m ₄ = 1.1662	331.5807	0.9907	0.9815	0.9155	14.0102	#NUM!
$F_c = 331.5807281 + 1.976634176*(b_a) - 72587.41601*(\mu_{RRC}) + 0*(F_R) + 1.1662163*(E_c)$								

5.17 EFFECT OF BRAKE APPLICATION AT FRONT WHEEL TOTAL TOE-OUT ANGLE OF -1.44°

Table D.13 in Appendix D shows the fuel consumption data as a function of the brake application (from 0 to 100 times) when the wheel front total toe-out angle is -1.44° . The testing procedure and other parameters associated with this experiment remained the same as in Section 5.15. Since the plotted graph shows a pattern that is very similar to the one shown in Section 5.15, the experimental results at a front wheel total toe-out angle of -1.44° with different brake applications are again discussed using a table of data instead of a graph.

The results of this experimental study show that the rolling resistance coefficient, rolling resistance, energy consumption, fuel consumption, and their rates increased as the number of brake applications increased. It is shown that the rolling resistance coefficient and its rate increased (from 0.0231 to 0.0714 and 208.94%, respectively) as the application brakes increased from 0 to 100 times. Consequently, the rolling resistance and energy consumption changed (from 247.28 N to 763.95 N and 1483.68 kJ to 4583.73 kJ) with the changes in the brake application (0 to 100 times), and their rates increased by up to 208.94% and 208.94%, respectively. Therefore, the fuel consumption is increased by up to 85.28% at 100 times the brake applied.

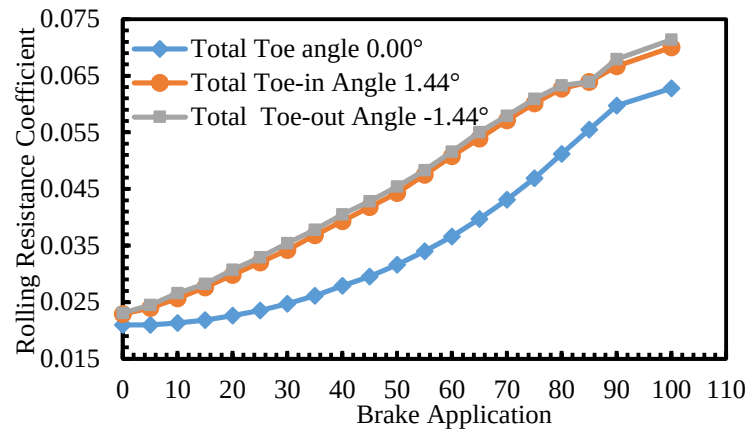
It is also observed that at -1.44° of front wheel total toe-out angle, the car's KPL is about 15.22 km/L when no brake is applied, and the KPL reduction rate is considered to be 0%. However, the car's KPL is reduced to 8.21 km/L as the brake is applied 100 times, and the KPL reduction rate is about 46.03%. So, it can be seen that due to the increase in brake application from 0 to 100 times, the car traveled 7.01 km/L (15.22 km/L–8.21 km/L) less.

Pearson's correlation coefficient has verified the result and shows a very strong positive correlation between the brake application at -1.44° of front wheel total toe-out angle and the fuel consumption of a light vehicle.

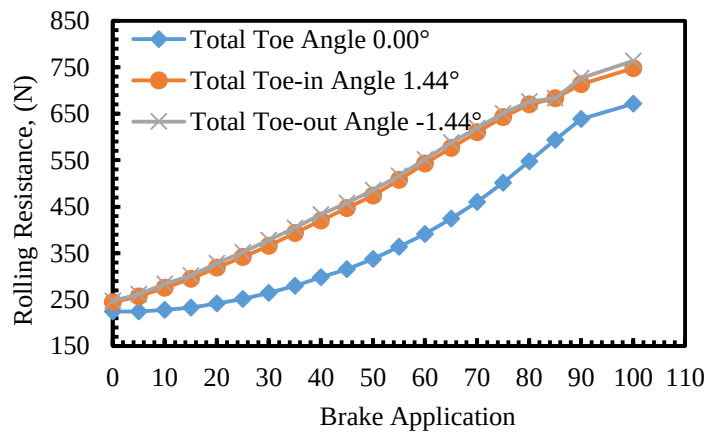
5.18 COMPARISON RESULT OF EFFECT OF BRAKE APPLICATION AT DIFFERENT FRONT WHEEL TOTAL TOE ANGLES OF 0° , 1.44° (TOE-IN), AND -1.44° (TOE-OUT)

Figures 5.23(a)–(d) show the comparison results of the rolling resistance coefficient, rolling resistance, energy consumption, and fuel consumption as a function of brake application. Once again, the experimental results show some dependence on the brake application. In Figure 5.23(d), it is seen that the rate of fuel consumption is a little bit higher at -1.44° of front wheel total toe-out angle with brake application compared to toe angles at 0.00° and 1.44° . This is because, at the front wheel's total toe-out angle of -1.44° , more rolling resistance is produced compared to the front wheel's total toe angle of 0.00° and total toe-in angle of 1.44° . But at 0.00° of front wheel total toe angle, rolling resistance goes up slowly when the brakes are used, and the trend is non-linear (2nd order polynomial), while a sharp increase with a linear trend is seen at 1.44° of toe-in and -1.44° of toe-out. Notably, variations in wheel toe angle have relatively little impact on rolling resistance when no brakes are applied; however, the difference becomes more noticeable as the frequency of brake applications increases (Figure 5.23(b)). It is also observed that the differences reduced again as the braking times varied from 90 to 100 times. Owing to the fact that increased road-contact surface of tires caused by frequent brake applications. It is also measured that the fuel consumption rate of 3.54% is higher at the front wheel total toe-out angle of -1.44° and 2.42% is higher at the front wheel total toe-in angle of 1.44° compared to the front wheel total toe angle of 0.00° . The deviation is most likely due to increased skidding caused by braking on the rubber through the increased grip friction of the driven tires.

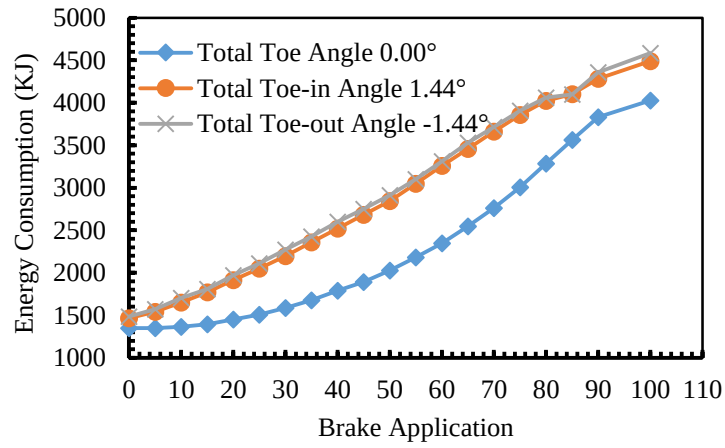
In comparison to the car's KPL, the KPL reduction rate of 1.05% is higher at the front wheel total toe-out angle of -1.44° and 0.71% is higher at the front wheel total toe-in angle of 1.44° compared to the front wheel total toe angle of 0.00° when 100 times brake is applied. Based on the experimental results, it is suggested that there is some dependency between brake application, wheel toe angle, and fuel consumption when compared to 0.00° of wheel toe angle. However, almost no difference is observed for toe-in/ toe-out with brake application.



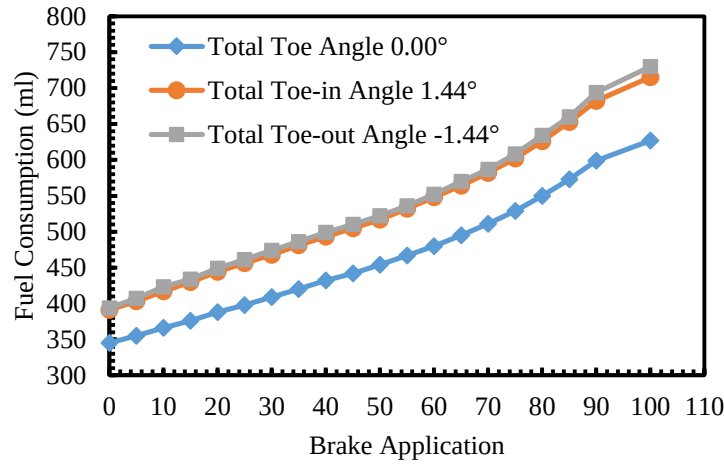
(a)



(b)



(c)



(d)

Figure 5.23 Comparison of (a) Rolling resistance coefficient, (b) Rolling resistance (N), (c) Energy consumption (Kj), and (d) Fuel consumption (ml) with respect to brake application at front wheel total toe angle 0.00°, at front wheel total toe-in angle 1.44° and at front wheel total toe-out angle -1.44°.

5.18.1 on Travelling Cost

A comparison of the traveling cost per km in USD of the light vehicle as a function of the brake application in Figure 5.24 from Tables 5.18, 5.20, and Appendix F.5 shows that the rate of the traveling cost is a little bit higher at -1.44° of front wheel total toe-out angle compared to 1.44° of front wheel total toe-in angle. In comparison to the brake application, the traveling cost rate of 3.54% is higher at the front wheel total toe-out angle of -1.44° and 2.42% at the front wheel total toe-in angle of 1.44° when compared to the front wheel total toe angle of 0.00°.

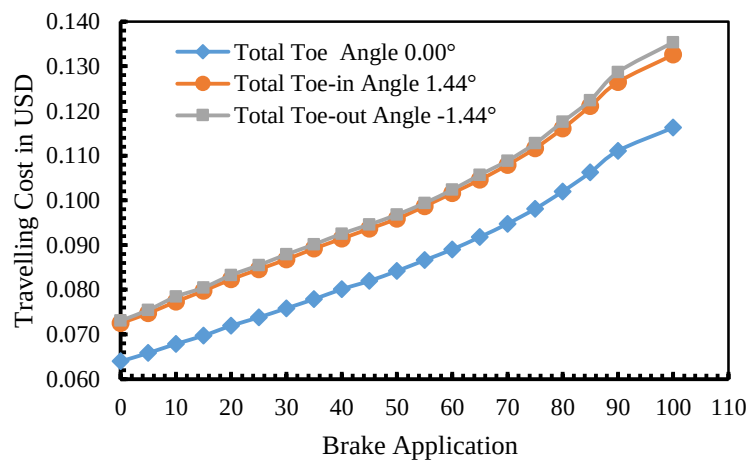


Figure 5.24 Comparison of travelling cost in USD with respect to brake application at front wheel total toe angles of 0.00°, 1.44° (toe-in) and -1.44° (toe-out).

5.19 EFFECT OF FRONT WHEEL TOTAL TOE-IN ANGLE

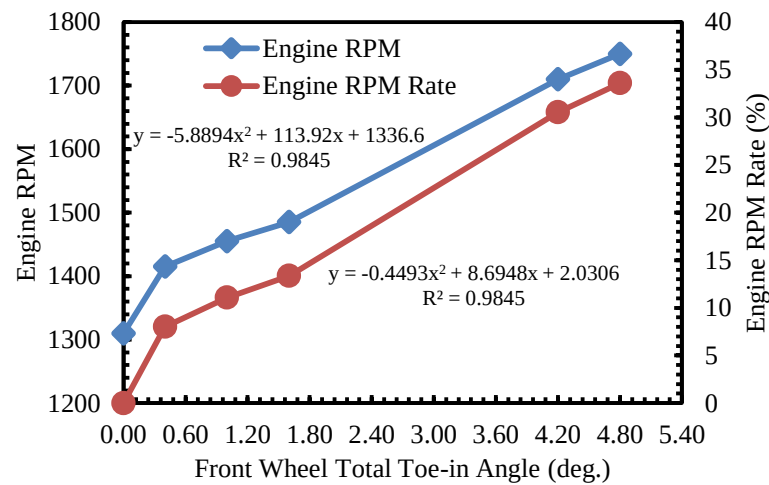
5.19.1 on Tire Wear and Tire Life

It is emphasized that front wheel toe angle is vital in determining vehicle handling as well as passenger comfort and safety. Excessive front wheel toe angle in cars can cause accidents and excessive tire wear [42]. If the tire is operated at the proper front wheel toe angle, the tire life may be greatly extended. On the other hand, an excessive front wheel toe angle may cause damage to the tires while riding over imperfections in road surfaces, such as pushing in one direction and unusual vibration. It is commonly understood that the front wheel toe angle is related to rolling resistance. A higher front wheel toe angle then results in more rolling resistance, which increases tire wear and decreases tire life.

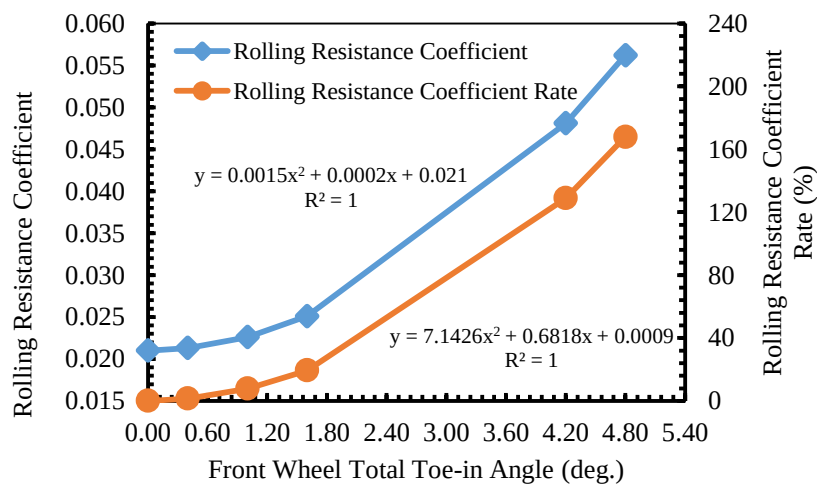
The experiment is carried out under steady-state test conditions (i.e., constant load, speed, and tire pressure), with the only variation in slip angle considered being the wheels' front total toe-in angle, which is thought to be the most important factor that affects the tire wear of the vehicle. During the tests, engine rpm (measured by a tachometer) and other conditions were maintained in the same manner as in Section 4.7.2 for the light-duty vehicle as a function of the front wheel total toe-in (left and right) angle. For each experiment, engine rpm, rolling resistance, rolling resistance coefficient, energy consumption, vehicle load, tire wear, tire traveling life, and tire traveling cost are calculated, as shown in Tables D.14 and D.15 in Appendix D and graphically illustrated in Figures 5.25(a)–5.25(d).

It should be noted that the toe-in/toe-out angle is initially only considered to be 2.53° ; however, for assessing tire wear as a result of the toe-in angle's effect, the front wheel toe-in angle is set to 4.80° . Therefore, further investigation is needed to see how a higher wheel toe-in angle affects the vehicle's fuel performance. Similar results were observed, that is, the engine rpm, rolling resistance coefficient and rolling resistance increased with the increase in the front wheel total toe-in angle (from 0.00° to 4.80°), as shown in Figures 5.25(a)–5.25(c). The rates of engine rpm, rolling resistance coefficient, and rolling resistance are increased by up to 33.59%, 167.84%, and 167.84%, respectively, when the front wheel total toe-in angle increases from 0.00° to 4.80° . It is well known that increases in the toe-in angle cause the rolling resistance to increase more rapidly, as shown in Figure 5.25(c). Because of this, the vehicle may require more energy to maintain a particular speed, reducing its top speed. However,

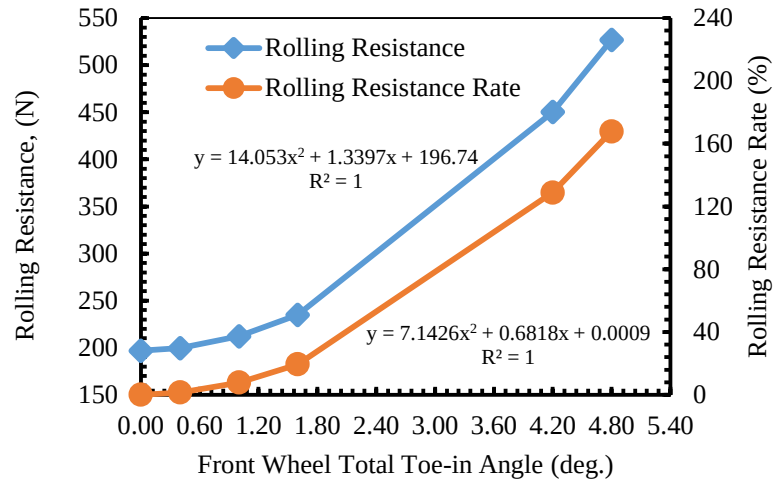
the rolling resistance generated is also dependent on a variety of factors, such as vehicle type, tire properties, tire materials, operating conditions, etc. Figure 5.25(d) shows the energy consumption varied for the front wheel's total toe-in angle. As shown in Figure 5.25(d), energy consumption increased non-linearly with front wheel total toe-in angles under steady-state conditions (i.e., constant load, vehicle speed, and tire pressure). Figure 5.25(d) shows that the energy consumption changed from 1180.44 kJ to 3161.68 kJ with the changes in the front wheel total toe-in angle (0.00° to 4.80°). For the same misalignment condition (0.00° to 4.80°), the energy consumption rate increases to 167.84%. The results presented in Figure 5.25 suggest that the front wheel's total toe-in angle is strongly correlated with rolling resistance.



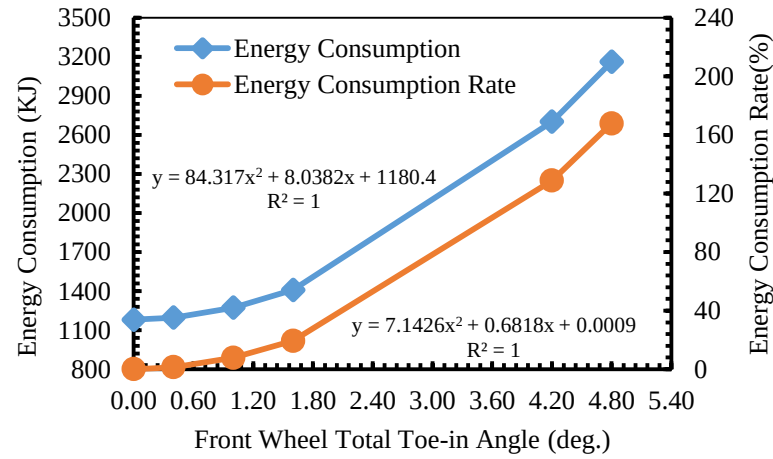
(a)



(b)



(c)



(d)

Figure 5.25(a) Variation of engine RPM and its rate (%), (b) Change of rolling resistance coefficient and its rate (%), (c) Rolling resistance (N) and its rate (%), (d) Variation of energy consumption (Kj) and its rate (%) with respect to front wheel total toe-in angle (deg.)

Figures 5.26(a) and 5.26(b) showed tire outside lateral groove wear, tire inside lateral groove wear, and tire circumferential groove wear, as well as how their wear rates changed with front wheel total toe-in angle (deg.). It is also observed that tire rolling resistance significantly impacts vehicle tire life. Uneven tire wear, especially on the outside edges of the tires, can result from toe-in misalignment. This is because the inside edges of the tires are traveling a shorter distance than the outside edges, which causes more friction and heat to be produced there. Furthermore, excessive toe-in can cause a car's tires to rub or drag on the pavement. This can have a detrimental effect on handling and fuel efficiency, as well as increase tire wear.

It should be noted that a positive toe angle (toe-in) benefits vehicle stability when driving straight. Yurko [38] has been reported to say that a slight change in toe-in angle (0° to 0.5°) should cause only a minimal increase in rolling resistance. Thus, it can be concluded that little changes in toe angle can be allowed; however, more changes in toe angle lead to rapid increases in tire wear, which are not recommended.

To figure out how the changes in the front wheel's total toe-in angle affect uneven tire wear, we tested and estimated tire groove wear for six different test conditions. The results are listed in Table D.15 in Appendix D, and they are also shown in Figures 5.26 (a) and (b). It should be noted that tire wear depends largely on the cornering forces, which are mainly developed by the effect of suspension and steering geometry on the slip angles of the two front wheels [208].

In Figure 5.26(a), the tire circumferential groove wear increases as the wheel front total toe-in angle increases. It is seen that tire outside lateral groove wear, tire inside lateral groove wear, and tire circumferential groove wear changed (from 0.03 mm to 3.01 mm, 0.03 mm to 0.29 mm, and 0.04 mm to 4.35 mm) with the changes in the front wheel total toe-in angle (from 0.00° to 4.80°). Their rates increased up to 9933.33%, 866.67%, and 10775.0%, respectively, when the front wheel total toe-in angle increased from 0.00° to 4.80° (Figure 5.26(b)). In that case, the tire traveling life is found to be 92250 km, whereas the tire wear increment rate and tire traveling life reduction rate with respect to the front wheel total toe-in angle at 0° are considered to be 0%. On the other hand, while the front wheel total toe-in angle increases to 4.80° , the car tire traveling life is reduced to approximately 848.28 km, and the tire wear reduction rate is about 99.08% (Figure 5.26(c)). Thereby, it is seen that due to misalignment of the front wheel's total toe-in angle (from 0.00° to 4.80°), the car tire travels approximately 91401.72 km less for the same size tire. It is also seen that the tire wear increment rate is minimal up to the front wheel total toe-in angle of 1.6° (Figure 5.26(a)); however, further increases in the front wheel total toe-in angle can cause a much faster increase in tire wear. It is worth mentioning that the toe angle over 1 degree is rarely used on vehicles; however, in the worst-case scenario, the 4.80° toe-in angle is used in this study. Figure 5.26(a) shows that tire circumferential groove wear is more severe, particularly in LG1 and CG1. Figure 5.27 shows the passenger car tires with asymmetric wear at a different total toe-in angle. The circumferential grooves CG1–CG3 (Table D.15) showed nearly similar wear patterns. The minimum groove wear is

observed at the tire's inside lateral groove (LG2) compared to the outside lateral groove (LG1). It should be noted that in cases of misalignment of the toe-in angle, tire wear primarily began from the outer shoulder to the inner shoulder of the tire. The outside of the tread wears out much faster for a front tire with an excessive toe-in (4.20°) in Figure 5.27(a). It is also observed that if one's hand rubs through worn-out tread (from inside to outside and vice versa), he or she will experience smoothness from outside to inside. However, sharp edges on the inside edges of the tread were observed. In either case, the tread wear is the same around the tire, and both tires showed the same type of asymmetrical wear. It can be seen that outside and inside shoulder wear around the tire depends on the wheel's front toe angle. The experimental result is well in agreement with the theoretical wear pattern for the toe-in angle of the wheel [209].

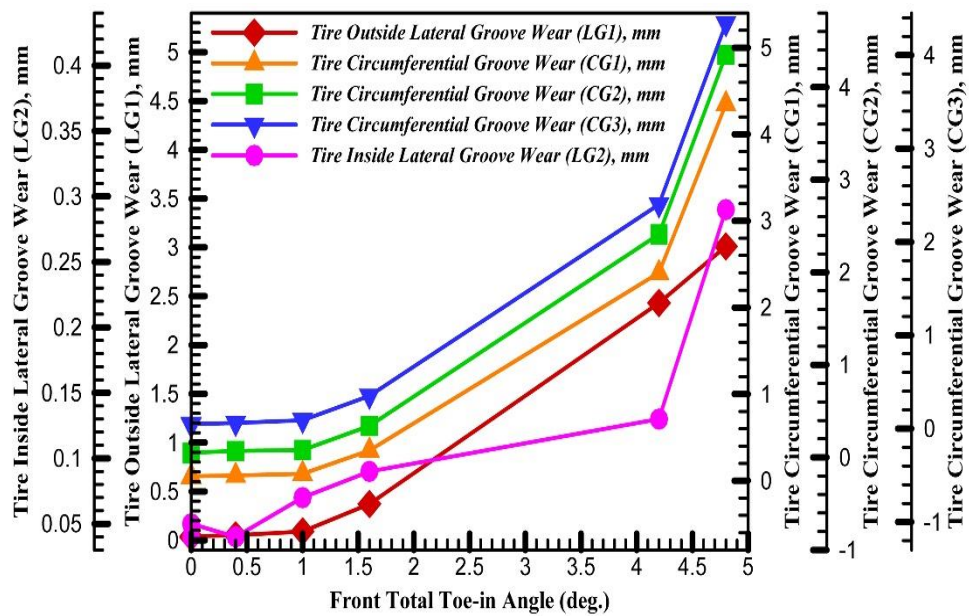


Figure 5.26(a) Variation of tire outside lateral groove wear (mm), tire inside lateral groove wear (mm) and tire circumferential groove wear (mm) with respect to front wheel total toe-in angle (deg.)

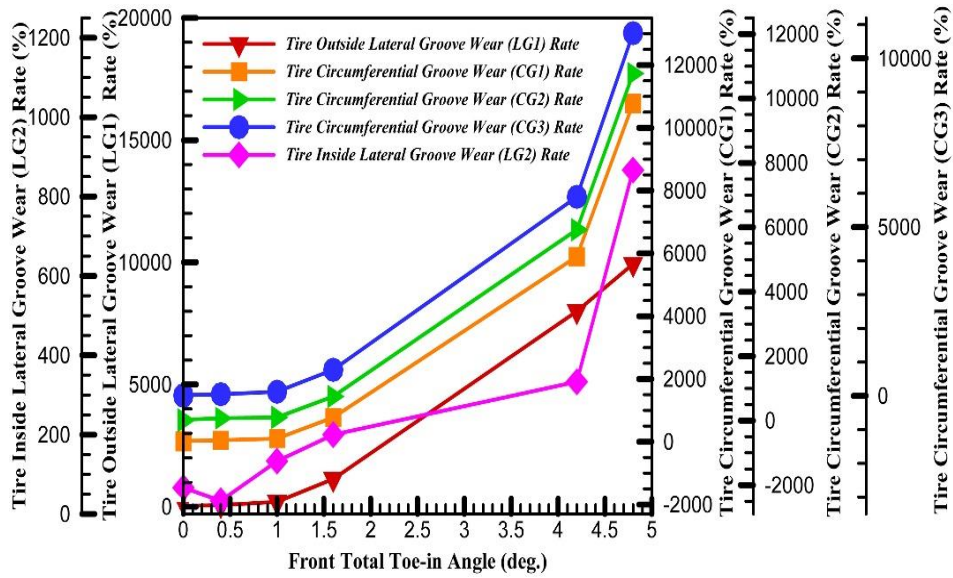


Figure 5.26(b) Change of tire outside lateral groove wear rate (%), tire inside lateral groove wear rate (%), and tire circumferential groove wear rate (%) with respect to front wheel total toe-in angle (deg.).

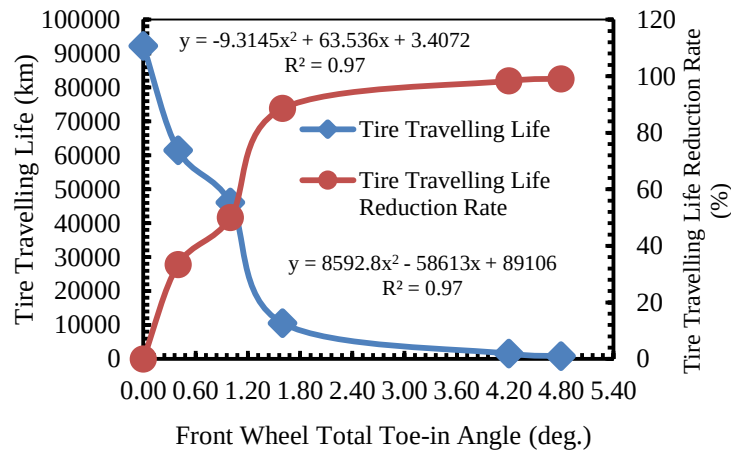


Figure 5.26(c) Tire travelling life (km) and its rate (%) change with respect to front wheel total toe-in angle (deg.).



4.20°

(a)

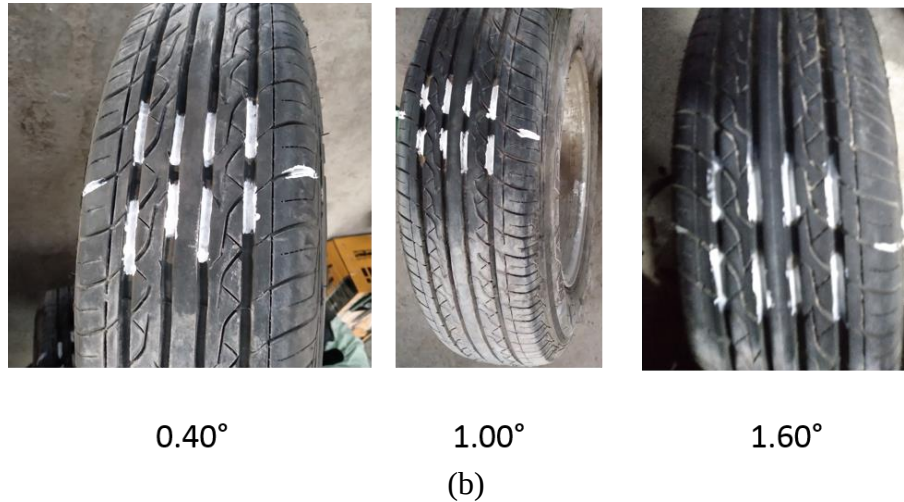


Figure 5.27 A photographic view of tire wear at different front wheel total toe-in angles (a) at 4.20° and (b) at 0.40°, 1.00°, and 1.60°.

In the second case, tire inside lateral groove wear is measured with the variation of vehicle load. Figure 5.28(a) shows that tire inside lateral groove wear increased as vehicle load increased (for constant speed, tire pressure, and front wheel total toe angle). It is also observed that tire travel life decreased and tire life reduction rate increased when vehicle load increased (Figure 5.28(b)). Figure 5.28(a) shows that the tire inside lateral groove wear went from 0.03 mm to 0.05 mm as the vehicle load went from 955 kg to 1075 kg (at a constant tire pressure of 40 psi and a front wheel total toe angle of 0.0°). For the same load condition (955 kg to 1075 kg), the tire's inside lateral groove wear rate increased by up to 66.67%. Figure 5.28(b) shows that tire traveling life decreased (from 123000 km to 73800 km) and tire traveling life reduction rate increased up to 40% when the vehicle load increased from 955 kg to 1075 kg.

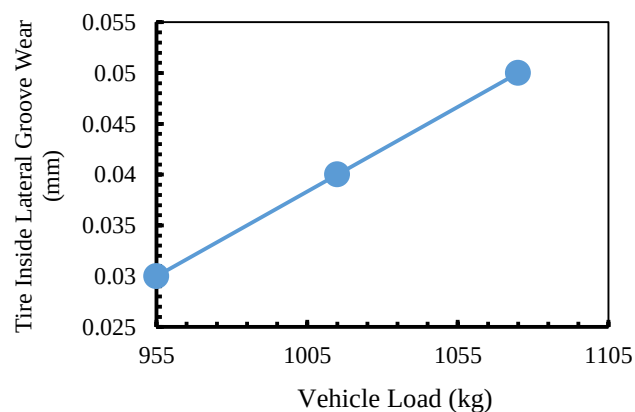


Figure 5.28(a) Variation of tire inside lateral groove wear (mm) with respect to vehicle load (kg).

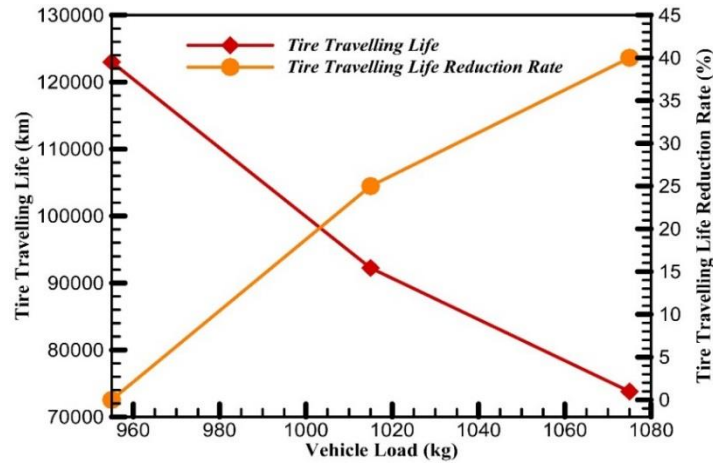


Figure 5.28(b) Change of tire travelling life (km) and tire travelling life reduction rate (%) with respect to vehicle load (kg).

5.19.2 on Travelling Cost

From Table 5.22 and Figure 5.29, it is shown that the traveling cost per km increased (from 0.0006\$ to 0.0672\$ or 0.0618Tk to 6.7195Tk) with the changes in front wheel total toe-in angle (from 0.00° to 4.80°), and their increasing rate is about 10774.95%. An increase in front wheel total toe-in angle can increase the tire wear of a light vehicle, as discussed in the above section, which increases travel costs.

Table 5.22 Experimental data for 6 test runs for different front wheel total toe-in angle and tire travelling cost

Front Wheel Total Toe-in Angle (deg.)	Tire Travelling Life, $(T_{tl}) = \frac{T_d \times T_{cgd}}{T_w}$ (km)	Tire Travelling Cost Per km $T_{tc/km} = \frac{T_p}{T_{tl}}$		Tire Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km(min)}}{T_{C/km(min)}}$
		USD	BDT	
0.00	92250.00	0.0006	0.0618	0.00
0.40	61500.00	0.0009	0.0927	50.00
1.00	46125.00	0.0012	0.1236	100.00
1.60	10542.86	0.0054	0.5407	775.00
4.20	1537.50	0.0371	3.7073	5900.00
4.80	848.28	0.0672	6.7195	10774.95

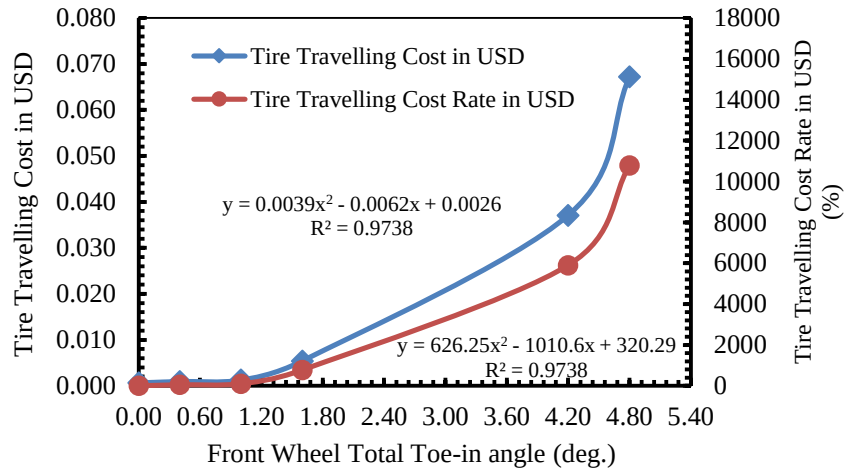


Figure 5.29 Change of tire travelling cost in USD and its rate in USD (%) with respect to front wheel total toe-in angle (deg.).

Similarly, the Pearson's correlation coefficient has verified the experimental result as presented in Table E.14 (Appendix E). The correlation coefficient, ($r_{at_{cgw}}$) is found to be 0.97 for the front wheel total toe-in angle and tire circumferential groove wear, which is within the range of $1 > r_{at_{cgw}} > 0.75$ [201]. It proves a very strong positive correlation between the front wheel total toe-in angle and tire wear of the light-duty vehicle.

5.19.3 Correlation Development

Based on experimental data as shown in Table 5.23 from Appendix Tables G.9(a), G.9(b), and G.9(c), ten different correlation models are developed using the regression method, which are: i) model 1- correlation between front wheel total toe-in angle and tire wear; ii) model 2- correlation between engine RPM and tire wear; iii) model 3- correlation between rolling resistance coefficient and tire wear; iv) model 4- correlation between rolling resistance and tire wear; v) model 5- correlation between energy consumption and tire wear; vi) model 6- correlation among front wheel total toe-in angle, engine RPM and tire wear; vii) model 7- correlation among front wheel total toe-in angle, rolling resistance coefficient and tire wear; viii) model 8- correlation among front wheel total toe-in angle, rolling resistance and tire wear; ix) model 9- correlation among front wheel total toe-in angle, energy consumption and tire wear; x) model 10- correlation among front wheel total toe-in angle, engine RPM, rolling resistance coefficient, rolling resistance, energy consumption, and tire wear.

Each regression model is subjected to a total of six tests. The R square, adjusted R square, and p -values of the corresponding regression model are shown in Table 5.23. This value is found to be 0.8930 in model 1. It can be said that the front

wheel's total toe-in angle explains 89.30% of the variation in the tire wear rate. It is observed that the values of R^2 and adjusted R^2 are 0.8930 and 0.8662, respectively. These values are very close, and the p -values are less than 0.05, indicating a perfect relationship between front wheel total toe-in angle and tire wear.

Similarly, models 2 to 10 show a trustworthy relationship between the dependent and independent variables that predict the tire wear rate. The correlation R -square values for models 2 to 10 are tending towards 1. Hence, the variance in tire wear can be explained by the developed regression model.

Table 5.23 Different simple and multiple correlation regression models for front wheel total toe-in angle for tire wear.

Model No.	Form of Equation	Equation constants		Multiple R	R Square	Adjusted R Square	Stand. Error	P-value
		m_i	k_i					
1	$(T_w)_{1,1} = m_1\alpha + k_1$	$m_1 = 0.8364$	-0.4596	0.9449	0.8930	0.8662	0.6538	0.004455166
2	$(T_w)_{2,1} = m_1R/M + k_1$	$m_1 = 0.0094$	-13.2077	0.9172	0.8414	0.8017	0.7960	0.009980273
3	$(T_w)_{3,1} = m_1\mu_{RRC} + k_1$	$m_1 = 112.6107$	-2.4333	0.9819	0.9643	0.9553	0.3776	0.000483476
4	$(T_w)_{4,1} = m_1F_R + k_1$	$m_1 = 0.01202$	-2.4338	0.9823	0.9649	0.9561	0.3743	0.000466939
5	$(T_w)_{5,1} = m_1E_c + k_1$	$m_1 = 0.0020$	-2.4338	0.9823	0.9649	0.9561	0.3743	0.00046694
6	$(T_w)_{6,1} = m_1\alpha + m_2^*R/M + k_1$	$m_1 = 1.6190$; -0.0092	12.0180	0.9535	0.9091	0.8486	0.6956	0.517901689
7	$(T_w)_{7,1} = m_1\alpha + m_2^*\mu_{RRC} + k_1$	$m_1 = -0.4279$; $m_2 = 166.9957$	-3.3386	0.9864	0.9731	0.9552	0.3782	0.058055175
8	$(T_w)_{8,1} = m_1\alpha + m_2^*F_R + k_1$	$m_1 = -0.4281$; $m_2 = 0.0178$	-3.3388	0.9868	0.9738	0.9563	0.3733	0.055698093
9	$(T_w)_{9,1} = m_1\alpha + m_2^*E_c + k_1$	$m_1 = -0.4281$; $m_2 = 0.0029$	-3.3388	0.9868	0.9738	0.9563	0.3733	0.055698133
10	$(T_w)_{10,1} = m_1\alpha + m_2R/M + m_3\mu_{RRC} + m_4F_R + m_5E_c + k_1$	$m_1 = -0.0201$; $m_2 = -0.0018$; $m_3 = -6142.6045$; $m_4 = 0$; $m_5 = -0.1116$	-0.2532	0.9929	0.9860	-0.0698	0.4724	#NUM!
	$T_w = -0.25322624 - 0.020158217^*(\alpha) - 0.001856967^*(R/M) - 6142.604588^*(\mu_{RRC}) + 0^*(F_R) + 0.111647121^*(E_c)$							

5.20 EFFECT OF FRONT WHEEL TOTAL TOE-OUT ANGLE ON TIRE WEAR AND TIRE LIFE

Similar test conditions are considered, as mentioned in Section 5.19. In this case, tire wear and life are estimated with the change of the vehicle's front wheel total toe-out angle from 0.00° to -4.80° . During the experiment, the weather temperature ranged from 25.1°C to 37.6°C , the humidity ranged from 50% to 86%, and the engine surface temperature ranged from 79.97°C to 85.56°C .

The experimental finding reveals that tire wear and tire traveling life reduction rates increased, and tire traveling life decreased considerably as the front wheel total toe-out angles increased. The rates of energy consumption and tire life reduction rate increased by up to 186.91% and 91.11%, respectively, when the front wheel total toe-out angle changed from 0.00° to -4.80° , as shown in Table D.16 and Table D.17 in Appendix D. Without misalignment of the front wheel total toe-out angle, the car tire traveling life is about 92250 km, where the tire wear increment with respect to a 0° alignment condition and tire wear rate is considered to be 0%, while with increases in the front wheel total toe-out angles to -4.80° , the car tire traveling life is reduced to approximately 823.81 km and the tire wear reduction rate is about 99.11%. Thereby, it can be seen that due to misalignment of the front wheel's total toe-out angle (from 0.00° to -4.80°), the car tire's traveling life is approximately 91426.19 km less for the same size tire.

In comparison to the front wheel total toe-in angle (4.80°), the tire traveling life of 24.47 km is less for the front wheel total toe-out angle (-4.80°), as shown in Figure 5.30. The deviation is most likely due to increased pressure on the rubber through the increased grip friction of the driven tires.

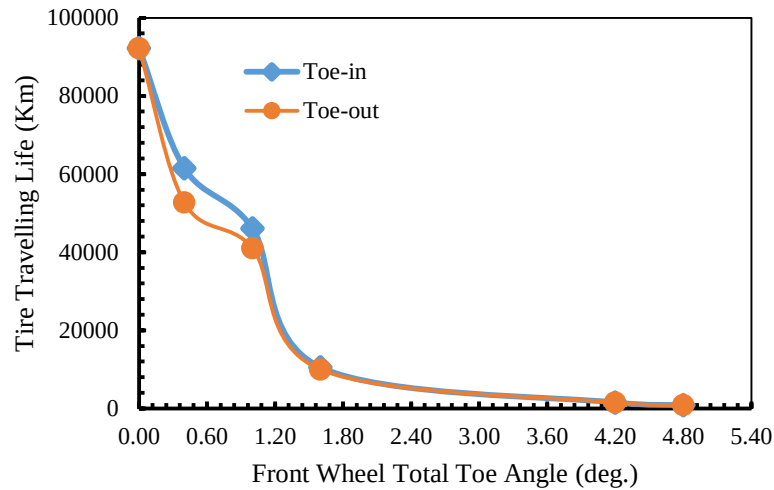


Figure 5.30 Comparison of tire travelling life with respect to front wheel total toe-in and toe-out angle.

Figures 5.31(a) and (b) showed the tire outside lateral groove wear, tire inside lateral groove wear, and tire circumferential groove wear rates as a function of the front wheel total toe-out angle (deg.). The results show that the tire outside lateral groove wear, tire inside lateral groove wear, and tire circumferential groove wear all changed when the front wheel total toe-out angle went from 0.00° to -4.80° (see Table D.17 in Appendix D). For the same misalignment condition, the rates of tire outside lateral groove wear, tire inside lateral groove wear, and tire circumferential groove wear are increased by 966.67%, 9766.67%, and 10650.0%, respectively, compared to a 0.00° alignment condition.

Toe-out can influence tire wear in a similar way as toe-in. The outside edges of tires may wear down more quickly than the inside edges when the front wheels of a vehicle are toe-out from the vehicle's centerline. This is because the outside edges of the tires are moving farther than the inside edges, which causes more friction and heat to be produced there. However, a slight toe-out is frequently recommended for rear-wheel-drive vehicles, as it can enhance turn-in and responsiveness. Figure 5.32 confirms the above explanation that when a car's total toe-in angle increases, tire wear from outside to inside steadily increases in the wheel, whereas when a car's general toe-out angle increases, tire wear from inside to outside gradually increases.

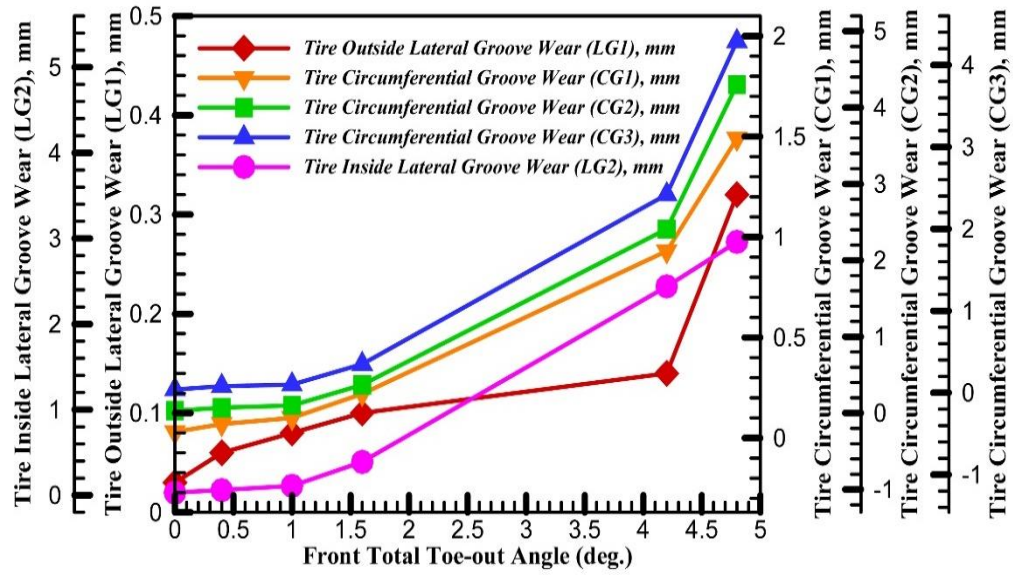


Figure 5.31(a) Variation of tire outside lateral groove wear (mm), tire inside lateral groove wear (mm) and tire circumferential groove wear (mm) with respect to front wheel total toe-out angle (deg.).

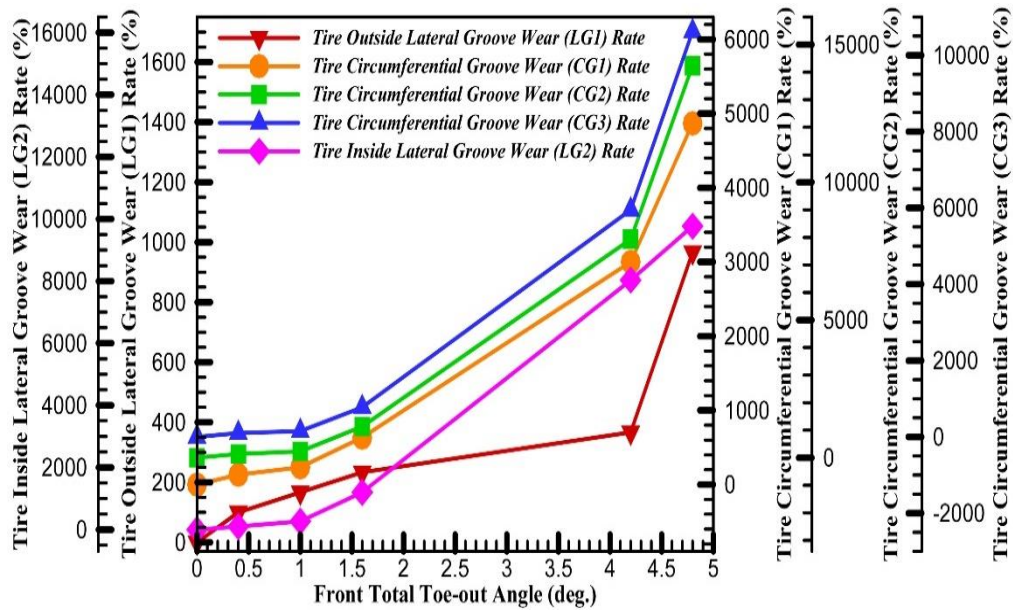


Figure 5.31(b) Change of tire outside lateral groove wear rate (%), tire inside lateral groove wear rate (%), and tire circumferential groove wear rate (%) with respect to front wheel total toe-out angle (deg.).



Figure 5.32 A photographic view of tire wear at front wheel total toe-in angle (4.20°) (left) and at front wheel total toe-out angle (-4.20°) (right).

A comparison of the traveling cost per km in USD of the light vehicle as a function of the front wheel total toe-in and toe-out angles is presented in Figure 5.33 from Table 5.22 and Appendix F.6. It is observed that the rate of travel showed very little change for the front wheel's total toe-out angle. When compared to the front wheel total toe-in angle, the rate of travel cost is about 0.02% higher at the front wheel total toe-out angle.

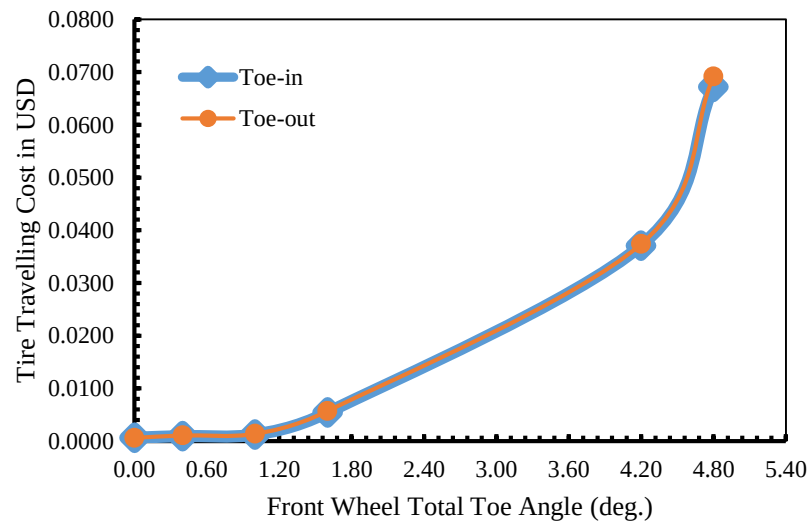


Figure 5.33 Comparison of tire travelling cost in USD with respect to front wheel total toe-in and toe-out angle (deg.).

The Pearson's correlation coefficient, ($r_{at_{cgw}}$) is found to be 0.97 for front wheel total toe-out angle and tire circumferential groove wear (Table E.15 (Appendix E)), which is within the range of $1 > r_{at_{cgw}} > 0.75$ [201], providing a very strong positive correlation between front wheel total toe-out angle and tire wear of the light-duty vehicle.

5.21 DEVELOPMENT OF A MATHEMATICAL MODEL FOR MULTIPLE CORRELATION

A mathematical model describes a system using mathematical concepts and language. The process of developing a mathematical model is termed mathematical modelling. Mathematical models are used in the natural sciences and engineering disciplines. A model may help explain a system, study the effects of different parameters, and make predictions about behavior. In this study, the main purpose of developing a mathematical model equation is to calculate the effects of various independent variables such as toe angle, tire pressure, vehicle load, and brake application on the dependent variable of fuel consumption.

The fuel performance of the Toyota Echo Plus-2ZZ-GE-02 light-duty vehicle has been investigated for the variation of toe angle, tire pressure, vehicle load, and brake application with fuel consumed. With the help of the Microsoft Office Excel data analysis tool, a multiple linear regression model analysis has been done using experimental data. Then, a mathematical model equation has been developed that shows the relation among fuel consumption, toe angle, tire pressure, vehicle load, and brake application [210,211]. The development of mathematical model equations is described below:

Let,

Y = dependent variable (fuel consumption) = F_c (ml)

X_1 = independent variable (toe angle) = α (deg.)

X_2 = independent variable (tire pressure) = T_p (psi)

X_3 = independent variable (vehicle load) = V_l (kg)

X_4 = independent variable (brake application) = B_a (times)

Thus, the multiple regression model equation has the following form from equation 3.20 of Section 3.7:

$$Y = a + b_1X_1 + b_2X_2 + b_3X_3 + b_4X_4 \quad (5.1)$$

By using the data analysis regression tool in Microsoft Office Excel, the values of the coefficients are obtained as shown in Table H.2 of Appendix H.

$$a = 168.649669210471$$

$$b_1 = 27.8667634831141$$

$$b_2 = -6.54987660648861$$

$$b_3 = 0.354622774434552$$

$$b_4 = 1.45664198796439$$

Putting the values of the coefficients in Equation (5.1), the Equation finally takes the following form:

$$\begin{aligned} Y &= 168.649669 + 27.866763 X_1 - 6.549876 X_2 + 0.354622 X_3 + 1.456641 X_4 \\ &= \text{Constant} + 27.866763 \times \text{Toe angle} - 6.549876 \times \text{Tire pressure} + 0.354622 \times \text{Vehicle} \\ &\quad \text{load} + 1.456641 \times \text{Brake application} \end{aligned} \quad (5.2)$$

Equation (5.2) shows the mathematical correlation among fuel consumption, toe angle, tire pressure, vehicle load, and brake application. It is found that toe angle, tire pressure, vehicle load, and brake application significantly affect fuel consumption. Using the equation, one can put in different independent variables and see what happens to the fuel consumption of a light-duty vehicle. For example, fuel consumption increased to 9.55% when the toe angle changed from 0.08° to 1.30°, while other parameters remained the same. Similarly, fuel consumption increased by 7.02% when tire pressure dropped from 32 to 28 psi, while all other variables stayed the same. When the vehicle's load climbed from 1089 kg to 1161 kg while keeping all other conditions constant, fuel consumption increased by 7.30%.

In the same way, by increasing the brake application from 5 times to 25 times while other parameters remained unchanged, fuel consumption increased up to 7.86%, as shown in Table H.1 in Appendix H. Thus, it may be concluded that toe angle, tire pressure, vehicle load, and brake application are the crucial parameters for light vehicle fuel performance. The developed multiple linear regression model is well suited to the previous model presented in Ref [206]. In Ref. [206], the authors have developed a multiple linear regression model equation, which is given below:

$$\text{Fuel Consumption (Y)} = 71.334 - 51.512 \times \text{tire pressure} + 0.2580 \times \text{vehicle load} + 0.0284 \times \text{brake application}$$

Chapter 6

Conclusions and Recommendations

6.1 CONCLUSIONS

According to the experimental results of two light vehicles (Toyota Echo Plus-2ZZ-GE-02 and Toyota Corolla-2E-86) based on front wheel toe alignment, tire pressure, vehicle loading, and brake application, the significant findings are as follows:

- According to the analysis, the vehicle's wheel started to become out of alignment when its travel distance increased over time. It has been observed that vehicle wheels became out of alignment as a result of changes in different factors, including vehicle load, road condition, brake application, tire pressure, suspension condition, etc.
- The engine rpm, engine rpm rate, rolling resistance, rolling resistance rate, fuel consumption, fuel consumption rate, traveling cost, and traveling cost rate are increased as the front wheel toe angle (such as front wheel left toe-in, left toe-out, total toe-in, and total toe-out) increases. The experimental results show that the front toe angle strongly correlates with rolling resistance. It has been found that the greatest increasing rate of rolling resistance is approximately 75.29%, 89.04%, 186.19%, and 197.94% with the change of the front wheel's left toe-in angle (0.00° to 2.53°), left toe-out angle (0.00° to -2.53°), total toe-in angle (0.00° to 5.06°), and total toe-out angle (0.00° to -5.06°), respectively. It has also been observed that due to misalignment of the front wheel toe angle, the car traveled about 4.77 km (left toe-in), 5.12 km (left toe-out), 7.37 km (total toe-in), and 7.63 km (total toe-out) less for the same amount of fuel, and the fuel consumption increment with respect to no misalignment condition is up to 38.22%, 42.24%, 73.99%, and 79.31%, respectively, of the car. The rate of travel cost per km is increased to 38.22% (left toe-in), 42.24% (left toe-out), 73.99% (total toe-in), and 79.31% (total toe-out). In comparison to the front wheel's left toe-in angle, the fuel consumption rate is 4.02% higher at the front wheel's left toe-out angle and 5.32% higher for the front wheel's total toe-out angle compared to the front wheel's total toe-in angle.

- It has been observed that the rolling resistance varies non-linearly with the tire inflation pressure under steady-state conditions (i.e., constant load, speed, and toe angle). In contrast, fuel consumption varies almost linearly. Interestingly, energy consumption also increases almost non-linearly with tire inflation pressure. The rates of engine rpm, rolling resistance, fuel consumption, and traveling cost per kilometer are increased up to 20.00%, 45.31%, 39.22%, and 39.22%, respectively, when the tire pressure changes from 42 psi to 22 psi at the front wheel toe angle of 0.00°. However, when the front wheel's total toe-in angle is 1.44°, their rates become 25.62%, 85.10%, 40.29%, and 40.29%, respectively. It has also been found that the rates of engine rpm, rolling resistance, fuel consumption, and traveling cost per kilometer are slightly higher at the front total toe-out angle (-1.44°). In comparison to the front wheel total toe angle of 0.00°, the fuel consumption rate of 4.69% is higher at the front wheel total toe-out angle of -1.44° and 3.62% is higher at the front wheel total toe-in angle of 1.44° when the tire pressure changes from 42 psi to 22 psi. Therefore, the travel cost for the total toe-out angle is higher compared to the total toe-in angle.
- It has been shown that the rate of engine rpm, rolling resistance, fuel consumption, and traveling cost per kilometer increase as the vehicle load increases. When the vehicle load increased from 1035 kg to 1377 kg at a front wheel toe angle of 0.00°, the rate of engine rpm, rolling resistance, fuel consumption, and traveling cost per kilometer increased up to 35.43%, 112.88%, 55.69%, and 55.69%, respectively. However, when the front wheel toe angle changes to 1.44° (toe-in) and -1.44° (toe-out) with a 1377 kg vehicle load, the rate of engine rpm, rolling resistance, fuel consumption, and traveling cost per kilometer are a little bit higher compared to the front wheel toe angle of 0.00°. It has also been observed that the front wheel's total toe-out angle of -1.44° consumed 3.35% more fuel, and the front wheel's total toe-in angle of 1.44° consumed 2.32% more fuel compared to a vehicle without wheel toe misalignment.
- According to the test results, increasing the number of brakes applied (0 to 100 times) resulted in increased rolling resistance and fuel consumption. The rates of rolling resistance coefficient and rolling resistance are increased up to 198.79% and 198.79%, respectively, and the rates of energy consumption and

fuel consumption are increased up to 198.79% and 81.74%, respectively, when the front wheel toe angle is 0.00° . Whereas changing the total toe angle of 1.44° (toe-in) and -1.44° (toe-out) with the same amount of brake application increases the rate of rolling resistance, fuel consumption, and travel cost per kilometer. It is found that the fuel consumption rate of 3.54% is higher at the front wheel total toe-out angle of -1.44° and 2.42% is higher at the front wheel total toe-in angle of 1.44° .

- It has been observed that the tire circumferential grooves' wear increases as the wheel front total toe-in angle increases. It can be seen that tire outside lateral groove wear, tire inside lateral groove wear, and tire circumferential groove wear changed (from 0.03 mm to 3.01 mm, 0.03 mm to 0.29 mm, and 0.04 mm to 4.35 mm) with the changes in the front wheel total toe-in angle (from 0.00° to 4.80°). Their rates increased to 9933.33%, 866.67%, and 10775.0%, respectively, when the front wheel total toe-in angle increased from 0.00° to 4.80° . The tire traveling life of a light vehicle is found to be about 92250 km at a front wheel total toe-in angle of 0.00° , with a tire wear increment rate of 0% with respect to no misalignment condition, while at a front wheel total toe-in angle of 4.80° , the tire traveling life is found to be nearly 848.28 km, with a tire traveling life reduction rate of approximately 99.08%.
- For front wheel total toe-in angles between 0.00° and 4.80° , the result is an increase in engine rpm, rolling resistance, tire circumferential groove wear, and tire traveling life reductions up to 33.59%, 167.84%, 10775.00%, and 99.08%, respectively. The tire traveling life of a light vehicle is found to be about 92250 km at a front wheel total toe-in angle of 0.00° , with a tire wear increment rate of 0% with respect to no misalignment condition, while at a front wheel total toe-in angle of 4.80° , the tire traveling life is found to be nearly 848.28 km, with a tire traveling life reduction rate of approximately 99.08%. On the other hand, due to the front wheel total toe-out angle of -4.80° , tire circumferential groove wear, and tire traveling life reductions of up to 10650.00% and 99.11%, respectively, In comparison to the front wheel total toe-in angle (4.80°), the tire traveling life of 24.47 km is less at the front wheel total toe-out angle (-4.80°). When compared to the front wheel total toe-in angle, the rate of travel cost was about 0.02% higher at the front wheel total toe-out angle.

- The fuel consumption rate is somewhat higher for the front wheel toe-out angle because the toe-out produces more rolling resistance compared to the front wheel toe-in angle. The deviation is most likely due to increased pressure on the rubber from the increased grip friction of the driven tires.
- It has been found that the proposed regression models have maintained a good correlation between dependent variables and independent variables with the experimental data. The correlated R-square values for the models tend towards 1, implying that the developed regression model can explain the variance in fuel consumption and tire wear.

Based on the observations and experimental results, it can be concluded that for light vehicles, proper front wheel alignment, especially toe angle and the manufacturer's standard tire pressure, a proper vehicle load, and less brake application can significantly improve fuel economy, reduce tire wear, reduce maintenance costs, and make passengers safer and more comfortable.

6.2 RECOMMENDATIONS

Future research efforts might be made to efficiently comprehend and improve vehicle fuel performance and tire life. The following recommendations are made for future work:

- Further experiments can be performed to investigate the effect of tire pressure on suspension components' life in the future.
- In this thesis, one can figure out how the car's fuel consumption changed by treating variations in tire pressure, vehicle load, and brake application as a 0.00° alignment condition and a front wheel total toe-in angle of 1.44° and a front wheel total toe-out angle of -1.44° as misalignment. However, the difference in fuel consumption can be examined in the future by adjusting the value of the front wheel total toe-in and toe-out angle for other ranges, travel distance, and running the vehicle at a higher speed.
- In this study, the front wheel toe-in setting is adjusted in a static loading configuration; however, the effect on the dynamic toe-in angle or toe-in alteration is yet to be developed through a proper experimental setup.

- The experiments can be performed in a controlled environment. The proper relationship between slip /toe angle can be generated using a drum or flat track test system.

This study determined the rate of change in vehicle fuel consumption and tire life by maintaining alignment and misalignment conditions in two model cars. Furthermore, maintaining the alignment and misalignment of the other model car may be considered.

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Appendices

APPENDIX-A: LIST OF EXPERIMENTAL APPARATUS



Measurement range: 0-150mm, 0-6", Made in China

Figure A.1 Digital Vernier Caliper.



Measurement range: Tire tread depth up to 9/16" in graduation of 1/64", Made in Taiwan

Figure A.2 Tire Tread & Brake Lining Gauge



Measurement range: 0-7.5m, Made in China

Figure A.3 Measuring Tape.



Measurement range: 0 to 600mm, Made in China

Figure A.4 Vernier Caliper.



Measurement range: Maximum pressure =10bar, Tank volume = 500m³

Figure A.5 Air Compressor.



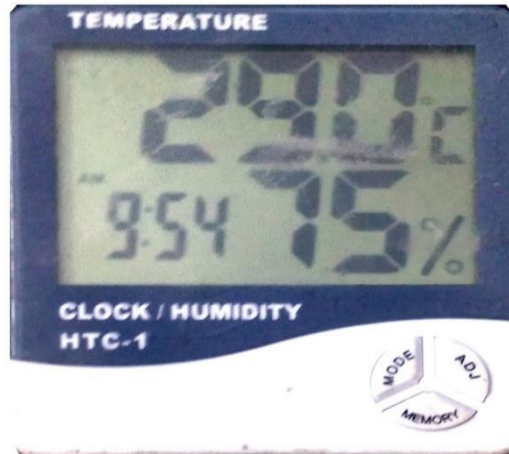
Measurement range: 0 to 16 bar/0 to 220psi, Made in China

Figure A.6 Tire Pressure Gauge.



Measurement range: Load capacity = 3 ton, Ingco brand, Made in china

Figure A.7 Hydraulic Jack.



Measurement range: Temperature = $-50^{\circ}\text{C}/+70^{\circ}\text{C}$, Humidity = $+10\%$ to $+99\%$, Temperature accuracy = $\pm 1^{\circ}$, Humidity accuracy = $\pm 5\%$, Display resolution = $0.1/1\%$, Made in China.

Figure A.8 Digital Temperature and Humidity Meter.



Measurement range: 0 to 7000 rpm, Made in China.

Figure A.9 Engine RPM Meter (Tachometer)



Measurement range: 0-25mm/0-1", Resolution = $0.0005''/0.01\text{mm}$, Code-1145-25, Insize brand, Made in Germany.

Figure A.10 Electronic Digital Tire Tread Depth Gauge.



Measurement range:-50°C to 300°C, Accuracy = 0.1°C, Model = SH-113V1, Made in China

Figure A.11 Digital Thermometer.

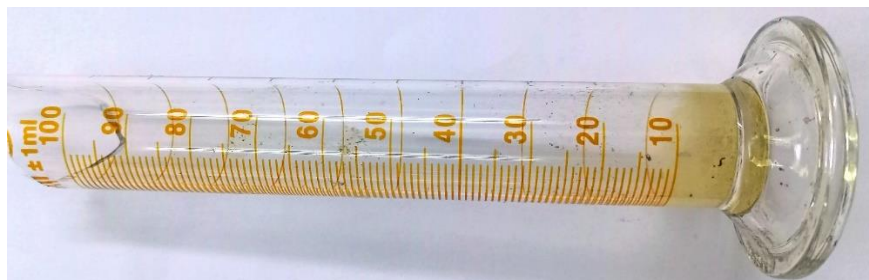


Measurement range: 19-inch

Figure A.12 Wheel Socket Wrench.



Figure A.13 Oil Funnel.



Measurement range: 0 to 100ml

Figure A.14 Oil Measurement Biker.



Measurement range: 0 to 1000ml

Figure A.15 Oil Measurement Biker.



Figure A.16 Open end ring spanner-14inch.



Figure A.17 Adjustable Wrench-12inch.



Measurement range: Caster $\pm 30^{\circ}$, Camber $\pm 10^{\circ}$, Toe $\pm 12^{\circ}$ ($\pm 150\text{mm}$), Set back $\pm 22^{\circ}$, Thrust $\pm 22^{\circ}$, S.A.I $\pm 30^{\circ}$

Figure A.18 Wheel Alignment Machine (BEST 5800).



Measurement range: 0 to 60kg, Mega brand, Made in China

Figure A.19 Load Scale.



DOT, 20E9, EEC, 2721, SNI, (83T), (E4), 027277, S2WR2, 0285667

Tubeless Radial, Max. Load - 487kg (1074lb), Max. Pressure - 44psi (300 kpi), MA-P3-04, Tread: 1 Polyester + 2 Steel + 1 Nylon, Sidewall: 1 Polyester, Treadware 440, Traction - A, Temperature – A, Made in Thailand

Figure A.20 New Tire.

APPENDIX-B: THE MATHEMATICAL RELATION FOR EXPERIMENTAL DATA.

B.1: Fuel consumption and Rolling Resistance Force relation by the Lagrange's interpolation formula (unequal interval) [195].

$$\begin{aligned}
 \int(x) &= \frac{(x-x_1)(x-x_2)(x-x_3)(x-x_4)}{(x_0-x_1)(x_0-x_2)(x_0-x_3)(x_0-x_4)} Y_0 \\
 &+ \frac{(x-x_0)(x-x_2)(x-x_3)(x-x_4)}{(x_1-x_0)(x_1-x_2)(x_1-x_3)(x_1-x_4)} Y_1 \\
 &+ \frac{(x-x_0)(x-x_1)(x-x_3)(x-x_4)}{(x_2-x_0)(x_2-x_1)(x_2-x_3)(x_2-x_4)} Y_2 \\
 &+ \frac{(x-x_0)(x-x_1)(x-x_2)(x-x_4)}{(x_3-x_0)(x_3-x_1)(x_3-x_2)(x_3-x_4)} Y_3 \\
 &+ \frac{(x-x_0)(x-x_1)(x-x_2)(x-x_3)}{(x_4-x_0)(x_4-x_1)(x_4-x_2)(x_4-x_3)} Y_4 \\
 &= \frac{(x-388)(x-449)(x-523)(x-602)}{(346-388)(346-449)(346-523)(346-602)} \times 224.551 \\
 &+ \frac{(x-346)(x-449)(x-523)(x-602)}{(388-346)(388-449)(388-523)(388-602)} \times 241.660 \\
 &+ \frac{(x-346)(x-388)(x-523)(x-602)}{(449-346)(449-388)(449-523)(449-602)} \times 328.272 \\
 &+ \frac{(x-346)(x-388)(x-449)(x-602)}{(523-346)(523-388)(523-449)(523-602)} \times 487.596 \\
 &+ \frac{(x-346)(x-388)(x-449)(x-523)}{(602-346)(602-388)(602-449)(602-523)} \times 642.643 \\
 &= -0.0000000248x^4 + 0.0000174535x^3 + 0.0123884602x^2 - \\
 &10.8264074438x + 2119.8776237218
 \end{aligned}$$

B.2: Relation between engine RPM and fuel consumption using by Lagrange's interpolation formula (unequal interval) [195].

$$\begin{aligned}
Y = \int (x) &= \frac{(x - x_1)(x - x_2)(x - x_3)(x - x_4)}{(x_0 - x_1)(x_0 - x_2)(x_0 - x_3)(x_0 - x_4)} Y_0 \\
&+ \frac{(x - x_0)(x - x_2)(x - x_3)(x - x_4)}{(x_1 - x_0)(x_1 - x_2)(x_1 - x_3)(x_1 - x_4)} Y_1 \\
&+ \frac{(x - x_0)(x - x_1)(x - x_3)(x - x_4)}{(x_2 - x_0)(x_2 - x_1)(x_2 - x_3)(x_2 - x_4)} Y_2 \\
&+ \frac{(x - x_0)(x - x_1)(x - x_2)(x - x_4)}{(x_3 - x_0)(x_3 - x_1)(x_3 - x_2)(x_3 - x_4)} Y_3 \\
&+ \frac{(x - x_0)(x - x_1)(x - x_2)(x - x_3)}{(x_4 - x_0)(x_4 - x_1)(x_4 - x_2)(x_4 - x_3)} Y_4 \\
&= \left[\frac{(x - 348)(x - 426)(x - 490)(x - 624)}{(306 - 348)(306 - 426)(306 - 490)(306 - 624)} \times 1000 \right] \\
&+ \left[\frac{(x - 306)(x - 426)(x - 490)(x - 624)}{(348 - 306)(348 - 426)(348 - 490)(348 - 624)} \times 1090 \right] \\
&+ \left[\frac{(x - 306)(x - 348)(x - 490)(x - 624)}{(426 - 306)(426 - 348)(426 - 490)(426 - 624)} \times 1200 \right] \\
&+ \left[\frac{(x - 306)(x - 348)(x - 426)(x - 624)}{(490 - 306)(490 - 348)(490 - 426)(490 - 624)} \times 1370 \right] \\
&+ \left[\frac{(x - 306)(x - 348)(x - 426)(x - 490)}{(624 - 306)(624 - 348)(624 - 426)(624 - 490)} \times 1510 \right] \\
&= -0.00000044701x^4 + 0.00078267424x^3 - 0.502140936x^2 \\
&+ 141.90424314351x - 13910.64619583585
\end{aligned}$$

B.3 Relation between toe angle and rolling resistance coefficient [144].

<u>Positive slip angle /Toe angle</u>	<u>Rolling resistance coefficient</u>
(x) (α) (deg.)	(y) (μ_{RRC})
0	0.020
1	0.024
2	0.028
3	0.035
4	0.045
5	0.058
6	0.077

Least Square Method:

The equation of second degree parabola is given as below:

$$y = a + bx + Cx^2$$

And the normal equations are

$$\sum y = an + b \sum x + C \sum x^2 \dots\dots\dots 1$$

$$\sum xy = a \sum x + b \sum x^2 + C \sum x^3 \dots\dots\dots 2$$

$$\sum x^2y = a \sum x^2 + b \sum x^3 + C \sum x^4 \dots\dots\dots 3$$

Here, $n = 7, \sum x = 21, \sum y = 0.287, \sum x^2 = 91, \sum x^3 = 441, \sum x^4 = 2275, \sum xy = 1.117, \sum x^2y = 5.393$

Putting all these values in equation 1, 2 and 3 we get solving these equations such as,

$$a = 0.021, b = 0.000143, c = 0.0015$$

Therefore, the required polynomial is $y = a + bx + cx^2$

$$= 0.021 + 0.000143x + 0.0015x^2$$

Then, the equation can be written as below as form:

$$\mu_{RRC} = 0.021 + 0.000143\alpha + 0.0015\alpha^2$$

APPENDIX-C: CALCULATION FOR FUEL TRAVELLING COST, TIRE WEAR AND TIRE TRAVELLING COST

C.1: Calculation for fuel cost

Travelling cost per kilometer = Amount of used fuel/ Travelling distance * Fuel price

C.2: Calculation for tire travelling life and cost

Tire travelling life = Travelling distance* Tire tread depth/ Tire wear

Tire travelling cost per kilometer = Tire price/ Tire travelling life

Table C.3 Calculation for tire wear measurement for front total toe-in angle.

Tire wear measurement	Front total toe-in angle (deg.)	Load (kg)	Outside lateral groove reading			Circumferential grooves reading												Inside lateral groove reading		
			Initial	Final	Wear (mm)	Initial	Final	Wear (mm)	Initial	Final	Wear (mm)	Initial	Final	Wear (mm)	Initial	Final	Wear (mm)	Initial	Final	Wear (mm)
Point - 1 Test - 1	0.00	955	6.06	6.03	0.03	7.38	7.34	0.04	7.38	7.34	0.04	7.38	7.34	0.04	7.38	7.34	0.04	6.06	6.03	0.03
Point - 1 Test - 2		1015	6.03	5.99	0.04	7.34	7.29	0.05	7.34	7.30	0.04	7.34	7.30	0.04	7.34	7.29	0.05	6.03	5.99	0.04
Point - 1 Test - 3		1075	5.99	5.95	0.04	7.29	7.24	0.05	7.30	7.25	0.05	7.29	7.24	0.05	7.29	7.24	0.05	5.99	5.94	0.05
Point - 1 Test - 4	0.40	955	5.95	5.90	0.05	7.24	7.18	0.06	7.25	7.18	0.07	7.24	7.18	0.06	7.24	7.18	0.06	5.94	5.89	0.05
Point - 1 Test - 5	1.00	955	5.90	5.81	0.09	7.18	7.10	0.08	7.18	7.10	0.08	7.18	7.10	0.08	7.18	7.09	0.09	5.89	5.82	0.07
Point - 1 Test - 6	1.60	955	5.81	5.44	0.37	7.10	6.75	0.35	7.10	6.76	0.34	7.09	6.74	0.35	7.10	6.90	0.20	5.82	5.73	0.09
Point - 1 Test - 7	4.20	955	5.44	3.01	2.43	6.75	4.35	2.40	6.76	4.35	2.41	6.74	4.34	2.40	6.90	6.00	0.90	5.73	5.60	0.13
Point - 1 Test - 8	4.80	955	3.01	0.00	3.01	4.35	0.00	4.35	4.35	0.00	4.35	4.34	0.00	4.34	6.00	4.48	1.52	5.60	5.31	0.29

APPENDIX-D: EXPERIMENTAL DATA

Table D.1 Experimental data for 20 test runs for different front wheel left toe-in angles.

Weather temperature = (20.06 °C to 37.63°C), Humidity = (29.00% to 85.67%), Engine surface temperature= (77.97°C to 82.73°C)

Front Wheel Left Toe-in Angle α (deg.)	Angle Rate, $\dot{\alpha} = \frac{T_d}{t \times V_s}$ (deg/sec)[200]	Engine RPM (Appendix-B.2) [195]	Engine RPM Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance Coefficient (μ_{RRC}) (Appendix B.3) [144]	Rolling Resistance Coefficient Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance $F_R = \mu_{RRC} \times V_w$, (N) (Appendix-B.1) [195]	Rolling Resistance Rate (%) w.r.t. 0° Alignment Condition	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment Condition	Fuel Consumption (ml)	Fuel Consumption Rate (%) w.r.t. 0° Alignment Condition	(KPL) _{ideal}	KPL Reduction Rate (%)
0.00	0.00	1090	0.00	0.0210	0.00	224.4223	0.00	1346.5340	0.00	348	0.00	17.241	0.00
0.08	0.01	1095	0.46	0.0210	0.01	224.4480	0.01	1346.6878	0.01	352	1.15	17.045	1.14
0.20	0.03	1100	0.92	0.0210	0.20	224.8698	0.20	1349.2189	0.20	356	2.30	16.854	2.25
0.35	0.05	1109	1.74	0.0212	0.94	226.5410	0.94	1359.2459	0.94	363	4.31	16.529	4.13
0.50	0.07	1117	2.48	0.0214	1.99	228.8943	1.99	1373.3656	1.99	369	6.03	16.260	5.69
0.65	0.09	1124	3.12	0.0217	3.41	232.0718	3.41	1392.4306	3.41	375	7.76	16.000	7.20
0.80	0.11	1135	4.13	0.0223	6.20	238.3282	6.20	1429.9689	6.20	384	10.34	15.625	9.37
0.95	0.13	1145	5.05	0.0229	9.31	245.3272	9.31	1471.9633	9.31	392	12.64	15.306	11.22
1.10	0.15	1153	5.78	0.0235	12.03	251.4241	12.03	1508.5444	12.03	398	14.37	15.075	12.56
1.25	0.17	1164	6.79	0.0243	15.59	259.4138	15.59	1556.4828	15.59	405	16.38	14.815	14.07
1.40	0.19	1178	8.07	0.0253	20.76	271.0025	20.76	1626.0152	20.76	414	18.97	14.493	15.94
1.55	0.21	1190	9.17	0.0263	25.20	280.9800	25.20	1685.8803	25.20	421	20.98	14.252	17.34
1.70	0.23	1204	10.46	0.0273	30.00	291.7472	30.00	1750.4832	30.00	428	22.99	14.019	18.69
1.84	0.25	1221	12.02	0.0285	35.88	304.9542	35.88	1829.7255	35.88	436	25.29	13.761	20.18
2.00	0.28	1237	13.49	0.0297	41.36	317.2408	41.36	1903.4447	41.36	443	27.30	13.544	21.44
2.14	0.30	1256	15.23	0.0311	47.96	332.0451	47.96	1992.2703	47.96	451	29.60	13.304	22.84
2.25	0.31	1277	17.16	0.0325	54.88	347.5835	54.88	2085.5007	54.88	459	31.90	13.072	24.18
2.36	0.33	1300	19.27	0.0340	62.09	363.7709	62.09	2182.6257	62.09	467	34.20	12.848	25.48
2.43	0.34	1317	20.83	0.0352	67.67	376.2850	67.67	2257.7100	67.67	473	35.92	12.685	26.43
2.53	0.35	1342	23.12	0.0368	75.29	393.3964	75.29	2360.3786	75.29	481	38.22	12.474	27.65

Table D.2 Experimental data for 20 test runs for different front wheel left toe-out angles.
Weather temperature = (21.70 °C to 32.27°C), Humidity = (52.00% to 86.67%), Engine surface temperature= (79.83°C to 82.47°C)

Front Wheel Left Toe-out Angle (deg.)	Angle Rate, $\dot{\alpha}_t = \frac{T_d}{V_s}$ (deg/sec)	Engine RPM	Engine RPM Rate (%) w.r.t. 0°	Alignment Condition	Rolling Resistance Coefficient (μ_{RC})	Rolling Resistance Coefficient Rate (%) w.r.t. 0° Alignment	Rolling Resistance $F_R = \mu_{RC} \times V_w$, (N)	Rolling Resistance Rate (%) w.r.t. 0° Alignment	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment	Fuel Consumption (ml)	Fuel Consumption Rate (%) w.r.t. 0° Alignment	(KPL) _{ideal}	KPL Reduction Rate (%)
0.00	0.00	1090	0.00	0.00	0.0210	0.00	224.4223	0.00	1346.5340	0.00	348	0.00	17.241	0.00
-0.08	0.01	1097	0.64	0.04	0.0210	0.04	224.5165	0.04	1347.0993	0.04	353	1.44	16.997	1.42
-0.20	0.03	1106	1.47	0.06	0.0211	0.06	225.6811	0.06	1354.0867	0.06	360	3.45	16.666	3.34
-0.35	0.05	1115	2.29	0.08	0.0214	0.08	228.4441	0.08	1370.6648	0.08	368	5.75	16.304	5.43
-0.50	0.07	1125	3.21	0.10	0.0218	0.10	232.6797	0.10	1396.0784	0.10	376	8.05	15.957	7.45
-0.65	0.09	1131	3.76	0.12	0.0221	0.12	236.0483	0.12	1416.2900	0.12	381	9.48	15.748	8.66
-0.80	0.11	1142	4.77	0.16	0.0228	0.16	243.4543	0.16	1460.7259	0.16	390	12.07	15.384	10.77
-0.95	0.13	1152	5.69	0.20	0.0234	0.20	250.3589	0.20	1502.1533	0.20	397	14.08	15.113	12.34
-1.10	0.15	1162	6.61	0.24	0.0241	0.24	258.2162	0.24	1549.2972	0.24	404	16.09	14.851	13.86
-1.25	0.17	1175	7.80	0.28	0.0251	0.28	268.3038	0.28	1609.8228	0.28	412	18.39	14.563	15.53
-1.40	0.19	1185	8.72	0.32	0.0259	0.32	276.6044	0.32	1659.6262	0.32	418	20.11	14.354	16.74
-1.55	0.21	1202	10.28	0.36	0.0271	0.36	290.1626	0.36	1740.9755	0.36	427	22.70	14.051	18.50
-1.70	0.23	1219	11.83	0.40	0.0284	0.40	303.2532	0.40	1819.5194	0.40	435	25.00	13.793	20.00
-1.84	0.25	1237	13.49	0.44	0.0297	0.44	317.2408	0.44	1903.4447	0.44	443	27.30	13.544	21.44
-2.00	0.28	1259	15.50	0.48	0.0312	0.48	333.9489	0.48	2003.6935	0.48	452	29.89	13.274	23.01
-2.14	0.30	1277	17.16	0.52	0.0325	0.52	347.5835	0.52	2085.5007	0.52	459	31.90	13.071	24.19
-2.25	0.31	1300	19.27	0.56	0.0340	0.56	363.7709	0.56	2182.6257	0.56	467	34.20	12.847	25.49
-2.36	0.33	1323	21.38	0.60	0.0356	0.60	380.5201	0.60	2283.1203	0.60	475	36.49	12.631	26.74
-2.43	0.34	1351	23.94	0.64	0.0374	0.64	399.9220	0.64	2399.5323	0.64	484	39.08	12.396	28.10
-2.53	0.35	1386	27.16	0.68	0.0397	0.68	424.2538	0.68	2545.5227	0.68	495	42.24	12.121	29.70

Table D.3 Experimental data for 20 test runs for different front wheel total toe-in angles.
Weather temperature = (23.43°C to 36.40°C), Humidity = (25.33% to 78.67%), Engine surface temperature= (80.30°C to 81.83°C)

Front Wheel Total Toe-in Angle (deg.)	Angle Rate, $Ar = \frac{a_t \times V_s}{T_d}$ (deg/sec)	Engine RPM	Engine RPM Rate (%)	Condition w.r.t. 0° Alignment	Rolling Resistance Coefficient (μ_{RC})	Rolling Resistance Coefficient Rate (%)	Rolling Resistance $F_R = \mu_{RC} \times V_w$, (N)	Rolling Resistance Rate (%) w.r.t. 0° Alignment	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment	Fuel Consumption (ml)	Fuel Consumption Rate (%) w.r.t. 0° Alignment	(KPL) $_{ideal}$	KPL Reduction Rate (%)
0.00	0.00	1087	0.00	0.00	0.0210	0.00	224.5509	0.00	1347.3054	0.00	346	0.00	17.341	0.00
0.16	0.02	1097	0.92	0.48	0.0211	0.48	225.6202	0.48	1353.7211	0.48	353	2.02	16.997	1.98
0.40	0.06	1113	2.39	1.43	0.0213	1.43	227.7588	1.43	1366.5526	1.43	366	5.78	16.393	5.47
0.70	0.10	1126	3.59	3.81	0.0218	3.81	233.1052	3.81	1398.6313	3.81	377	8.96	15.915	8.22
1.00	0.14	1140	4.88	7.62	0.0226	7.62	241.6595	7.62	1449.9572	7.62	388	12.14	15.463	10.83
1.30	0.18	1153	6.07	12.86	0.0237	12.86	253.4217	12.86	1520.5304	12.86	398	15.03	15.075	13.07
1.60	0.22	1171	7.73	19.52	0.0251	19.52	268.3918	19.52	1610.3507	19.52	410	18.50	14.634	15.61
1.90	0.26	1192	9.66	27.14	0.0267	27.14	285.5004	27.14	1713.0026	27.14	422	21.97	14.218	18.01
2.20	0.30	1219	12.14	36.19	0.0286	36.19	305.8169	36.19	1834.9016	36.19	435	25.72	13.793	20.46
2.50	0.35	1251	15.09	46.19	0.0307	46.19	328.2720	46.19	1969.6322	46.19	449	29.77	13.363	22.94
2.80	0.39	1285	18.22	58.10	0.0332	58.10	355.0043	58.10	2130.0257	58.10	462	33.53	12.987	25.11
3.10	0.43	1329	22.26	70.95	0.0359	70.95	383.8751	70.95	2303.2507	70.95	477	37.86	12.578	27.47
3.40	0.47	1370	26.03	84.76	0.0388	84.76	414.8845	84.76	2489.3071	84.76	490	41.62	12.244	29.39
3.68	0.51	1425	31.09	99.05	0.0418	99.05	446.9632	99.05	2681.7793	99.05	507	46.53	11.834	31.76
4.00	0.55	1477	35.88	117.14	0.0456	117.14	487.5962	117.14	2925.5774	117.14	523	51.16	11.472	33.84
4.28	0.59	1524	40.20	133.81	0.0491	133.81	525.0214	133.81	3150.1283	133.81	539	55.78	11.131	35.81
4.50	0.62	1561	43.61	147.62	0.0520	147.62	556.0308	147.62	3336.1848	147.62	554	60.12	10.83	37.55
4.72	0.65	1609	48.02	162.38	0.0551	162.38	589.1788	162.38	3535.0727	162.38	571	65.03	10.507	39.41
4.86	0.67	1654	52.16	171.90	0.0571	171.90	610.5646	171.90	3663.3875	171.90	587	69.65	10.221	41.06
5.06	0.70	1694	55.84	186.19	0.0601	186.19	642.6433	186.19	3855.8597	186.19	602	73.99	9.966	42.53

Table D.4 Experimental data for 20 test runs for different front wheel total toe-out angles.
Weather temperature = (22.40°C to 33.40°C), Humidity = (40.67% to 68.33%), Engine surface temperature= (80.30°C to 82.23°C)

Front Wheel Total Toe-out Angle (deg.)	Angle Rate, $\dot{\alpha} = \frac{\alpha \times V_s}{T_d}$ (deg/sec)	Engine RPM	Engine RPM Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance Coefficient (μ_{RRC})	Rolling Resistance Coefficient Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance $F_R = \mu_{RRC} \times V_w$, (N)	Rolling Resistance Rate (%) w.r.t. 0° Alignment Condition	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment Condition	Fuel Consumption (ml)	Fuel Consumption Rate (%) w.r.t. 0° Alignment Condition	(KPL) $_{ideal}$	KPL Reduction Rate (%)
0.00	0.00	1090	0.00	0.0210	0.00	224.4223	0.00	1346.5340	0.00	348	0.00	17.241	0.00
-0.16	0.02	1103	1.19	0.0211	0.36	225.2272	0.36	1351.3633	0.36	358	2.87	16.759	2.80
-0.40	0.06	1120	2.75	0.0215	2.66	230.3815	2.66	1382.2893	2.66	372	6.90	16.129	6.45
-0.70	0.10	1135	4.13	0.0223	6.20	238.3282	6.20	1429.9689	6.20	384	10.34	15.625	9.37
-1.00	0.14	1148	5.32	0.0231	10.19	247.2805	10.19	1483.6829	10.19	394	13.22	15.228	11.68
-1.30	0.18	1161	6.51	0.0240	14.53	257.0371	14.53	1542.2227	14.53	403	15.80	14.888	13.65
-1.60	0.22	1180	8.26	0.0255	21.37	272.3777	21.37	1634.2660	21.37	415	19.25	14.457	16.15
-1.90	0.26	1200	10.09	0.0270	28.59	288.5932	28.59	1731.5593	28.59	426	22.41	14.084	18.31
-2.20	0.30	1227	12.57	0.0290	38.19	310.1398	38.19	1860.8391	38.19	439	26.15	13.667	20.73
-2.50	0.35	1261	15.69	0.0314	49.66	335.8641	49.66	2015.1846	49.66	453	30.17	13.245	23.18
-2.80	0.39	1294	18.72	0.0336	60.26	359.6680	60.26	2158.0078	60.26	465	33.62	12.903	25.16
-3.10	0.43	1338	22.75	0.0366	74.33	391.2336	74.33	2347.4015	74.33	480	37.93	12.5	27.50
-3.40	0.47	1380	26.61	0.0393	87.05	419.7892	87.05	2518.7354	87.05	493	41.67	12.17	29.41
-3.68	0.51	1438	31.93	0.0431	105.14	460.3729	105.14	2762.2372	105.14	511	46.84	11.741	31.90
-4.00	0.55	1492	36.88	0.0467	122.30	498.8896	122.30	2993.3374	122.30	528	51.72	11.363	34.09
-4.28	0.59	1543	41.56	0.0504	140.05	538.7328	140.05	3232.3966	140.05	546	56.90	10.989	36.26
-4.50	0.62	1594	46.24	0.0539	156.83	576.3790	156.83	3458.2739	156.83	564	62.07	10.638	38.30
-4.72	0.65	1642	50.64	0.0569	171.24	608.7314	171.24	3652.3886	171.24	581	66.95	10.327	40.10
-4.86	0.67	1698	55.78	0.0600	185.71	641.2073	185.71	3847.2441	185.71	601	72.70	9.983	42.10
-5.06	0.70	1763	61.74	0.0625	197.94	668.6372	197.94	4011.8235	197.94	624	79.31	9.615	44.23

Table D.5 Experimental data for 21 test runs with the variation of tire pressure at 0.00° of front wheel total toe angle.
Weather temperature = (19.47 °C to 28.03°C), Humidity = (64.00% to 85.00%), Engine surface temperature= (77.17°C to 82.17°C)

Tire Pressure (psi)	Tire Pressure Variation	Engine RPM	Engine RPM Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance Coefficient (μ_{RC})	Rolling Resistance Coefficient Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance $F_R = \mu_{RC} \times V_w$, (N)	Rolling Resistance Rate (%) w.r.t. 0° Alignment Condition	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment Condition	Fuel Consumption (ml)	Fuel Consumption Rate (%) w.r.t. 0° Alignment Condition	(KPL) $_{ideal}$	KPL Reduction Rate (%)
22	-10	1200	20.00	0.0270	45.31	288.5932	45.31	1731.5593	45.31	426	39.22	14.084	28.17
23	-9	1178	17.80	0.0253	36.46	271.0025	36.46	1626.0152	36.46	414	35.29	14.492	26.09
24	-8	1164	16.40	0.0243	30.62	259.4138	30.62	1556.4828	30.62	405	32.35	14.814	24.45
25	-7	1149	14.90	0.0232	25.02	248.2870	25.02	1489.7217	25.02	395	29.08	15.189	22.53
26	-6	1135	13.50	0.0223	20.00	238.3282	20.00	1429.9689	20.00	384	25.49	15.625	20.31
27	-5	1124	12.40	0.0217	16.85	232.0718	16.85	1392.4306	16.85	375	22.55	16.000	18.40
28	-4	1114	11.40	0.0213	14.81	228.0170	14.81	1368.1022	14.81	367	19.93	16.348	16.62
29	-3	1107	10.70	0.0211	13.77	225.9440	13.77	1355.6639	13.77	361	17.97	16.620	15.23
30	-2	1101	10.10	0.0210	13.23	224.8698	13.23	1349.2189	13.23	356	16.34	16.853	14.05
31	-1	1095	9.50	0.0210	13.17	224.7459	13.17	1348.4754	13.17	352	15.03	17.045	13.07
32	Std. pressure	1087	8.05	0.0210	12.34	224.5602	13.07	1347.3612	13.07	346	13.07	17.341	11.56
33	1	1076	7.60	0.0206	10.78	220.0170	10.78	1320.1020	10.78	339	10.78	17.699	9.73
34	2	1068	6.80	0.0203	9.15	216.7712	9.15	1300.6272	9.15	334	9.15	17.964	8.38
35	3	1060	6.00	0.0200	7.84	214.1759	7.84	1285.0554	7.84	330	7.84	18.181	7.27
36	4	1054	5.40	0.0198	6.86	212.2288	6.86	1273.3728	6.86	327	6.86	18.348	6.42
37	5	1044	4.40	0.0195	5.23	208.9837	5.23	1253.9022	5.23	322	5.23	18.633	4.97
38	6	1036	3.60	0.0194	4.25	207.0367	4.25	1242.2202	4.25	319	4.25	18.808	4.08
39	7	1026	2.60	0.0191	2.94	204.4406	2.94	1226.6436	2.94	315	2.94	19.047	2.86
40	8	1018	1.80	0.0189	1.96	202.4935	1.96	1214.9610	1.96	312	1.96	19.230	1.92
41	9	1006	0.60	0.0187	0.65	199.8975	0.65	1199.3850	0.65	308	0.65	19.480	0.65
42	10	1000	0.00	0.0186	0.00	198.5994	0.00	1191.5964	0.00	306	0.00	19.607	0.00

Table D.6 Experimental data for 21 test runs with the variation of tire pressure at 1.44° of front wheel total toe-in angle.
Weather temperature = (26.736 °C to 36.00°C), Humidity = (46.67% to 83.00%), Engine surface temperature= (79.87°C to 81.53°C)

Tire Pressure (psi)	Tire Pressure Variation	Engine RPM	Engine RPM Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance Coefficient (μ_{RC})	Rolling Resistance Coefficient Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance $F_R = \mu_{RC} \times W_v$, (N)	Rolling Resistance Rate (%) w.r.t. 0° Alignment Condition	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment Condition	Fuel Consumption (ml)	Fuel Consumption Rate (%) w.r.t. 0° Alignment Condition	(KPL) $_{ideal}$	KPL Reduction Rate (%)
22	-10	1373	25.62	0.0388	85.10	415.3410	85.10	2492.0459	85.10	491	40.29	12.219	28.72
23	-9	1329	21.59	0.0360	71.48	384.7846	71.48	2308.7076	71.48	477	36.29	12.578	26.62
24	-8	1300	18.94	0.0340	62.12	363.7709	62.12	2182.6257	62.12	467	33.43	12.847	25.06
25	-7	1269	16.10	0.0320	52.27	341.6759	52.27	2050.0552	52.27	456	30.29	13.157	23.25
26	-6	1239	13.36	0.0298	42.19	319.0483	42.19	1914.2899	42.19	444	26.86	13.513	21.17
27	-5	1216	11.25	0.0282	34.40	301.5662	34.40	1809.3974	34.40	434	24.00	13.824	19.36
28	-4	1202	9.97	0.0271	29.31	290.1626	29.31	1740.9755	29.31	427	22.00	14.051	18.03
29	-3	1190	8.87	0.0263	25.22	280.9800	25.22	1685.8803	25.22	421	20.29	14.251	16.87
30	-2	1180	7.96	0.0255	21.39	272.3777	21.39	1634.2660	21.39	415	18.57	14.457	15.66
31	-1	1171	7.14	0.0248	18.40	265.6747	18.40	1594.0481	18.40	410	17.14	14.634	14.63
32	Std. pressure	1161	5.45	0.0240	14.55	257.0371	14.55	1542.2227	14.55	403	15.14	14.888	13.15
33	1	1149	5.12	0.0232	10.65	248.2870	10.65	1489.7217	10.65	395	12.86	15.189	11.39
34	2	1141	4.39	0.0227	8.09	242.5483	8.09	1455.2900	8.09	389	11.14	15.424	10.02
35	3	1135	3.84	0.0223	6.21	238.3282	6.21	1429.9689	6.21	384	9.71	15.625	8.85
36	4	1130	3.39	0.0220	4.88	235.3312	4.88	1411.9870	4.88	380	8.57	15.789	7.89
37	5	1124	2.84	0.0217	3.43	232.0718	3.43	1392.4306	3.43	375	7.14	16.000	6.66
38	6	1119	2.38	0.0215	2.44	229.8631	2.44	1379.1785	2.44	371	6.00	16.172	5.66
39	7	1114	1.92	0.0213	1.62	228.0170	1.62	1368.1022	1.62	367	4.86	16.348	4.63
40	8	1108	1.37	0.0212	0.82	226.2306	0.82	1357.3838	0.82	362	3.43	16.574	3.31
41	9	1101	0.73	0.0210	0.22	224.8698	0.22	1349.2189	0.22	356	1.71	16.853	1.69
42	10	1093	0.00	0.0210	0.00	224.3852	0.00	1346.3112	0.00	350	0.00	17.142	0.00

Table D.7 Experimental data for 21 test runs with the variation of tire pressure at -1.44° of front wheel total toe-out angle.
Weather temperature = (22.36 °C to 34.35°C), Humidity = (48.23% to 1.50%), Engine surface temperature= (78.23°C to 81.80°C)

Tire Pressure (psi)	Tire Pressure Variation	Engine RPM	Engine RPM Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance Coefficient (μ_{RC})	Rolling Resistance Coefficient Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance $F_R = \mu_{RC} \times W_v$, (N)	Rolling Resistance Rate (%) w.r.t. 0° Alignment Condition	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment Condition	Fuel Consumption (ml)	Fuel Consumption Rate (%) w.r.t. 0° Alignment Condition	(KPL) _{ideal}	KPL Reduction Rate (%)
22	-10	1428	30.17	0.0424	102.02	453.5649	102.02	2721.3895	102.02	508	43.91	11.811	30.51
23	-9	1367	24.61	0.0384	83.02	410.9105	83.02	2465.4632	83.02	489	38.53	12.269	27.82
24	-8	1338	21.97	0.0366	74.26	391.2336	74.26	2347.4015	74.26	480	35.98	12.500	26.46
25	-7	1291	17.68	0.0334	59.29	357.6301	59.29	2145.7804	59.29	464	31.44	12.931	23.92
26	-6	1261	14.95	0.0314	49.59	335.8641	49.59	2015.1846	49.59	453	28.33	13.245	22.07
27	-5	1230	12.12	0.0292	38.92	311.8954	38.92	1871.3723	38.92	440	24.65	13.636	19.77
28	-4	1206	9.94	0.0274	30.66	293.3469	30.66	1760.0813	30.66	429	21.53	13.986	17.71
29	-3	1198	9.21	0.0268	27.85	287.0392	27.85	1722.2354	27.85	425	20.40	14.117	16.94
30	-2	1185	8.02	0.0259	23.20	276.6044	23.20	1659.6262	23.20	418	18.41	14.354	15.55
31	-1	1180	7.57	0.0255	21.32	272.3777	21.32	1634.2660	21.32	415	17.56	14.457	14.94
32	Std. pressure	1167	5.52	0.0245	16.63	261.8640	16.63	1571.1839	16.63	407	15.30	14.742	13.27
33	1	1156	5.38	0.0237	12.96	253.6123	12.96	1521.6737	12.96	400	13.31	15.000	11.75
34	2	1146	4.47	0.0230	9.70	246.2939	9.70	1477.7632	9.70	393	11.33	15.267	10.18
35	3	1141	4.01	0.0227	8.03	242.5483	8.03	1455.2900	8.03	389	10.20	15.424	9.25
36	4	1136	3.56	0.0224	6.51	239.1305	6.51	1434.7828	6.51	385	9.07	15.584	8.31
37	5	1131	3.10	0.0221	5.14	236.0483	5.14	1416.2900	5.14	381	7.93	15.748	7.35
38	6	1127	2.73	0.0219	4.21	233.9618	4.21	1403.7707	4.21	378	7.08	15.873	6.61
39	7	1121	2.19	0.0216	2.85	230.9226	2.85	1385.5354	2.85	373	5.67	16.085	5.37
40	8	1115	1.64	0.0214	1.75	228.4441	1.75	1370.6648	1.75	368	4.25	16.304	4.08
41	9	1106	0.82	0.0211	0.52	225.6811	0.52	1354.0867	0.52	360	1.98	16.666	1.95
42	10	1097	0.00	0.0210	0.00	224.5165	0.00	1347.0993	0.00	353	0.00	16.997	0.00

Table D.8 Experimental data for 20 test runs with the variation of vehicle load condition at 0.00° of front wheel total toe angle.
Weather temperature = (24.53 °C to 35.67°C), Humidity = (46.67% to 90.00%), Engine surface temperature= (79.93°C to 82.37°C

Vehicle Load (kg)	Engine RPM	Engine RPM Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance Coefficient (μ_{RC})	Rolling Resistance Coefficient Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance $F_R = \mu_{RC} \times V_w$, (N)	Rolling Resistance Rate (%) w.r.t. 0° Alignment Condition	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment Condition	Fuel Consumption (ml)	Fuel Consumption Rate (%) w.r.t. 0° Alignment Condition	(KPL) $_{ideal}$	KPL Reduction Rate (%)
1035	1050	0.00	0.0197	0.00	210.9306	0.00	1265.5836	0.00	325	0.00	18.461	0.00
1053	1062	1.14	0.0201	1.85	214.8247	1.85	1288.9482	1.85	331	1.85	18.126	1.81
1071	1075	2.38	0.0205	4.00	219.3678	4.00	1316.2068	4.00	338	4.00	17.751	3.84
1089	1084	3.24	0.0209	5.85	223.2619	5.85	1339.5714	5.85	344	5.85	17.441	5.53
1107	1094	4.19	0.0210	6.39	224.4042	6.39	1346.4249	6.39	351	8.00	17.094	7.40
1125	1105	5.24	0.0211	6.88	225.4422	6.88	1352.6529	6.88	359	10.46	16.713	9.47
1143	1114	6.10	0.0213	8.10	228.0170	8.10	1368.1022	8.10	367	12.92	16.348	11.45
1161	1125	7.14	0.0218	10.31	232.6797	10.31	1396.0784	10.31	376	15.69	15.957	13.56
1179	1135	8.10	0.0223	12.99	238.3282	12.99	1429.9689	12.99	384	18.15	15.625	15.36
1197	1148	9.33	0.0231	17.23	247.2805	17.23	1483.6829	17.23	394	21.23	15.228	17.51
1215	1161	10.57	0.0240	21.86	257.0371	21.86	1542.2227	21.86	403	24.00	14.888	19.35
1233	1175	11.90	0.0251	27.20	268.3038	27.20	1609.8228	27.20	412	26.77	14.563	21.11
1251	1192	13.52	0.0264	33.92	282.4710	33.92	1694.8262	33.92	422	29.85	14.218	22.98
1269	1214	15.62	0.0280	42.18	299.8934	42.18	1799.3605	42.18	433	33.23	13.856	24.94
1287	1241	18.19	0.0300	52.12	320.8685	52.12	1925.2108	52.12	445	36.92	13.483	26.96
1305	1275	21.43	0.0323	63.85	345.6039	63.85	2073.6233	63.85	458	40.92	13.100	29.04
1323	1308	24.57	0.0346	75.41	369.9908	75.41	2219.9450	75.41	470	44.62	12.765	30.85
1341	1338	27.43	0.0366	85.48	391.2336	85.48	2347.4015	85.48	480	47.69	12.500	32.29
1359	1376	31.05	0.0391	97.96	417.5630	97.96	2505.3779	97.96	492	51.38	12.195	33.94
1377	1422	35.43	0.0420	112.88	449.0318	112.88	2694.1909	112.88	506	55.69	11.857	35.77

Table D.9 Experimental data for 20 test runs with the variation of vehicle load condition at 1.44° of front wheel total toe-in angle. Weather temperature = (26.30 °C to 34.03°C), Humidity = (60.33% to 93.33%), Engine surface temperature = (80.08°C to 82.10°C)

Vehicle Load (kg)	Engine RPM	Engine RPM Rate (%)	Rolling Resistance Coefficient (μ_{RC})	Rolling Resistance Coefficient Rate (%)	Rolling Resistance $F_R = \mu_{RC} \times V_w$, (N)	Rolling Resistance Rate (%) w.r.t. 0° Alignment Condition	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment Condition	Fuel Consumption (ml)	Fuel Consumption Rate (%) w.r.t. 0° Alignment Condition	(KPL) $_{ideal}$	KPL Reduction Rate (%)
1035	1120	0.00	0.0215	0.00	230.3815	0.00	1382.2893	0.00	372	0.00	16.129	0.00
1053	1131	0.98	0.0221	2.46	236.0483	2.46	1416.2900	2.46	381	2.42	15.748	2.36
1071	1144	2.14	0.0229	6.08	244.3806	6.08	1466.2839	6.08	391	5.11	15.345	4.86
1089	1156	3.21	0.0237	10.08	253.6123	10.08	1521.6737	10.08	400	7.53	15.000	7.00
1107	1171	4.55	0.0248	15.32	265.6747	15.32	1594.0481	15.32	410	10.22	14.634	9.27
1125	1190	6.25	0.0263	21.96	280.9800	21.96	1685.8803	21.96	421	13.17	14.251	11.64
1143	1206	7.68	0.0274	27.33	293.3469	27.33	1760.0813	27.33	429	15.32	13.986	13.29
1161	1230	9.82	0.0292	35.38	311.8954	35.38	1871.3723	35.38	440	18.28	13.636	15.46
1179	1251	11.70	0.0307	42.49	328.2719	42.49	1969.6313	42.49	449	20.70	13.363	17.15
1197	1280	14.29	0.0327	51.74	349.5732	51.74	2097.4390	51.74	460	23.66	13.043	19.13
1215	1308	16.79	0.0346	60.60	369.9908	60.60	2219.9450	60.60	470	26.34	12.765	20.86
1233	1342	19.82	0.0368	70.76	393.3964	70.76	2360.3786	70.76	481	29.30	12.474	22.66
1251	1380	23.21	0.0393	82.21	419.7892	82.21	2518.7354	82.21	493	32.53	12.170	24.55
1269	1415	26.34	0.0416	92.94	444.5050	92.94	2667.0300	92.94	504	35.48	11.904	26.20
1287	1458	30.18	0.0443	105.75	473.9990	105.75	2843.9941	105.75	517	38.98	11.605	28.05
1305	1490	33.04	0.0471	118.50	503.3856	118.50	3020.3135	118.50	530	42.47	11.320	29.82
1323	1532	36.79	0.0496	130.06	530.0235	130.06	3180.1409	130.06	542	45.70	11.070	31.37
1341	1559	39.20	0.0518	140.34	553.7050	140.34	3322.2301	140.34	553	48.66	10.849	32.74
1359	1583	41.34	0.0543	151.92	580.3697	151.92	3482.2183	151.92	566	52.15	10.600	34.28
1377	1598	42.68	0.0573	165.77	612.2783	165.77	3673.6700	165.77	583	56.72	10.291	36.20

Table D.10 Experimental data for 20 test runs with the variation of vehicle load condition at -1.44° of front wheel total toe-out angle. Weather temperature = (25.23 °C to 36.00 °C), Humidity = (50.60% to 85.80%), Engine surface temperature= (79.30°C to 82.00°C)

Vehicle Load (kg)	Engine RPM	Engine RPM Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance Coefficient (μ_{RC})	Rolling Resistance Coefficient Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance $F_R = \mu_{RC} \times V_w$, (N)	Rolling Resistance Rate (%) w.r.t. 0° Alignment Condition	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment Condition	Fuel Consumption (ml)	Fuel Consumption Rate (%) w.r.t. 0° Alignment Condition	(KPL) _{ideal}	KPL Reduction Rate (%)
1035	1125	0.00	0.0218	0.00	232.6797	0.00	1396.0784	0.00	376	0.00	15.957	0.00
1053	1136	0.98	0.0224	2.77	239.1305	2.77	1434.7828	2.77	385	2.39	15.584	2.34
1071	1151	2.31	0.0233	7.15	249.3131	7.15	1495.8788	7.15	396	5.32	15.151	5.05
1089	1165	3.56	0.0244	12.01	260.6298	12.01	1563.7787	12.01	406	7.98	14.778	7.39
1107	1181	4.98	0.0256	17.66	273.7698	17.66	1642.6187	17.66	416	10.64	14.423	9.61
1125	1204	7.02	0.0273	25.39	291.7472	25.39	1750.4832	25.39	428	13.83	14.018	12.15
1143	1216	8.09	0.0282	29.61	301.5662	29.61	1809.3974	29.61	434	15.43	13.824	13.37
1161	1241	10.31	0.0300	37.90	320.8685	37.90	1925.2108	37.90	445	18.35	13.483	15.50
1179	1264	12.36	0.0316	45.17	337.7904	45.17	2026.7426	45.17	454	20.74	13.215	17.18
1197	1294	15.02	0.0336	54.58	359.6680	54.58	2158.0078	54.58	465	23.67	12.903	19.14
1215	1320	17.33	0.0354	62.63	378.3988	62.63	2270.3925	62.63	474	26.06	12.658	20.67
1233	1354	20.36	0.0376	72.82	402.1090	72.82	2412.6539	72.82	485	28.99	12.371	22.47
1251	1389	23.47	0.0399	83.30	426.4917	83.30	2558.9501	83.30	496	31.91	12.096	24.20
1269	1432	27.29	0.0426	95.91	455.8333	95.91	2734.9999	95.91	509	35.37	11.787	26.13
1287	1474	31.02	0.0454	108.59	485.3387	108.59	2912.0323	108.59	522	38.83	11.494	27.97
1305	1516	34.76	0.0483	122.10	516.7872	122.10	3100.7233	122.10	536	42.55	11.194	29.85
1323	1552	37.96	0.0512	135.23	547.3338	135.23	3284.0028	135.23	550	46.28	10.909	31.64
1341	1575	40.00	0.0533	145.11	570.3122	145.11	3421.8735	145.11	561	49.20	10.695	32.98
1359	1622	44.18	0.0564	159.28	603.3004	159.28	3619.8023	159.28	578	53.72	10.380	34.95
1377	1679	49.24	0.0596	173.68	636.7915	173.68	3820.7488	173.68	598	59.04	10.033	37.12

Table D.11 Experimental data for 20 test runs with the variation of brake application at 0.00° of front wheel total toe angle.
Weather temperature = (25.03 °C to 36.07°C), Humidity = (50.00% to 86.33%), Engine surface temperature= (80.10°C to 82.40°C)

Brake Application (times)	Rolling Resistance Coefficient (μ_{RC})	Rolling Resistance Coefficient Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance $F_R = \mu_{RC} \times V_w$, (N)	Rolling Resistance Rate (%) w.r.t. 0° Alignment Condition	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment Condition	Fuel Consumption (ml)	Fuel Consumption Rate (%) w.r.t. 0° Alignment Condition	(KPL) ideal	KPL Reduction Rate (%)
0	0.0210	0.00	224.6671	0.00	1348.0028	0.00	345	0.00	17.391	0.00
5	0.0210	0.03	224.7276	0.03	1348.3655	0.03	355	2.90	16.901	2.82
10	0.0213	1.31	227.6131	1.31	1365.6783	1.31	366	6.09	16.393	5.74
15	0.0218	3.57	232.6797	3.57	1396.0784	3.57	376	8.99	15.957	8.25
20	0.0226	7.56	241.6628	7.56	1449.9770	7.56	388	12.46	15.463	11.09
25	0.0235	11.91	251.4241	11.91	1508.5444	11.91	398	15.36	15.075	13.32
30	0.0247	17.68	264.3866	17.68	1586.3195	17.68	409	18.55	14.669	15.65
35	0.0261	24.41	279.5052	24.41	1677.0311	24.41	420	21.74	14.285	17.86
40	0.0279	32.75	298.2349	32.75	1789.4095	32.75	432	25.22	13.888	20.14
45	0.0295	40.41	315.4460	40.41	1892.6760	40.41	442	28.12	13.574	21.95
50	0.0316	50.35	337.7904	50.35	2026.7426	50.35	454	31.59	13.215	24.01
55	0.0340	61.92	363.7709	61.92	2182.6257	61.92	467	35.36	12.847	26.13
60	0.0366	74.14	391.2336	74.14	2347.4015	74.14	480	39.13	12.500	28.12
65	0.0397	88.84	424.2538	88.84	2545.5227	88.84	495	43.48	12.121	30.30
70	0.0431	104.91	460.3729	104.91	2762.2372	104.91	511	48.12	11.741	32.49
75	0.0469	123.06	501.1391	123.06	3006.8348	123.06	529	53.33	11.342	34.78
80	0.0512	143.62	547.3338	143.62	3284.0028	143.62	550	59.42	10.909	37.27
85	0.0555	164.38	593.9695	164.38	3563.8168	164.38	573	66.09	10.471	39.79
90	0.0597	184.10	638.2828	184.10	3829.6967	184.10	599	73.62	10.016	42.41
100	0.0628	198.79	671.2790	198.79	4027.6741	198.79	627	81.74	9.569	44.98

Table D.12 Experimental data for 20 test runs with the variation of brake application at 1.44° of front wheel total toe-in angle.
Weather temperature = (26.17 °C to 36.37°C), Humidity = (47.00% to 84.67%), Engine surface temperature= (80.50°C to 83.06°C)

Brake Application (times)	Rolling Resistance Coefficient (μ_{RC})	Rolling Resistance Coefficient Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance $F_R = \mu_{RC} \times V_w$, (N)	Rolling Resistance Rate (%) w.r.t. 0° Alignment Condition	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment Condition	Fuel Consumption (ml)	Fuel Consumption Rate (%) w.r.t. 0° Alignment Condition	(KPL) _{ideal}	KPL Reduction Rate (%)
0	0.0229	0.00	244.3806	0.00	1466.2839	0.00	391	0.00	15.345	0.00
5	0.0240	5.18	257.0371	5.18	1542.2227	5.18	403	3.07	14.888	2.98
10	0.0257	12.60	275.1787	12.60	1651.0724	12.60	417	6.65	14.388	6.24
15	0.0276	20.70	294.9615	20.70	1769.7689	20.70	430	9.97	13.953	9.07
20	0.0298	30.55	319.0483	30.55	1914.2899	30.55	444	13.55	13.513	11.94
25	0.0320	39.81	341.6759	39.81	2050.0552	39.81	456	16.62	13.157	14.26
30	0.0342	49.70	365.8357	49.70	2195.0142	49.70	468	19.69	12.820	16.45
35	0.0368	60.98	393.3964	60.98	2360.3786	60.98	481	23.02	12.474	18.71
40	0.0393	71.78	419.7892	71.78	2518.7354	71.78	493	26.09	12.170	20.69
45	0.0418	82.82	446.7675	82.82	2680.6051	82.82	505	29.16	11.881	22.57
50	0.0443	93.96	473.9990	93.96	2843.9941	93.96	517	32.23	11.605	24.37
55	0.0475	107.82	507.8683	107.82	3047.2099	107.82	532	36.06	11.278	26.50
60	0.0508	122.21	543.0477	122.21	3258.2862	122.21	548	40.15	10.948	28.65
65	0.0539	135.85	576.3790	135.85	3458.2739	135.85	564	44.25	10.638	30.67
70	0.0571	149.82	610.5124	149.82	3663.0745	149.82	582	48.85	10.309	32.82
75	0.0601	162.97	642.6401	162.97	3855.8405	162.97	602	53.96	9.966	35.05
80	0.0627	174.34	670.4246	174.34	4022.5478	174.34	626	60.10	9.584	37.54
85	0.0639	179.64	683.3747	179.64	4100.2481	179.64	653	67.01	9.188	40.12
90	0.0667	192.05	713.7236	192.05	4282.3416	192.05	682	74.42	8.797	42.67
100	0.0700	206.19	748.2586	206.19	4489.5516	206.19	715	82.86	8.391	45.32

Table D.13 Experimental data for 20 test runs with the variation of brake application at -1.44° of front wheel total toe-out angle. Weather temperature = (24.50 °C to 35.80°C), Humidity = (48.00% to 83.80%), Engine surface temperature= (77.80°C to 82.80°C)

Brake Application (times)	Rolling Resistance Coefficient (μ_{RC})	Rolling Resistance Coefficient Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance $F_R = \mu_{RC} \times V_w$, (N)	Rolling Resistance Rate (%) w.r.t. 0° Alignment Condition	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment Condition	Fuel Consumption (ml)	Fuel Consumption Rate (%) w.r.t. 0° Alignment Condition	(KPL) _{ideal}	KPL Reduction Rate (%)
0	0.0231	0.00	247.2805	0.00	1483.6829	0.00	394	0.00	15.228	0.00
5	0.0245	5.90	261.8640	5.90	1571.1839	5.90	407	3.30	14.742	3.19
10	0.0266	14.84	283.9780	14.84	1703.8679	14.84	423	7.36	14.184	6.86
15	0.0282	21.95	301.5662	21.95	1809.3974	21.95	434	10.15	13.824	9.22
20	0.0307	32.75	328.2719	32.75	1969.6313	32.75	449	13.96	13.363	12.25
25	0.0329	42.18	351.5729	42.18	2109.4371	42.18	461	17.01	13.015	14.53
30	0.0354	53.02	378.3988	53.02	2270.3925	53.02	474	20.30	12.658	16.88
35	0.0378	63.50	404.3015	63.50	2425.8089	63.50	486	23.35	12.345	18.93
40	0.0405	75.20	433.2257	75.20	2599.3540	75.20	499	26.65	12.024	21.04
45	0.0428	85.26	458.1027	85.26	2748.6162	85.26	510	29.44	11.764	22.75
50	0.0454	96.27	485.3387	96.27	2912.0323	96.27	522	32.49	11.494	24.52
55	0.0483	108.99	516.7872	108.99	3100.7233	108.99	536	36.04	11.194	26.49
60	0.0516	123.06	551.5893	123.06	3309.5358	123.06	552	40.10	10.869	28.62
65	0.0550	137.87	588.2139	137.87	3529.2833	137.87	570	44.67	10.526	30.88
70	0.0579	150.40	619.1863	150.40	3715.1180	150.40	587	48.98	10.221	32.88
75	0.0609	163.19	650.8069	163.19	3904.8416	163.19	608	54.31	9.868	35.20
80	0.0633	173.58	676.5010	173.58	4059.0063	173.58	634	60.91	9.463	37.86
85	0.0639	176.18	682.9389	176.18	4097.6333	176.18	660	67.51	9.090	40.31
90	0.0679	193.71	726.2818	193.71	4357.6908	193.71	694	76.14	8.645	43.23
100	0.0714	208.94	763.9564	208.94	4583.7384	208.94	730	85.28	8.219	46.03

Table D.14 Experimental data for 8 test runs for different front wheel total toe-in angle
Weather temperature = (16.7 °C to 36.40°C), Humidity = (39.00% to 97.00%), Engine surface temperature= (85.30°C to 87.90°C)

Front Wheel Total Toe-in Angle (deg.)	Angle Rate, $\dot{\alpha}_t = \frac{T_d}{V_s}$ (deg/sec)	Vehicle Load (kg)	Engine RPM	Engine RPM Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance Coefficient (μ_{RRC})	Rolling Resistance Coefficient Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance $F_R = (\mu_{RRC} \times V_w)$, N	Rolling Resistance Rate (%) w.r.t. 0° Alignment Condition	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment Condition	CO ₂ Emissions (PPM/%) for Octane	CO ₂ Emissions Rate (%)
0.00	0.00	955	1310	0.00	0.0210	0.00	196.7396	0.00	1180.4373	0.00	30000/ (3%)	0.00
0.00	0.00	1015	1350	3.05	0.0211	0.48	197.6764	0.48	1186.0584	0.48	30000/ (3%)	0.00
0.00	0.00	1075	1385	5.73	0.0212	0.95	198.6133	0.95	1191.6796	0.95	30000/ (3%)	0.00
0.40	0.06	955	1415	8.02	0.0213	1.42	199.5239	1.42	1197.1433	1.42	33000/ (3.3%)	10.00
1.00	0.14	955	1455	11.07	0.0226	7.82	212.1321	7.82	1272.7925	7.82	34000/ (3.4%)	13.33
1.60	0.22	955	1485	13.36	0.0251	19.38	234.8583	19.38	1409.1498	19.38	35000/ (3.5%)	16.67
4.20	0.58	955	1710	30.53	0.0481	128.86	450.2581	128.86	2701.5488	128.86	41000/ (4.1%)	36.67
4.80	0.66	955	1750	33.59	0.0562	167.84	526.9472	167.84	3161.6833	167.84	42000/ (4.2%)	40.00

Table D.15 Experimental data for 8 test runs for different front wheel total toe-in angle and tire wear
Weather temperature = (16.7 °C to 36.40°C), Humidity = (39.00% to 97.00%), Engine surface temperature= (85.30°C to 87.90°C)

Front Wheel Total Toe-in Angle (deg.)	Tire Outside Lateral Groove Wear (L_{g1}), mm	Tire Outside Lateral Groove Wear Rate (%)	Tire Circumferential Groove Wear (C_{g1}), mm	Tire Circumferential Groove Wear Rate (%)	Tire Circumferential Groove Wear (C_{g2}), mm	Tire Circumferential Groove Wear Rate (%)	Tire Circumferential Groove Wear (C_{g3}), mm	Tire Circumferential Groove Wear Rate (%)	Tire Circumferential Groove Wear (C_{g4}), mm	Tire Circumferential Groove Wear Rate (%)	Tire Inside Lateral Groove Wear (L_{g2}), mm	Tire Inside Lateral Groove Wear Rate (%)	Tire Travelling Life, $\frac{T_d \times T_{cgd}}{T_w} (T_{tl})$	Tire Travelling Life Reduction Rate (%)
0.00	0.03	0.00	0.04	0.00	0.04	0.00	0.04	0.00	0.04	0.00	0.03	0.00	92250.00	0.00
0.00	0.04	33.33	0.05	25.00	0.04	0.00	0.05	25.00	0.05	25.00	0.04	33.33	73800.00	20.00
0.00	0.04	33.33	0.05	25.00	0.05	25.00	0.05	25.00	0.05	25.00	0.05	66.67	73800.00	20.00
0.40	0.05	66.67	0.06	50.00	0.07	75.00	0.06	50.00	0.06	50.00	0.05	66.67	61500.00	33.33
1.00	0.09	200.00	0.08	100.00	0.08	100.00	0.09	125.00	0.08	100.00	0.07	133.33	46125.00	50.00
1.60	0.37	1133.33	0.35	775.00	0.34	750.00	0.35	775.00	0.20	400.00	0.09	200.00	10542.86	88.57
4.20	2.43	8000.00	2.40	5900.00	2.41	5925.00	2.40	5900.00	0.90	2150.00	0.13	333.33	1537.50	98.33
4.80	3.01	9933.33	4.35	10775.00	4.35	10775.00	4.34	10750.00	1.52	3700.00	0.29	866.67	848.28	99.08

Table D.16 Experimental data for 8 test runs for different front total toe-out angle.
Weather temperature = (25.10 °C to 37.60°C), Humidity = (50.00% to 86.00%), Engine surface temperature= (79.97°C to 85.56°C)

Front Wheel Total Toe-out Angle (deg.)	Angle Rate, $\dot{\alpha}_t \times \dot{V}_s$ (deg/sec)	Vehicle Load (kg)	Engine RPM	Engine RPM Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance Coefficient (μ_{RC})	Rolling Resistance Coefficient Rate (%) w.r.t. 0° Alignment Condition	Rolling Resistance $F_R = (\mu_{RC} \times V_W)$, N	Rolling Resistance Rate (%) w.r.t. 0° Alignment Condition	Energy Consumption $E_c = (F_R \times T_d)$, KJ	Energy Consumption Rate (%) w.r.t. 0° Alignment Condition
0.00	0.00	955	1310	0.00	0.0210	0.00	196.7396	0.00	1180.4376	0.00
0.00	0.00	1015	1350	3.05	0.0211	0.48	197.6764	0.48	1186.0584	0.48
0.00	0.00	1075	1385	5.73	0.0212	0.95	198.6133	0.95	1191.6796	0.95
-0.40	-0.06	955	1460	11.45	0.0330	57.22	309.3104	57.22	1855.8622	57.22
-1.00	-0.14	955	1475	12.60	0.0342	62.94	320.5675	62.94	1923.4047	62.94
-1.60	-0.22	955	1520	16.03	0.0378	80.11	354.3387	80.11	2126.0321	80.11
-4.20	-0.58	955	1750	33.59	0.0562	167.84	526.9472	167.84	3161.6832	167.84
-4.80	-0.66	955	1800	37.40	0.0603	186.91	564.4708	186.91	3386.8247	186.91

Table D.17 Experimental Data for 8 test runs for different front wheel total toe-out angle and tire wear
Weather temperature = (25.10°C to 37.60°C), Humidity = (50.00% to 86.00%), Engine surface temperature= (79.97°C to 85.56°C)

Front Wheel Total Toe-out Angle (deg.)	Tire Outside Lateral Groove Wear (L_{g1}), mm	Tire Outside Lateral Groove Wear Rate (%)	Tire Circumferential Groove Wear (C_{g1}), mm	Tire Circumferential Groove Wear Rate (%)	Tire Circumferential Groove Wear (C_{g2}), mm	Tire Circumferential Groove Wear Rate (%)	Tire Circumferential Groove Wear (C_{g3}), mm	Tire Circumferential Groove Wear Rate (%)	Tire Circumferential Groove Wear (C_{g4}), mm	Tire Circumferential Groove Wear Rate (%)	Tire Inside Lateral Groove Wear (L_{g2}), mm	Tire Inside Lateral Groove Wear Rate (%)	Tire Travelling Life, $\frac{T_a \times T_{cgd}}{T_w} (T_{tl})$	Tire Travelling Life Reduction Rate (%)
0.00	0.03	0.00	0.03	0.00	0.03	0.00	0.04	0.00	0.04	0.00	0.03	0.00	92250.00	0.00
0.00	0.03	0.00	0.04	33.33	0.05	66.67	0.04	0.00	0.04	0.00	0.03	0.00	92250.00	0.00
0.00	0.05	66.67	0.05	66.67	0.05	66.67	0.06	50.00	0.05	25.00	0.04	33.33	73800.00	20.00
-0.40	0.06	100.00	0.07	133.33	0.07	133.33	0.08	100.00	0.07	75.00	0.06	100.00	52714.29	42.86
-1.00	0.08	166.67	0.10	233.33	0.10	233.33	0.10	150.00	0.09	125.00	0.11	266.67	41000.00	55.56
-1.60	0.10	233.33	0.22	633.33	0.37	1133.33	0.35	775.00	0.37	825.00	0.39	1200.00	9972.97	89.19
-4.20	0.14	366.67	0.93	3000.00	2.41	7933.33	2.42	5950.00	2.42	5950.00	2.44	8033.33	1524.79	98.35
-4.80	0.32	966.67	1.49	4866.67	4.30	14233.33	4.29	10625.00	4.30	10650.00	2.96	9766.67	823.81	99.11

APPENDIX-E: PEARSON'S CORRELATION COEFFICIENT FORMULA

Table E.1 Relation between the front wheel left toe-in angle (deg.) and fuel consumption (ml) using by Pearson's correlation coefficient formula

Front Wheel Left Toe-in angle α_i (°)	Fuel Consumption f_c (ml)	$(\alpha_i - \bar{\alpha})$	$(f_c - \bar{f}_c)$	$(\alpha_i - \bar{\alpha}) \times (f_c - \bar{f}_c)$	$(\alpha_i - \bar{\alpha})^2$	$(f_c - \bar{f}_c)^2$	$r_{\alpha f_c}$
0.00	348	-1.30	-62.75	81.83	1.700416	3937.5625	0.99
0.08	352	-1.22	-58.75	71.91	1.498176	3451.5625	
0.20	356	-1.10	-54.75	60.44	1.218816	2997.5625	
0.35	363	-0.95	-47.75	45.55	0.910116	2280.0625	
0.50	369	-0.80	-41.75	33.57	0.646416	1743.0625	
0.65	375	-0.65	-35.75	23.38	0.427716	1278.0625	
0.80	384	-0.50	-26.75	13.48	0.254016	715.5625	
0.95	392	-0.35	-18.75	6.64	0.125316	351.5625	
1.10	398	-0.20	-12.75	2.60	0.041616	162.5625	
1.25	405	-0.05	-5.75	0.31	0.002916	33.0625	
1.40	414	0.10	3.25	0.31	0.009216	10.5625	
1.55	421	0.25	10.25	2.52	0.060516	105.0625	
1.70	428	0.40	17.25	6.83	0.156816	297.5625	
1.84	436	0.54	25.25	13.53	0.287296	637.5625	
2.00	443	0.70	32.25	22.45	0.484416	1040.0625	
2.14	451	0.84	40.25	33.65	0.698896	1620.0625	
2.25	459	0.95	48.25	45.64	0.894916	2328.0625	
2.36	467	1.06	56.25	59.40	1.115136	3164.0625	
2.43	473	1.13	62.25	70.09	1.267876	3875.0625	
2.53	481	1.23	70.25	86.13	1.503076	4935.0625	
$\bar{\alpha} = 1.30$	$\bar{f}_c = 410.75$			680.27	13.30	34963.75	

Table E.2 Relation between the front wheel left toe-out angle (deg.) and fuel consumption (ml) using by Pearson's correlation coefficient formula

Front Wheel Left Toe-out Angle α_i (°)	Fuel Consumption (ml) f_c	$(\alpha_i - \bar{\alpha})$	$(f_c - \bar{f}_c)$	$(\alpha_i - \bar{\alpha}) \times (f_c - \bar{f}_c)$	$(\alpha_i - \bar{\alpha})^2$	$(f_c - \bar{f}_c)^2$	$r_{\alpha f_c}$
0.00	348	1.30	-69.20	-90.24	1.700416	4788.64	0.99
-0.08	353	1.22	-64.20	-78.58	1.498176	4121.64	
-0.20	360	1.10	-57.20	-63.15	1.218816	3271.84	
-0.35	368	0.95	-49.20	-46.94	0.910116	2420.64	
-0.50	376	0.80	-41.20	-33.12	0.646416	1697.44	
-0.65	381	0.65	-36.20	-23.67	0.427716	1310.44	
-0.80	390	0.50	-27.20	-13.71	0.254016	739.84	
-0.95	397	0.35	-20.20	-7.15	0.125316	408.04	
-1.10	404	0.20	-13.20	-2.69	0.041616	174.24	
-1.25	412	0.05	-5.20	-0.28	0.002916	27.04	
-1.40	418	-0.10	0.80	-0.08	0.009216	0.64	
-1.55	427	-0.25	9.80	-2.41	0.060516	96.04	
-1.70	435	-0.40	17.80	-7.05	0.156816	316.84	
-1.84	443	-0.54	25.80	-13.83	0.287296	665.64	
-2.00	452	-0.70	34.80	-24.22	0.484416	1211.04	
-2.14	459	-0.84	41.80	-34.94	0.698896	1747.24	
-2.25	467	-0.95	49.80	-47.11	0.894916	2480.04	
-2.36	475	-1.06	57.80	-61.04	1.115136	3340.84	
-2.43	484	-1.13	66.80	-75.22	1.267876	4462.24	
-2.53	495	-1.23	77.80	-95.38	1.503076	6052.84	
$\bar{\alpha} = -1.30$	$\bar{f}_c = 417.20$			-720.81	13.30	39333.20	

Table E.3 Relation between the front wheel total toe-in angle (deg.) and fuel consumption (ml) using by Pearson's correlation coefficient formula

Front Wheel Total Toe-in Angle α_i (°)	Fuel Consumption (ml) f_c	$(\alpha_i - \bar{\alpha})$	$(f_c - \bar{f}_c)$	$(\alpha_i - \bar{\alpha}) \times (f_c - \bar{f}_c)$	$(\alpha_i - \bar{\alpha})^2$	$(f_c - \bar{f}_c)^2$	$r_{\alpha f_c}$
0.00	346	-2.61	-116.80	304.61	6.801664	13642.24	0.99
0.16	353	-2.45	-109.80	268.79	5.992704	12056.04	
0.40	366	-2.21	-96.80	213.73	4.875264	9370.24	
0.70	377	-1.91	-85.80	163.71	3.640464	7361.64	
1.00	388	-1.61	-74.80	120.28	2.585664	5595.04	
1.30	398	-1.31	-64.80	84.76	1.710864	4199.04	
1.60	410	-1.01	-52.80	53.22	1.016064	2787.84	
1.90	422	-0.71	-40.80	28.89	0.501264	1664.64	
2.20	435	-0.41	-27.80	11.34	0.166464	772.84	
2.50	449	-0.11	-13.80	1.49	0.011664	190.44	
2.80	462	0.19	-0.80	-0.15	0.036864	0.64	
3.10	477	0.49	14.20	6.99	0.242064	201.64	
3.40	490	0.79	27.20	21.54	0.627264	739.84	
3.68	507	1.07	44.20	47.38	1.149184	1953.64	
4.00	523	1.39	60.20	83.80	1.937664	3624.04	
4.28	539	1.67	76.20	127.41	2.795584	5806.44	
4.50	554	1.89	91.20	172.55	3.579664	8317.44	
4.72	571	2.11	108.20	228.52	4.460544	11707.24	
4.86	587	2.25	124.20	279.70	5.071504	15425.64	
5.06	602	2.45	139.20	341.32	6.012304	19376.64	
$\bar{\alpha} = 2.61$	$\bar{f}_c = 462.80$			2559.87	53.21	124793.20	

Table E.4 Relation between the front total wheel toe-out angle (deg.) and fuel consumption (ml) using by Pearson's correlation coefficient formula

Front Wheel Total Toe-out Angle α_i (°)	Fuel Consumption (ml) f_c	$(\alpha_i - \bar{\alpha})$	$(f_c - \bar{f}_c)$	$(\alpha_i - \bar{\alpha}) \times (f_c - \bar{f}_c)$	$(\alpha_i - \bar{\alpha})^2$	$(f_c - \bar{f}_c)^2$	$r_{\alpha f_c}$
0.00	348	2.61	-121.25	-316.22	6.801664	14701.5625	0.99
-0.16	358	2.45	-111.25	-272.34	5.992704	12376.5625	
-0.40	372	2.21	-97.25	-214.73	4.875264	9457.5625	
-0.70	384	1.91	-85.25	-162.66	3.640464	7267.5625	
-1.00	394	1.61	-75.25	-121.00	2.585664	5662.5625	
-1.30	403	1.31	-66.25	-86.66	1.710864	4389.0625	
-1.60	415	1.01	-54.25	-54.68	1.016064	2943.0625	
-1.90	426	0.71	-43.25	-30.62	0.501264	1870.5625	
-2.20	439	0.41	-30.25	-12.34	0.166464	915.0625	
-2.50	453	0.11	-16.25	-1.76	0.011664	264.0625	
-2.80	465	-0.19	-4.25	0.82	0.036864	18.0625	
-3.10	480	-0.49	10.75	-5.29	0.242064	115.5625	
-3.40	493	-0.79	23.75	-18.81	0.627264	564.0625	
-3.68	511	-1.07	41.75	-44.76	1.149184	1743.0625	
-4.00	528	-1.39	58.75	-81.78	1.937664	3451.5625	
-4.28	546	-1.67	76.75	-128.33	2.795584	5890.5625	
-4.50	564	-1.89	94.75	-179.27	3.579664	8977.5625	
-4.72	581	-2.11	111.75	-236.02	4.460544	12488.0625	
-4.86	601	-2.25	131.75	-296.70	5.071504	17358.0625	
-5.06	624	-2.45	154.75	-379.45	6.012304	23947.5625	
$\bar{\alpha} = -2.61$	$\bar{f}_c = 469.25$			-2642.58	53.21	134401.75	

Table E.5 Relation between the tire pressure at 0.00° of front wheel total toe angle and fuel consumption (ml) using by Pearson's correlation coefficient formula

Tire Pressure t_p (psi)	Fuel Consumption f_c (ml)	$(t_p - \bar{t}_p)$	$(f_c - \bar{f}_c)$	$(t_p - \bar{t}_p) \times (f_c - \bar{f}_c)$	$(t_p - \bar{t}_p)^2$	$(f_c - \bar{f}_c)^2$	$r_{t_p f_c}$
22	426	-11.60	56.35	-653.66	134.56	3175.3225	-0.76
23	414	-10.60	44.35	-470.11	112.36	1966.9225	
24	405	-9.60	35.35	-339.36	92.16	1249.6225	
25	395	-8.60	25.35	-218.01	73.96	642.6225	
26	384	-7.60	14.35	-109.06	57.76	205.9225	
27	375	4.00	5.35	21.40	16.00	28.6225	
28	367	-5.60	-2.65	14.84	31.36	7.0225	
29	361	-4.60	55.00	-253.00	21.16	3025.0000	
30	356	-3.60	-13.65	49.14	12.96	186.3225	
31	352	-2.60	-17.65	45.89	6.76	311.5225	
32	346	-1.60	-23.65	37.84	2.56	559.3225	
33	339	-0.60	-30.65	18.39	0.36	939.4225	
34	334	0.40	-35.65	-14.26	0.16	1270.9225	
35	330	1.40	-39.65	-55.51	1.96	1572.1225	
36	327	2.40	-42.65	-102.36	5.76	1819.0225	
37	322	3.40	-47.65	-162.01	11.56	2270.5225	
38	319	4.40	-50.65	-222.86	19.36	2565.4225	
39	315	5.40	-54.65	-295.11	29.16	2986.6225	
40	312	6.40	-57.65	-368.96	40.96	3323.5225	
41	308	7.40	-61.65	-456.21	54.76	3800.7225	
42	306	8.40	-63.65	-534.66	70.56	4051.3225	
$\bar{t}_p = 33.60$	$\bar{f}_c = 369.65$			-4067.64	796.20	35957.85	

Table E.6 Relation between the tire pressure at 1.44° of front wheel total toe-in angle and fuel consumption (ml) using by Pearson's correlation coefficient formula

Tire Pressure t_p (psi)	Fuel Consumption f_c (ml)	$(t_p - \bar{t}_p)$	$(f_c - \bar{f}_c)$	$(t_p - \bar{t}_p) \times (f_c - \bar{f}_c)$	$(t_p - \bar{t}_p)^2$	$(f_c - \bar{f}_c)^2$	$r_{t_p f_c}$
22	491	-11.60	62.30	-722.68	134.56	3881.290	-0.76
23	477	-10.60	48.30	-511.98	112.36	2332.890	
24	467	-9.60	38.30	-367.68	92.16	1466.890	
25	456	-8.60	27.30	-234.78	73.96	745.290	
26	444	-7.60	15.30	-116.28	57.76	234.090	
27	434	4.00	5.30	21.20	16.00	28.090	
28	427	-5.60	-1.70	9.52	31.36	2.890	
29	421	-4.60	71.00	-326.60	21.16	5041.000	
30	415	-3.60	-13.70	49.32	12.96	187.690	
31	410	-2.60	-18.70	48.62	6.76	349.690	
32	403	-1.60	-25.70	41.12	2.56	660.490	
33	395	-0.60	-33.70	20.22	0.36	1135.690	
34	389	0.40	-39.70	-15.88	0.16	1576.090	
35	384	1.40	-44.70	-62.58	1.96	1998.090	
36	380	2.40	-48.70	-116.88	5.76	2371.690	
37	375	3.40	-53.70	-182.58	11.56	2883.690	
38	371	4.40	-57.70	-253.88	19.36	3329.290	
39	367	5.40	-61.70	-333.18	29.16	3806.890	
40	362	6.40	-66.70	-426.88	40.96	4448.890	
41	356	7.40	-72.70	-537.98	54.76	5285.290	
42	350	8.40	-78.70	-661.08	70.56	6193.690	
$\bar{t}_p =$ 33.60	$\bar{f}_c = 428.70$			-4680.92	796.20	47959.60	

Table E.7 Relation between the tire pressure at -1.44° of front wheel total toe-out angle and fuel consumption (ml) using by Pearson's correlation coefficient formula

Tire Pressure t_p (psi)	Fuel Consumption f_c (ml)	$(t_p - \bar{t}_p)$	$(f_c - \bar{f}_c)$	$(t_p - \bar{t}_p) \times (f_c - \bar{f}_c)$	$(t_p - \bar{t}_p)^2$	$(f_c - \bar{f}_c)^2$	$r_{t_p f_c}$
22	508	-11.60	72.60	-842.16	134.56	5270.76	-0.76
23	489	-10.60	53.60	-568.16	112.36	2872.96	
24	480	-9.60	44.60	-428.16	92.16	1989.16	
25	464	-8.60	28.60	-245.96	73.96	817.96	
26	453	-7.60	17.60	-133.76	57.76	309.76	
27	440	4.00	4.60	18.40	16	21.16	
28	429	-5.60	-6.40	35.84	31.36	40.96	
29	425	-4.60	72.00	-331.20	21.16	5184	
30	418	-3.60	-17.40	62.64	12.96	302.76	
31	415	-2.60	-20.40	53.04	6.76	416.16	
32	407	-1.60	-28.40	45.44	2.56	806.56	
33	400	-0.60	-35.40	21.24	0.36	1253.16	
34	393	0.40	-42.40	-16.96	0.16	1797.76	
35	389	1.40	-46.40	-64.96	1.96	2152.96	
36	385	2.40	-50.40	-120.96	5.76	2540.16	
37	381	3.40	-54.40	-184.96	11.56	2959.36	
38	378	4.40	-57.40	-252.56	19.36	3294.76	
39	373	5.40	-62.40	-336.96	29.16	3893.76	
40	368	6.40	-67.40	-431.36	40.96	4542.76	
41	360	7.40	-75.40	-557.96	54.76	5685.16	
42	353	8.40	-82.40	-692.16	70.56	6789.76	
$\bar{t}_p = 33.60$	$\bar{f}_c = 435.40$			-4971.64	796.20	52941.80	

Table E.8 Relation between the vehicle load at 0.00° of front wheel total toe angle and fuel consumption (ml) using by Pearson's correlation coefficient formula.

Vehicle Load Condition (v_l) kg	Fuel Consumption f_c (ml)	$(v_l - \bar{v}_l)$	$(f_c - \bar{f}_c)$	$(v_l - \bar{v}_l) \times (f_c - \bar{f}_c)$	$(v_l - \bar{v}_l)^2$	$(f_c - \bar{f}_c)^2$	$r_{v_l f_c}$
1035	325	-171.00	-79.50	13594.50	29241	6320.25	0.99
1053	331	-153.00	-73.50	11245.50	23409	5402.25	
1071	338	-135.00	-66.50	8977.50	18225	4422.25	
1089	344	-117.00	-60.50	7078.50	13689	3660.25	
1107	351	-99.00	-53.50	5296.50	9801	2862.25	
1125	359	-81.00	-45.50	3685.50	6561	2070.25	
1143	367	-63.00	-37.50	2362.50	3969	1406.25	
1161	376	-45.00	-28.50	1282.50	2025	812.25	
1179	384	-27.00	-20.50	553.50	729	420.25	
1197	394	-9.00	-10.50	94.50	81	110.25	
1215	403	9.00	-1.50	-13.50	81	2.25	
1233	412	27.00	7.50	202.50	729	56.25	
1251	422	45.00	17.50	787.50	2025	306.25	
1269	433	63.00	28.50	1795.50	3969	812.25	
1287	445	81.00	40.50	3280.50	6561	1640.25	
1305	458	99.00	53.50	5296.50	9801	2862.25	
1323	470	117.00	65.50	7663.50	13689	4290.25	
1341	480	135.00	75.50	10192.50	18225	5700.25	
1359	492	153.00	87.50	13387.50	23409	7656.25	
1377	506	171.00	101.50	17356.50	29241	10302.25	
$\bar{v}_l = 1206.00$	$\bar{f}_c = 404.50$			114120.00	215460.00	61115.00	

Table E.9 Relation between the vehicle load at 1.44° of front wheel total toe-in angle and fuel consumption (ml) Using by Pearson's correlation coefficient formula.

Vehicle Load Condition (v_l) kg	Fuel Consumption f_c (ml)	$(v_l - \bar{v}_l)$	$(f_c - \bar{f}_c)$	$(v_l - \bar{v}_l) \times (f_c - \bar{f}_c)$	$(v_l - \bar{v}_l)^2$	$(f_c - \bar{f}_c)^2$	$r_{v_l f_c}$
1035	372	-171.00	-97.60	16689.60	29241	9525.76	0.99
1053	381	-153.00	-88.60	13555.80	23409	7849.96	
1071	391	-135.00	-78.60	10611.00	18225	6177.96	
1089	400	-117.00	-69.60	8143.20	13689	4844.16	
1107	410	-99.00	-59.60	5900.40	9801	3552.16	
1125	421	-81.00	-48.60	3936.60	6561	2361.96	
1143	429	-63.00	-40.60	2557.80	3969	1648.36	
1161	440	-45.00	-29.60	1332.00	2025	876.16	
1179	449	-27.00	-20.60	556.20	729	424.36	
1197	460	-9.00	-9.60	86.40	81	92.16	
1215	470	9.00	0.40	3.60	81	0.16	
1233	481	27.00	11.40	307.80	729	129.96	
1251	493	45.00	23.40	1053.00	2025	547.56	
1269	504	63.00	34.40	2167.20	3969	1183.36	
1287	517	81.00	47.40	3839.40	6561	2246.76	
1305	530	99.00	60.40	5979.60	9801	3648.16	
1323	542	117.00	72.40	8470.80	13689	5241.76	
1341	553	135.00	83.40	11259.00	18225	6955.56	
1359	566	153.00	96.40	14749.20	23409	9292.96	
1377	583	171.00	113.40	19391.40	29241	12859.56	
$\bar{v}_l =$ 1206.00	$\bar{f}_c =$ 469.60			130590.00	215460.00	79458.80	

Table E.10 Relation between the vehicle load at -1.44° of front wheel total toe-out angle and fuel consumption (ml) using by Pearson's correlation coefficient formula

Vehicle Load Condition (v_l) kg	Fuel Consumption (ml) f_c	$(v_l - \bar{v}_l)$	$(f_c - \bar{f}_c)$	$(v_l - \bar{v}_l) \times (f_c - \bar{f}_c)$	$(v_l - \bar{v}_l)^2$	$(f_c - \bar{f}_c)^2$	r_{vlf_c}
1035	376	-171.00	-99.70	17048.70	29241	9940.09	0.99
1053	385	-153.00	-90.70	13877.10	23409	8226.49	
1071	396	-135.00	-79.70	10759.50	18225	6352.09	
1089	406	-117.00	-69.70	8154.90	13689	4858.09	
1107	416	-99.00	-59.70	5910.30	9801	3564.09	
1125	428	-81.00	-47.70	3863.70	6561	2275.29	
1143	434	-63.00	-41.70	2627.10	3969	1738.89	
1161	445	-45.00	-30.70	1381.50	2025	942.49	
1179	454	-27.00	-21.70	585.90	729	470.89	
1197	465	-9.00	-10.70	96.30	81	114.49	
1215	474	9.00	-1.70	-15.30	81	2.89	
1233	485	27.00	9.30	251.10	729	86.49	
1251	496	45.00	20.30	913.50	2025	412.09	
1269	509	63.00	33.30	2097.90	3969	1108.89	
1287	522	81.00	46.30	3750.30	6561	2143.69	
1305	536	99.00	60.30	5969.70	9801	3636.09	
1323	550	117.00	74.30	8693.10	13689	5520.49	
1341	561	135.00	85.30	11515.50	18225	7276.09	
1359	578	153.00	102.30	15651.90	23409	10465.29	
1377	598	171.00	122.30	20913.30	29241	14957.29	
$\bar{v}_l = 1206.00$	$\bar{f}_c = 475.70$			134046.00	215460.00	84092.20	

Table E.11 Relation between the brake application at 0.00° of front wheel total toe angle and fuel consumption (ml) using by Pearson's correlation coefficient formula

Brake Application (b_a) times	Fuel Consumption f_c (ml)	$(b_a - \bar{b}_a)$	$(f_c - \bar{f}_c)$	$(b_a - \bar{b}_a) \times (f_c - \bar{f}_c)$	$(b_a - \bar{b}_a)^2$	$(f_c - \bar{f}_c)^2$	r_{bafc}
0	345	-47.75	-115.80	5529.45	2280.0625	13409.64	0.98
5	355	-42.75	-105.80	4522.95	1827.5625	11193.64	
10	366	-37.75	-94.80	3578.70	1425.0625	8987.04	
15	376	-32.75	-84.80	2777.20	1072.5625	7191.04	
20	388	-27.75	-72.80	2020.20	770.0625	5299.84	
25	398	-22.75	-62.80	1428.70	517.5625	3943.84	
30	409	-17.75	-51.80	919.45	315.0625	2683.24	
35	420	-12.75	-40.80	520.20	162.5625	1664.64	
40	432	-7.75	-28.80	223.20	60.0625	829.44	
45	442	-2.75	-18.80	51.70	7.5625	353.44	
50	454	2.25	-6.80	-15.30	5.0625	46.24	
55	467	7.25	6.20	44.95	52.5625	38.44	
60	480	12.25	19.20	235.20	150.0625	368.64	
65	495	17.25	34.20	589.95	297.5625	1169.64	
70	511	22.25	50.20	1116.95	495.0625	2520.04	
75	529	27.25	68.20	1858.45	742.5625	4651.24	
80	550	32.25	89.20	2876.70	1040.0625	7956.64	
85	573	37.25	112.20	4179.45	1387.5625	12588.84	
90	599	42.25	138.20	5838.95	1785.0625	19099.24	
100	627	52.25	166.20	8683.95	2730.0625	27622.44	
$\bar{b}_a = 47.75$	$\bar{f}_c = 460.80$			46981.00	17123.75	131617.20	

Table E.12 Relation between the brake application at 1.44° of front wheel total toe-in angle and fuel consumption (ml) using by Pearson's correlation coefficient formula

Brake Application (b_a) times	Fuel Consumption f_c (ml)	$(b_a - \bar{b}_a)$	$(f_c - \bar{f}_c)$	$(b_a - \bar{b}_a) \times (f_c - \bar{f}_c)$	$(b_a - \bar{b}_a)^2$	$(f_c - \bar{f}_c)^2$	r_{bafc}
0	391	-47.75	-134.45	6419.99	2280.0625	18076.8025	0.99
5	403	-42.75	-122.45	5234.74	1827.5625	14994.0025	
10	417	-37.75	-108.45	4093.99	1425.0625	11761.4025	
15	430	-32.75	-95.45	3125.99	1072.5625	9110.7025	
20	444	-27.75	-81.45	2260.24	770.0625	6634.1025	
25	456	-22.75	-69.45	1579.99	517.5625	4823.3025	
30	468	-17.75	-57.45	1019.74	315.0625	3300.5025	
35	481	-12.75	-44.45	566.74	162.5625	1975.8025	
40	493	-7.75	-32.45	251.49	60.0625	1053.0025	
45	505	-2.75	-20.45	56.24	7.5625	418.2025	
50	517	2.25	-8.45	-19.01	5.0625	71.4025	
55	532	7.25	6.55	47.49	52.5625	42.9025	
60	548	12.25	22.55	276.24	150.0625	508.5025	
65	564	17.25	38.55	664.99	297.5625	1486.1025	
70	582	22.25	56.55	1258.24	495.0625	3197.9025	
75	602	27.25	76.55	2085.99	742.5625	5859.9025	
80	626	32.25	100.55	3242.74	1040.0625	10110.3025	
85	653	37.25	127.55	4751.24	1387.5625	16269.0025	
90	682	42.25	156.55	6614.24	1785.0625	24507.9025	
100	715	52.25	189.55	9903.99	2730.0625	35929.2025	
$\bar{b}_a = 47.75$	$\bar{f}_c = 525.45$			53435.25	17123.75	170130.95	

Table E.13 Relation between the brake application at -1.44° of front wheel total toe-out angle and fuel consumption (ml) using by Pearson's correlation coefficient formula

Brake Application (b_a) times	Fuel Consumption f_c (ml)	$(b_a - \bar{b}_a)$	$(f_c - \bar{f}_c)$	$(b_a - \bar{b}_a) \times (f_c - \bar{f}_c)$	$(b_a - \bar{b}_a)^2$	$(f_c - \bar{f}_c)^2$	r_{bafc}
0	394	-47.75	-137.50	6565.63	2280.0625	18906.25	0.98
5	407	-42.75	-124.50	5322.38	1827.5625	15500.25	
10	423	-37.75	-108.50	4095.88	1425.0625	11772.25	
15	434	-32.75	-97.50	3193.13	1072.5625	9506.25	
20	449	-27.75	-82.50	2289.38	770.0625	6806.25	
25	461	-22.75	-70.50	1603.88	517.5625	4970.25	
30	474	-17.75	-57.50	1020.63	315.0625	3306.25	
35	486	-12.75	-45.50	580.13	162.5625	2070.25	
40	499	-7.75	-32.50	251.88	60.0625	1056.25	
45	510	-2.75	-21.50	59.13	7.5625	462.25	
50	522	2.25	-9.50	-21.38	5.0625	90.25	
55	536	7.25	4.50	32.63	52.5625	20.25	
60	552	12.25	20.50	251.13	150.0625	420.25	
65	570	17.25	38.50	664.13	297.5625	1482.25	
70	587	22.25	55.50	1234.88	495.0625	3080.25	
75	608	27.25	76.50	2084.63	742.5625	5852.25	
80	634	32.25	102.50	3305.63	1040.0625	10506.25	
85	660	37.25	128.50	4786.63	1387.5625	16512.25	
90	694	42.25	162.50	6865.63	1785.0625	26406.25	
100	730	52.25	198.50	10371.63	2730.0625	39402.25	
$\bar{b}_a = 47.75$	$\bar{f}_c = 531.50$			54557.50	17123.75	178129.00	

Table E.14 Relation between the front wheel total toe-in angle (deg.) and tire wear (mm) using by Pearson's correlation coefficient formula

Front Wheel Total Toe-in Angle (deg.) α_i	Tire wear, mm (CG1) t_{cgw}	$(\alpha_i - \bar{\alpha})$	$(t_{cgw} - \bar{t}_{cgw})$	$(\alpha_i - \bar{\alpha}) \times (t_{cgw} - \bar{t}_{cgw})$	$(\alpha_i - \bar{\alpha})^2$	$(t_{cgw} - \bar{t}_{cgw})^2$	$r_{at_{cgw}}$
0.00	0.04	-2.00	-1.17	2.35	4.00	1.3767	0.97
0.40	0.06	-1.60	-1.15	1.85	2.56	1.3302	
1.00	0.08	-1.00	-1.13	1.13	1.00	1.2844	
1.60	0.35	-0.40	-0.86	0.35	0.16	0.7453	
4.20	2.40	2.20	1.19	2.61	4.84	1.4082	
4.80	4.35	4.80	4.35	20.88	23.04	18.9225	
$\bar{\alpha} = 2.00$	$\bar{t}_{cgw} = 1.21$			29.16	35.60	25.07	

Table E.15 Relation between the front wheel total toe-out angle (deg.) and tire wear (mm) using by Pearson's correlation coefficient formula

Front Wheel Total Toe-out Angle (deg.) α_i	Tire wear, mm (CG4) t_{cgw}	$(\alpha_i - \bar{\alpha})$	$(t_{cgw} - \bar{t}_{cgw})$	$(\alpha_i - \bar{\alpha}) \times (t_{cgw} - \bar{t}_{cgw})$	$(\alpha_i - \bar{\alpha})^2$	$(t_{cgw} - \bar{t}_{cgw})^2$	$r_{at_{cgw}}$
0.00	0.04	2.00	-1.18	-2.35	4.00	1.3806	0.97
-0.40	0.07	1.60	-1.15	-1.83	2.56	1.3110	
-1.00	0.09	1.00	-1.13	-1.13	1.00	1.2656	
-1.60	0.37	0.40	-0.85	-0.34	0.16	0.7140	
-4.20	2.42	-2.20	1.21	-2.65	4.84	1.4520	
-4.80	4.30	-4.80	4.30	-20.64	23.04	18.4900	
$\bar{\alpha} = -2.00$	$\bar{t}_{cgw} = 1.22$			-28.94	35.60	24.61	

APPENDIX–F: EXPERIMENTAL DATA FOR TRAVELLING COST

Table F.1 Experimental data for 20 test runs for different front wheel left toe-out angles and travelling costs

Front Wheel Left Toe-out Angle (deg.)	Travelling Cost Per km $T_{C/km} = f_c \times f_p$		Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km (min)}}{T_{C/km (min)}}$
	USD	BDT	
0.00	0.0645	5.1620	0.00
-0.08	0.0655	5.2362	1.44
-0.20	0.0668	5.3400	3.45
-0.35	0.0682	5.4587	5.75
-0.50	0.0697	5.5773	8.05
-0.65	0.0706	5.6515	9.48
-0.80	0.0723	5.7850	12.07
-0.95	0.0736	5.8888	14.08
-1.10	0.0749	5.9927	16.09
-1.25	0.0764	6.1113	18.39
-1.40	0.0775	6.2003	20.11
-1.55	0.0792	6.3338	22.70
-1.70	0.0807	6.4525	25.00
-1.84	0.0821	6.5712	27.30
-2.00	0.0838	6.7047	29.89
-2.14	0.0851	6.8085	31.90
-2.25	0.0866	6.9272	34.20
-2.36	0.0881	7.0458	36.49
-2.43	0.0897	7.1793	39.08
-2.53	0.0918	7.3425	42.24

Table F.2 Experimental data for 20 test runs for different front wheel total toe-out angle and travelling costs.

Front Wheel Total Toe-out Angle (deg.)	Travelling Cost Per km $T_{C/km} = f_c \times f_p$		Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km (min)}}{T_{C/km (min)}}$
	USD	BDT	
0.00	0.0645	5.1620	0.00
-0.16	0.0664	5.3103	2.87
-0.40	0.0690	5.5180	6.90
-0.70	0.0712	5.6960	10.34
-1.00	0.0731	5.8443	13.22
-1.30	0.0747	5.9778	15.80
-1.60	0.0769	6.1558	19.25
-1.90	0.0790	6.3190	22.41
-2.20	0.0814	6.5118	26.15
-2.50	0.0840	6.7195	30.17
-2.80	0.0862	6.8975	33.62
-3.10	0.0890	7.1200	37.93
-3.40	0.0914	7.3128	41.67
-3.68	0.0947	7.5798	46.84
-4.00	0.0979	7.8320	51.72
-4.28	0.1012	8.0990	56.90
-4.50	0.1046	8.3660	62.07
-4.72	0.1077	8.6182	66.95
-4.86	0.1114	8.9148	72.70
-5.06	0.1157	9.2560	79.31

Table F.3 Experimental data for 21 test runs for different tire pressure at -1.44° of front wheel total toe-out angles and travelling costs.

Tire Pressure (psi)	Travelling Cost Per km $T_{C/km} = f_c \times f_p$		Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km (min)}}{T_{C/km (min)}}$
	USD	BDT	
22	0.0942	7.5353	43.91
23	0.0907	7.2535	38.53
24	0.0890	7.1200	35.98
25	0.0860	6.8827	31.44
26	0.0840	6.7195	28.33
27	0.0816	6.5267	24.65
28	0.0795	6.3635	21.53
29	0.0788	6.3042	20.40
30	0.0775	6.2003	18.41
31	0.0769	6.1558	17.56
32	0.0755	6.0372	15.30
33	0.0742	5.9333	13.31
34	0.0729	5.8295	11.33
35	0.0721	5.7702	10.20
36	0.0714	5.7108	9.07
37	0.0706	5.6515	7.93
38	0.0701	5.6070	7.08
39	0.0692	5.5328	5.67
40	0.0682	5.4587	4.25
41	0.0668	5.3400	1.98
42	0.0655	5.2362	0.00

Table F.4 Experimental data for 20 test runs for different vehicle load condition at - 1.44° of front wheel total toe-out angles and travelling costs.

Vehicle Load Condition (kg)	Travelling Cost Per km $T_{C/km} = f_c \times f_p$		Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km (min)}}{T_{C/km (min)}}$
	USD	BDT	
1035	0.0697	5.5773	0.00
1053	0.0714	5.7108	2.39
1071	0.0734	5.8740	5.32
1089	0.0753	6.0223	7.98
1107	0.0771	6.1707	10.64
1125	0.0794	6.3487	13.83
1143	0.0805	6.4377	15.43
1161	0.0825	6.6008	18.35
1179	0.0842	6.7343	20.74
1197	0.0862	6.8975	23.67
1215	0.0879	7.0310	26.06
1233	0.0899	7.1942	28.99
1251	0.0920	7.3573	31.91
1269	0.0944	7.5502	35.37
1287	0.0968	7.7430	38.83
1305	0.0994	7.9507	42.55
1323	0.1020	8.1583	46.28
1341	0.1040	8.3215	49.20
1359	0.1072	8.5737	53.72
1377	0.1109	8.8703	59.04

Table F.5 Experimental data for 20 test runs for different brake application at -1.44° of front wheel total toe-out angles and travelling costs.

Brake Application (times)	Travelling Cost Per km $T_{C/km} = f_c \times f_p$		Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km (min)}}{T_{C/km (min)}}$
	USD	BDT	
0	0.0731	5.8443	0.00
5	0.0755	6.0372	3.30
10	0.0784	6.2745	7.36
15	0.0805	6.4377	10.15
20	0.0833	6.6602	13.96
25	0.0855	6.8382	17.01
30	0.0879	7.0310	20.30
35	0.0901	7.2090	23.35
40	0.0925	7.4018	26.65
45	0.0946	7.5650	29.44
50	0.0968	7.7430	32.49
55	0.0994	7.9507	36.04
60	0.1024	8.1880	40.10
65	0.1057	8.4550	44.67
70	0.1088	8.7072	48.98
75	0.1127	9.0187	54.31
80	0.1176	9.4043	60.91
85	0.1224	9.7900	67.51
90	0.1287	10.2943	76.14
100	0.1354	10.8283	85.28

Table F.6 Experimental data for 6 test runs for different front wheel total toe-out angle and tire travelling cost.

Front Wheel Total Toe-out Angle (deg.)	Tire Travelling Life, $(T_{tl}) = \frac{T_d \times T_{cgd}}{T_w}$ (km)	Tire Travelling Cost Per km $T_{tc/km} = \frac{T_p}{T_{tl}}$		Travelling Cost Rate Per km $\frac{T_{C/km} - T_{C/km (min)}}{T_{C/km (min)}}$
		USD	BDT	
0.00	92250.00	0.0006	0.0618	0.00
-0.40	52714.29	0.0011	0.1081	75.00
-1.00	41000.00	0.0014	0.1390	125.00
-1.60	9972.97	0.0057	0.5715	825.00
-4.20	1524.79	0.0374	3.7382	5950.01
-4.80	823.81	0.0692	6.9191	11097.97

APPENDIX-G: REGRESSION MODEL

Table G.1(a) Regression model and the R square, adjusted R square and p-values for the front wheel left toe-in angle for simple correlation.

SUMMARY OUTPUT	
<i>Regression Statistics</i>	
Multiple R	0.997439174
R Square	0.994884905
Adjusted R Square	0.994600733
Standard Error	3.152097178
Observations	20

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	1	34784.9071	34784.9071	3500.996299	4.45615E-22
Residual	18	178.8428991	9.935716617		
Total	19	34963.75			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	344.0713006	1.329181717	258.8594894	1.34765E-33	341.2787934	346.8638077	341.2787934	346.8638077
Front Left Toe-in angle (deg.)	51.13397195	0.864198888	59.16921749	4.45615E-22	49.31835746	52.94958644	49.31835746	52.94958644

Table G.1(b) Regression model and the R square, adjusted R square and p-values for the front wheel left toe-in angle for two variable correlation.

SUMMARY OUTPUT					
Regression Statistics					
Multiple R	0.999820132				
R Square	0.999640295				
Adjusted R Square	0.999597977				
Standard Error	0.860117014				
Observations	20				

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	2	34951.17338	17475.58669	23622.00122	5.31546E-30
Residual	17	12.57662172	0.739801278		
Total	19	34963.75			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	169.8920013	11.62421134	14.61535723	4.67231E-11	145.3670591	194.4169435	145.3670591	194.4169435
Front Left Toe-in angle (deg.)	36.25484278	1.020135352	35.53924752	2.13968E-17	34.10254533	38.40714024	34.10254533	38.40714024
Engine RPM	0.162988704	0.010872088	14.99148132	3.12419E-11	0.140050603	0.185926805	0.140050603	0.185926805

Table G.1(c) Regression model and the R square, adjusted R square and p-values for the front wheel left toe-in angle for five variable correlation.

SUMMARY OUTPUT	
Regression Statistics	
Multiple R	0.999881787
R Square	0.999763587
Adjusted R Square	0.999679154
Standard Error	0.768388142
Observations	20

ANOVA					
		<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>
Regression		5	34955.48412	6991.096823	11840.88078
Residual		14	8.265884724	0.590420337	7.34701E-25
Total		19	34963.75		

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	160.2802329	41.3676353	3.87453215	0.001684127	71.55547943	249.0049864	71.55547943	249.0049864
Front Left Toe-in angle (deg.)	36.11015186	1.308894417	27.58828473	1.32412E-13	33.30285253	38.91745118	33.30285253	38.91745118
Engine RPM	0.175167315	0.048375112	3.621021425	0.002780095	0.07141302	0.278921611	0.07141302	0.278921611
Rolling Resistance Coefficient	-3072.705761	8189.670328	-0.375192851	0.713140031	-20637.80166	14492.39014	-20637.80166	14492.39014
Rolling Resistance Force, (N)	-12859.14726	5741.256053	-2.239779438	0.041852281	-25172.91681	-545.3777036	-25172.91681	-545.3777036
Energy Consumption	2143.23636	956.8174547	2.239963694	0.04183776	91.06702055	4195.4057	91.06702055	4195.4057

Table G.2(a) Regression model and the R square, adjusted R square and p-values for front wheel total toe-in angle for simple correlation.

SUMMARY OUTPUT	
Regression Statistics	
Multiple R	0.993360528
R Square	0.986765138
Adjusted R Square	0.986029868
Standard Error	9.578972401
Observations	20

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	1	123141.5792	123141.5792	1342.044371	2.32462E-18
Residual	18	1651.620821	91.75671226		
Total	19	124793.2			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	337.3432434	4.039277429	83.51573997	9.21143E-25	328.8570364	345.8294503	328.8570364	345.8294503
Front total toe-in angle (deg.)	48.1045846	1.313115813	36.63392377	2.32462E-18	45.34583064	50.86333855	45.34583064	50.86333855

Table G.2(b) Regression model and the R square, adjusted R square and p-values for front wheel total toe-in angle for two variable correlation.

SUMMARY OUTPUT									
Regression Statistics									
Multiple R	0.999302036								
R Square	0.998604558								
Adjusted R Square	0.998440389								
Standard Error	3.200566427								
Observations	20								

ANOVA									
	df	SS	MS	F	Significance F				
Regression	2	124619.0584	62309.52918	6082.761367	5.37093E-25				
Residual	17	174.1416328	10.24362546						
Total	19	124793.2							

	Coefficients	Standard Error	t Stat	P-value	Lower 95%	Upper 95%	Lower 95.0%	Upper 95.0%
Intercept	151.4921632	15.53376168	9.752445435	2.23291E-08	118.7187908	184.2655356	118.7187908	184.2655356
Front total toe-in angle (deg.)	26.77254947	1.829611485	14.63291507	4.58444E-11	22.91240666	30.63269228	22.91240666	30.63269228
Engine RPM	0.18241117	0.015188595	12.00974651	9.9381E-10	0.150366037	0.214456304	0.150366037	0.214456304

Table G.2(c) Regression model and the R square, adjusted R square and p-values for front wheel total toe-in angle for five variable correlation.

SUMMARY OUTPUT					
Regression Statistics					
Multiple R	0.999470365				
R Square	0.998941011				
Adjusted R Square	0.873742451				
Standard Error	2.873962993				
Observations	20				

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	5	124661.0454	24932.20908	5030.917164	2.93166E-22
Residual	16	132.1546126	8.259663287		
Total	21	124793.2			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	-21.74833029	78.09328113	-0.278491696	0.784202014	-187.2986908	143.8020302	-187.2986908	143.8020302
Front total toe-in angle (deg.)	25.933335352	1.684543211	15.39488767	5.17165E-11	22.36228144	29.5044256	22.36228144	29.5044256
Engine RPM	0.399728948	0.097347243	4.106217458	0.000825719	0.193362012	0.606095884	0.193362012	0.606095884
Rolling Resistance Coefficient	0	0	65535	#NUM!	0	0	0	0
Rolling Resistance Force, (N)	0	0	65535	#NUM!	0	0	0	0
Energy Consumption	-0.049201987	0.02182259	-2.254635575	#NUM!	-0.095463811	-0.002940164	-0.095463811	-0.002940164

Table G.3(a) Regression model and the R square, adjusted R square and p-values for tire pressure at 0.00° for simple correlation.

SUMMARY OUTPUT	
<i>Regression Statistics</i>	
Multiple R	0.979131079
R Square	0.95869767
Adjusted R Square	0.956523863
Standard Error	7.58999023
Observations	21

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	1	25406.4013	25406.4013	441.0224726	1.30883E-14
Residual	19	1094.551082	57.6079517		
Total	20	26500.95238			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	535.8606061	8.908104737	60.15427769	3.78764E-23	517.2157286	554.5054836	517.2157286	554.5054836
Tire Pressure	-5.744155844	0.273524262	-21.00053505	1.30883E-14	-6.316648703	-5.171662985	-6.316648703	-5.171662985

Table G.3(b) Regression model and the R square, adjusted R square and p-values for tire pressure at 0.00° for two variable correlation.

SUMMARY OUTPUT	
<i>Regression Statistics</i>	
Multiple R	0.992921146
R Square	0.985892401
Adjusted R Square	0.98432489
Standard Error	4.557440805
Observations	21

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	2	26127.08758	13063.54379	628.9540708	2.21349E-17
Residual	18	373.8648005	20.77026669		
Total	20	26500.95238			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	-682.1026369	206.8365858	-3.297785226	0.004000818	-1116.650179	-247.555095	-1116.650179	-247.555095
Tire Pressure	2.320786079	1.378959712	1.682997741	0.109639612	-0.576300772	5.21787293	-0.576300772	5.21787293
Engine RPM	0.882479079	0.149813975	5.890499062	1.41141E-05	0.567731597	1.197226561	0.567731597	1.197226561

Table G.3(c) Regression model and the R square, adjusted R square and p-values for tire pressure at 0.00° for five variable correlation.

SUMMARY OUTPUT					
Regression Statistics					
Multiple R	0.997823209				
R Square	0.995651156				
Adjusted R Square	0.994201541				
Standard Error	2.771864835				
Observations	21				

ANOVA					
	<i>df</i>	SS	MS	<i>F</i>	Significance <i>F</i>
Regression	5	26385.70386	5277.140772	686.838422	3.75174E-17
Residual	15	115.2485199	7.683234663		
Total	20	26500.95238			

	<i>Coefficients</i>	Standard Error	<i>t Stat</i>	<i>P-value</i>	Lower 95%	Upper 95%	Lower 95.0%	Upper 95.0%
Intercept	1839.909726	503.122882	3.65697878	0.00233619	767.5286746	2912.290777	767.5286746	2912.290777
Tire Pressure	-12.03407166	2.92171013	-4.118845172	0.000910801	-18.26154939	-5.806593935	-18.26154939	-5.806593935
Engine RPM	-1.389646154	0.444156135	-3.128733445	0.006899968	-2.336342545	-0.442949762	-2.336342545	-0.442949762
Rolling Resistance Coefficient	-19361.81679	23668.80578	-0.818030997	0.426143901	-69810.6821	31087.04853	-69810.6821	31087.04853
Rolling Resistance Force, (N)	66404.13208	32808.77863	2.023974523	0.061162733	-3526.124237	136334.3884	-3526.124237	136334.3884
Energy Consumption	-11066.75262	5468.041314	-2.023897039	0.061171545	-22721.6068	588.1015511	-22721.6068	588.1015511

Table G.4(a) Regression model and the R square, adjusted R square and p-values for tire pressure at 1.44° for simple correlation.

SUMMARY OUTPUT	
<i>Regression Statistics</i>	
Multiple R	0.985850563
R Square	0.971901334
Adjusted R Square	0.970422456
Standard Error	7.114525494
Observations	21

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	1	33264.57273	33264.57273	657.1886731	3.34933E-16
Residual	19	961.712987	50.616473		
Total	20	34226.28571			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	618.612987	8.350068488	74.08478001	7.35497E-25	601.1360928	636.0898812	601.1360928	636.0898812
Tire Pressure	-6.572727273	0.256389702	-25.63569139	3.34933E-16	-7.109357086	-6.036097459	-7.109357086	-6.036097459

Table G.4(b) Regression model and the R square, adjusted R square and p-values for tire pressure at 1.44° for two variable correlation.

SUMMARY OUTPUT	
<i>Regression Statistics</i>	
Multiple R	0.999800542
R Square	0.999601124
Adjusted R Square	0.999556805
Standard Error	0.870888403
Observations	21

ANOVA				
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>
Regression	2	34212.63368	17106.31684	22554.41134
Residual	18	13.65203899	0.758446611	
Total	20	34226.28571		
				<i>Significance F</i>
				2.55588E-31

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	218.6554941	11.35857326	19.25026049	1.86093E-13	194.7920172	242.518971	194.7920172	242.518971
Tire Pressure	-3.518483737	0.091911356	-38.28127336	1.06326E-18	-3.71158233	-3.325385144	-3.71158233	-3.325385144
Engine RPM	0.2554603	0.007225498	35.35538912	4.36954E-18	0.240280092	0.270640508	0.240280092	0.270640508

Table G.4(c) Regression model and the R square, adjusted R square and p-values for tire pressure at 1.44° for five variable correlation.

SUMMARY OUTPUT	
<i>Regression Statistics</i>	
Multiple R	0.999845889
R Square	0.999691801
Adjusted R Square	0.999589068
Standard Error	0.838590315
Observations	21

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	5	34215.73721	6843.147442	9730.971771	9.01549E-26
Residual	15	10.54850574	0.703233716		
Total	20	34226.28571			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	282.4920259	46.41005877	6.086870679	2.08045E-05	183.5713273	381.4127246	183.5713273	381.4127246
Tire Pressure	-3.628299445	0.114673751	-31.64019156	3.78586E-15	-3.872720759	-3.38387813	-3.872720759	-3.38387813
Engine RPM	0.184007956	0.051391397	3.58052058	0.002732679	0.074469785	0.293546127	0.074469785	0.293546127
Rolling Resistance Coefficient	-330.0107558	6796.567057	-0.048555506	0.96191415	-14816.55052	14156.52901	-14816.55052	14156.52901
Rolling Resistance Force, (N)	-6124.337761	6226.082586	-0.983658292	0.340884697	-19394.91866	7146.243138	-19394.91866	7146.243138
Energy Consumption	1020.742659	1037.66978	0.983687371	0.340870857	-1190.998123	3232.48344	-1190.998123	3232.48344

Table G.5(a) Regression model and the R square, adjusted R square and p-values for vehicle load at 0.00° for simple correlation.

SUMMARY OUTPUT	
<i>Regression Statistics</i>	
Multiple R	0.994499404
R Square	0.989029065
Adjusted R Square	0.988419568
Standard Error	6.103226477
Observations	20

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	1	60444.51128	60444.51128	1622.698738	4.29164E-19
Residual	18	670.4887218	37.24937343		
Total	19	6115			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	-234.2669173	15.91570743	-14.71922743	1.76763E-11	-267.7045778	-200.8292568	-267.7045778	-200.8292568
Vehicle Load	0.529657477	0.013148498	40.28273499	4.29164E-19	0.502033507	0.557281447	0.502033507	0.557281447

Table G.5(b) Regression model and the R square, adjusted R square and p-values for vehicle load at 0.00° for two variable correlation.

SUMMARY OUTPUT	
Regression Statistics	
Multiple R	0.999374522
R Square	0.998749435
Adjusted R Square	0.99860231
Standard Error	2.120325302
Observations	20

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	2	61038.57175	30519.28588	6788.430491	2.11545E-25
Residual	17	76.42824959	4.495779388		
Total	19	61115			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	-233.0405225	5.530313952	-42.13875098	1.21965E-18	-244.708465	-221.37258	-244.708465	-221.37258
Vehicle Load	0.342996143	0.016868596	20.33341392	2.2837E-13	0.307406516	0.378585769	0.307406516	0.378585769
Engine RPM	0.188982168	0.016440234	11.49510208	1.93663E-09	0.154296306	0.22366803	0.154296306	0.22366803

Table G.5(c) Regression model and the R square, adjusted R square and p-values for vehicle load at 0.00° for five variable correlation.

SUMMARY OUTPUT					
Regression Statistics					
Multiple R	0.999840992				
R Square	0.999682009				
Adjusted R Square	0.874622385				
Standard Error	1.102101347				
Observations	20				

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	5	61095.56596	12219.11319	16766.6142	6.44069E-26
Residual	16	19.43403806	1.214627379		
Total	21	61115			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	-64.78135307	24.73082421	-2.619457909	0.018590703	-117.2083584	-12.35434777	-117.2083584	-12.35434777
Vehicle Load	0.428341236	0.015234982	28.11563704	4.75483E-15	0.396044516	0.460637956	0.396044516	0.460637956
Engine RPM	-0.121780069	0.046164197	-2.637976556	0.01790273	-0.219643794	-0.023916343	-0.219643794	-0.023916343
Rolling Resistance Coefficient	0	0	65535	#NUM!	0	0	0	0
Rolling Resistance Force, (N)	0	0	65535	#NUM!	0	0	0	0
Energy Consumption	0.057045263	0.008327713	6.850051582	#NUM!	0.0393913	0.074699225	0.0393913	0.074699225

Table G.6(a) Regression model and the R square, adjusted R square and p-values for vehicle load at 1.44° for simple correlation.

SUMMARY OUTPUT	
<i>Regression Statistics</i>	
Multiple R	0.99805757
R Square	0.996118914
Adjusted R Square	0.995903298
Standard Error	4.139152263
Observations	20

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	1	79150.41353	79150.41353	4619.87668	3.71206E-23
Residual	18	308.3864662	17.13258145		
Total	19	79458.8			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	-261.3548872	10.79388691	-24.21323193	3.47122E-15	-284.0320021	-238.6777723	-284.0320021	-238.6777723
Vehicle Load	0.60609858	0.008917191	67.96967471	3.71206E-23	0.587364256	0.624832903	0.587364256	0.624832903

Table G.6(b) Regression model and the R square, adjusted R square and p-values for vehicle load at 1.44° for two variable correlation.

SUMMARY OUTPUT	
<i>Regression Statistics</i>	
Multiple R	0.999303355
R Square	0.998607195
Adjusted R Square	0.998443335
Standard Error	2.551477425
Observations	20

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	2	79348.12937	39674.06469	6094.291687	5.28528E-25
Residual	17	110.6706299	6.510037051		
Total	19	79458.8			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	-199.6047938	13.03153296	-15.31706165	2.22062E-11	-227.098925	-172.1106625	-227.098925	-172.1106625
Vehicle Load	0.419566956	0.034290689	12.23559416	7.46918E-10	0.347219926	0.491913987	0.347219926	0.491913987
Engine RPM	0.12297095	0.0223138	5.510982068	3.81751E-05	0.075892948	0.170048952	0.075892948	0.170048952

Table G.6(c) Regression model and the R square, adjusted R square and p-values for vehicle load at 1.44° for five variable correlation.

SUMMARY OUTPUT	
Regression Statistics	
Multiple R	0.999903565
R Square	0.999807138
Adjusted R Square	0.933089042
Standard Error	1.010760303
Observations	20

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	5	79443.47545	15888.69509	19440.252	2.28675E-26
Residual	15	15.32454584	1.02163639		
Total	20	79458.8			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	10.4295272	22.46086535	0.464342181	0.649070558	-37.44467406	58.30372845	-37.44467406	58.30372845
Vehicle Load	0.415817152	0.013599136	30.57673376	6.27601E-15	0.38683128	0.444803023	0.38683128	0.444803023
Engine RPM	-0.12884117	0.027759212	-4.641384216	0.000319674	-0.18800853	-0.069673809	-0.18800853	-0.069673809
Rolling Resistance Coefficient	4875.299079	8833.537522	0.551907893	0.589140356	-13952.94046	23703.53862	-13952.94046	23703.53862
Rolling Resistance Force, (N)	0	0	65535	#NUM!	0	0	0	0
Energy Consumption	-0.020120122	0.138337473	-0.145442314	#NUM!	-0.314979466	0.274739222	-0.314979466	0.274739222

Table G.7(a) Regression model and the R square, adjusted R square and p-values for brake application at 0.00° for simple correlation.

SUMMARY OUTPUT	
Regression Statistics	
Multiple R	0.989615511
R Square	0.979338859
Adjusted R Square	0.978191018
Standard Error	12.29128318
Observations	20

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	1	128897.8384	128897.8384	853.2006654	1.28452E-16
Residual	18	2719.361559	151.0756422		
Total	19	131617.2			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	329.7923206	5.260210482	62.69565101	1.57939E-22	318.7410285	340.8436127	318.7410285	340.8436127
Brake Application	2.743616322	0.093928586	29.20959886	1.28452E-16	2.546279686	2.940952959	2.546279686	2.940952959

Table G.7(b) Regression model and the R square, adjusted R square and p-values for brake application at 0.00° for two variable correlation.

SUMMARY OUTPUT	
Regression Statistics	
Multiple R	0.998675209
R Square	0.997352173
Adjusted R Square	0.997040663
Standard Error	4.527690151
Observations	20

ANOVA				
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>
Regression	2	131268.7004	65634.35019	3201.679038
Residual	17	348.4996278	20.49997811	
Total	19	131617.2		
				<i>Significance F</i>
				1.24326E-22

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	289.0272312	4.257173556	67.89181305	3.84926E-22	280.0453801	298.0090823	280.0453801	298.0090823
Brake Application	1.452414975	0.124951368	11.62384208	1.63537E-09	1.188790631	1.716039318	1.188790631	1.716039318
Rolling Resistance Coefficient	2924.195682	271.9130169	10.7541585	5.2791E-09	2350.509364	3497.882001	2350.509364	3497.882001

Table G.7(c) Regression model and the R square, adjusted R square and p-values for brake application at 0.00° for four variable correlation.

SUMMARY OUTPUT	
Regression Statistics	
Multiple R	0.998684812
R Square	0.997371355
Adjusted R Square	0.879415043
Standard Error	4.511259995
Observations	20

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	4	131271.2251	32817.80627	3225.104774	8.24792E-22
Residual	17	345.9749346	20.35146674		
Total	21	131617.2			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	288.9989074	4.242271271	68.12362739	3.63328E-22	280.0484974	297.9493174	280.0484974	297.9493174
Brake Application	1.452581241	0.124421994	11.67463397	1.53047E-09	1.190073779	1.715088702	1.190073779	1.715088702
Rolling Resistance Coefficient	0	0	65535	#NUM!	0	0	0	0
Rolling Resistance Force, (N)	0	0	65535	#NUM!	0	0	0	0
Energy Consumption	0.045591208	0.004221771	10.79907079	#NUM!	0.036684049	0.054498367	0.036684049	0.054498367

Table G.8(a) Regression model and the R square, adjusted R square and p-values for brake application at 1.44° for simple correlation.

SUMMARY OUTPUT	
<i>Regression Statistics</i>	
Multiple R	0.990003529
R Square	0.980106988
Adjusted R Square	0.97900182
Standard Error	13.7121542
Observations	20

ANOVA				
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>
Regression	1	166746.5329	166746.5329	886.8403314
Residual	18	3384.41711	188.0231728	9.13069E-17
Total	19	170130.95		

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	376.444485	5.868290251	64.14892054	1.0474E-22	364.1156647	388.7733053	364.1156647	388.7733053
Brake Application	3.120534346	0.104786721	29.77986453	9.13069E-17	2.900385615	3.340683077	2.900385615	3.340683077

Table G.8(b) Regression model and the R square, adjusted R square and p-values for brake application at 1.44° for two variable correlation.

SUMMARY OUTPUT					
Regression Statistics					
Multiple R	0.990484732				
R Square	0.981060004				
Adjusted R Square	0.978831769				
Standard Error	13.76756536				
Observations	20				

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	2	166908.6704	83454.33522	440.2857282	2.27892E-15
Residual	17	3222.279552	189.545856		
Total	19	170130.95			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	328.6256536	52.03746756	6.315173836	7.75714E-06	218.8361939	438.4151133	218.8361939	438.4151133
Brake Application	1.911893289	1.311039026	1.458303873	0.16298477	-0.854157271	4.677943848	-0.854157271	4.677943848
Rolling Resistance Coefficient,	2368.565635	2560.947971	0.924878467	0.367972435	-3034.562289	7771.693558	-3034.562289	7771.693558

Table G.8(c) Regression model and the R square, adjusted R square and p-values for brake application at 1.44° for four variable correlation.

SUMMARY OUTPUT	
Regression Statistics	
Multiple R	0.990727012
R Square	0.981540013
Adjusted R Square	0.915578765
Standard Error	14.01029798
Observations	20

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	4	166990.3348	41747.5837	283.5798292	6.1911E-14
Residual	16	3140.61519	196.2884494		
Total	20	170130.95			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	331.5807281	53.1527394	6.238262258	1.18521E-05	218.9019542	444.259502	218.9019542	444.259502
Brake Application	1.976634176	1.33792389	1.477389103	0.15898442	-0.859637768	4.812906121	-0.859637768	4.812906121
Rolling Resistance Coefficient	-72587.41601	116237.5814	-0.62447459	0.541119151	-319000.0808	173825.2488	-319000.0808	173825.2488
Rolling Resistance Force, (N)	0	0	65535	#NUM!	0	0	0	0
Energy Consumption	1.1662163	1.808048989	0.645013662	#NUM!	-2.666676333	4.999108933	-2.666676333	4.999108933

Table G.9(a) Regression model and the R square, adjusted R square and p-values for front wheel total toe-in angle for simple correlation.

SUMMARY OUTPUT	
Regression Statistics	
Multiple R	0.944994679
R Square	0.893014944
Adjusted R Square	0.866268679
Standard Error	0.653834692
Observations	6

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	1	14.27353412	14.27353412	33.38839921	0.004455166
Residual	4	1.709999216	0.427499804		
Total	5	15.98353333			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	-0.459607843	0.393793675	-1.16712856	0.30798983	0.633738678	-1.552954364	0.633738678
Total Toe-in Angle (deg.)	0.836470588	0.144761434	5.778269569	0.004455166	1.238392763	0.434548414	1.238392763

Table G.9 (b) Regression model and the R square, adjusted R square and P-values for front wheel total toe-in angle for two variable correlation.

SUMMARY OUTPUT	
<i>Regression Statistics</i>	
Multiple R	0.953507014
R Square	0.909175625
Adjusted R Square	0.848626042
Standard Error	0.695627875
Observations	6

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	2	14.53183891	7.265919456	15.01539032	0.027371817
Residual	3	1.451694422	0.483898141		
Total	5	15.98353333			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	12.01802963	17.08337807	0.703492575	0.532432571	-42.3489038	66.38496306	-42.3489038	66.38496306
Total Toe-in Angle (deg.)	1.619066997	1.082161656	1.496141531	0.231512075	-1.824854366	5.062988359	-1.824854366	5.062988359
RPM	-0.009233642	0.012638158	-0.73061613	0.517901689	-0.0494539	0.030986616	-0.0494539	0.030986616

Table G.9 (c) Regression model and the R square, adjusted R square and P-values for front wheel total toe-in angle for five variable correlation.

SUMMARY OUTPUT	
Regression Statistics	
Multiple R	0.992993739
R Square	0.986036566
Adjusted R Square	-0.06981717
Standard Error	0.472424611
Observations	6

ANOVA					
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>	<i>Significance F</i>
Regression	5	15.76034832	3.152069664	17.65390526	#NUM!
Residual	1	0.223185013	0.223185013		
Total	6	15.98353333			

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	-0.25322624	14.8195105	-0.017087355	0.98912291	-188.5529607	188.0465082	-188.5529607	188.0465082
Total Toe-in Angle (deg.)	-0.020158017	1.368079089	-0.014734541	0.990620379	-17.40325102	17.36293499	-17.40325102	17.36293499
RPM	-0.001856967	0.010299054	-0.180304612	0.886434688	-0.132718855	0.129004921	-0.132718855	0.129004921
Rolling Resistance Coefficient	-6142.604588	6612.331892	-0.928961929	0.523434269	-90160.24739	77875.03821	-90160.24739	77875.03821
Rolling Resistance Force, (N)	0	0	65535	#NUM!	0	0	0	0
Energy Consumption	0.111647121	0.116917448	0.954922668	#NUM!	-1.373929905	1.597224147	-1.373929905	1.597224147

APPENDIX-H: MULTIPLE CORRELATION

Table H.1 Experimental data for 20 test runs with the variation of toe angle, tire pressure, vehicle load and brake application

Tire Pressure (psi)	Vehicle Load (kg)	Brake Application (times)	Toe Angle (deg.)	Fuel Consumption (ml)
32	1089	5	0.08	356
31	1089	5	0.08	361
30	1089	5	0.08	367
29	1089	5	0.08	375
28	1089	5	0.08	381
32	1089	5	0.08	356
32	1107	5	0.08	360
32	1125	5	0.08	366
32	1143	5	0.08	373
32	1161	5	0.08	382
32	1089	5	0.08	356
32	1089	10	0.08	362
32	1089	15	0.08	367
32	1089	20	0.08	378
32	1089	25	0.08	384
32	1089	5	0.08	356
32	1089	5	0.40	363
32	1089	5	0.70	369
32	1089	5	1.00	381
32	1089	5	1.30	390

Table H.2 Regression model and the R square, adjusted R square and P-values for toe angle, tire pressure, vehicle load and brake application.

SUMMARY OUTPUT	
<i>Regression Statistics</i>	
Multiple R	0.992243923
R Square	0.984548004
Adjusted R Square	0.980427471
Standard Error	1.498064236
Observations	20

ANOVA				
	<i>df</i>	<i>SS</i>	<i>MS</i>	<i>F</i>
Regression	4	2144.887053	536.2217633	238.9370868
Residual	15	33.66294683	2.244196455	2.192E-13
Total	19	2178.55		

	<i>Coefficients</i>	<i>Standard Error</i>	<i>t Stat</i>	<i>P-value</i>	<i>Lower 95%</i>	<i>Upper 95%</i>	<i>Lower 95.0%</i>	<i>Upper 95.0%</i>
Intercept	168.6496692	19.76898934	8.531021303	3.86814E-07	126.5130659	210.7862726	126.5130659	210.7862726
Tire Pressure	-6.549876606	0.335218905	-19.53910271	4.42202E-12	-7.264378788	-5.835374425	-7.264378788	-5.835374425
Vehicle Load	0.354622774	0.018623272	19.04191515	6.41807E-12	0.314928209	0.39431734	0.314928209	0.39431734
Brake Application	1.456641988	0.067043781	21.72672794	9.47379E-13	1.313741552	1.599542424	1.313741552	1.599542424
Toe Angle	27.86676348	1.094627072	25.45776931	9.33022E-14	25.53362111	30.19990586	25.53362111	30.19990586