

ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL

**RIDE QUALITY AND HANDLING CHARACTERISTICS OF AN OFF-ROAD
VEHICLE WITH ACTIVE ANTI-ROLL BAR SUSPENSION**



M.Sc. THESIS

Berkan CANPOLAT

Department of Mechanical Engineering

Automotive Programme

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OCAK 2024

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**AKTİF VİRAJ DEMİRİ SÜSPANSİYONU OLAN ARAZİ ARACINDA SÜRÜŞ
KALİTESİ VE YOL TUTUŞU ÖZELLİKLERİ**

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To my family,



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ABBREVIATIONS

ARB	: Anti-roll bar
BS	: British Standards
DC	: Direct current
ECU	: Electronic control unit
EMAS	: Electromagnetic active suspension system
EMS	: Electromechanical suspension system
ESVH	: Electronic servo-valve hydraulic
FED	: Fuel Efficiency Demonstrator
Freq	: Frequency
IRI	: International Roughness Index
ISO	: International Organization for Standardization
MR	: Magneto-rheological
MS	: Microsoft
NVH	: Noise, vibration, and harshness
PID	: Proportional integral derivative
PSD	: Power spectral density
Rms	: Root mean square
SAE	: Society of Automotive Engineers
SUV	: Sport utilized vehicle
VDI	: Association of German Engineers
VDV	: Vibration dose value



SYMBOLS

a_v	: The total vibration value of weighted rms acceleration
a_w	: The weighted acceleration as a function of time expressed in meters per second squared or radians per second squared
a_{wx}, a_{wy}, a_{wz}	: The weighted rms accelerations with respect to orthogonal axes x, y, z, respectively
cm	: Centimeter
deg, °	: Degree
E	: Elastic modulus, MPa
f_A	: Deflection at bar end
f_w	: Wheel travel
ft	: Feet
h	: Hour
Hz	: Hertz
I	: Moment of inertia, mm ⁴
G	: Modulus of rigidity
g	: Gravitational constant, 9.81 m/s ²
J	: Polar moment inertia
k_{bar}	: Roll rate of the anti-roll bar, Nmm/rad
kg	: Kilogram
km	: Kilometer
k_R	: Bar roll rate, Nmm/rad
k_{RW}	: Effective roll rate at wheels, Nmm/rad
kW	: Kilowatt
k_x, k_y, k_z	: Multiplying factors
L	: Liter, Bar length
M	: Bending moment
m	: Meter, Bending moment equations due to a force of one Newton acting at bar end
mm	: Milimeter
N	: Newton

P	: Applied load at bar end, N
R	: Radial tire
rad	: Radian
s	: Second
T	: The measurement period in seconds, Torsional moment, Wheel tread
t	: Time, Torsional moment equations due to a force of one Newton acting at bar end
v	: Speed, m/s^2
W	: Watt
%	: Percentage



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RIDE QUALITY AND HANDLING CHARACTERISTICS OF AN OFF-ROAD VEHICLE WITH ACTIVE ANTI-ROLL BAR SUSPENSION

SUMMARY

A vehicle's suspension system isolates the car from road vibrations, resulting in a comfortable ride. In addition, it provides sufficient handling in the vehicle and helps the vehicle to drive stably and reliably. The anti-roll bar system contributes significantly to safety by reducing body roll during cornering. However, when used passively, this system negatively affects the vehicle's ride quality. Active anti-roll bar systems can adjust stiffness according to the driver's input or road conditions. Most current studies have focused on this feature to improve the stability and handling of the vehicle. However, ride quality is essential for many reasons in military vehicles. It is possible to observe effects such as loss of attention and discomfort and the emergence of health problems when the driver is exposed to poor road conditions.

In this study, it is desired to examine the effect of the anti-roll bar on the system. According to the results, the goal is to develop a controller for a vehicle with better ride quality. In this context, analyses were made on roads with the same waviness and roughness values to compare each result clearly. In addition, analyses were made on roads with different phase angles to observe the vehicle's response to various conditions.

It would need to be important to check essential parameters such as road holding and stability by increasing the ride quality while performing analyses on the off-road vehicle. As a result, analyses of these suspension systems, which are essential for vehicle dynamics such as cornering and lane change, should also be made. The aim is to create a system that will not adversely affect road holding and stability while contributing to the improvement of ride quality.

Four different suspension systems were installed in the analyses to examine the ride quality with different anti-roll bar thicknesses. One of them is a thicker anti-roll bar. A front suspension produces higher torsional stiffness and load transfer in this case.

The second is the anti-roll bar, which is actively used in the vehicle. The third one is a lower-thickness anti-roll bar. Thus, the results in the case where the effect of the anti-roll bar is reduced wanted to be examined. Finally, the anti-roll bar has been removed from the suspension system. The anti-roll bar model used does not have a complex structure. For this reason, the calculation method created by the SAE (Society of Automotive Engineers) was used while calculating the torsional stiffness.

There are standards and methods such as ISO 2631, BS 6841, absorbed power, and VDI 2057 to evaluate ride quality. Within the scope of this study, the absorbed power method, frequency-weighted RMS, and vibration dose value (VDV) have been used to evaluate ride quality. The absorbed power is expressed in Watts to represent how much power the human body absorbs to measure the level of discomfort. In this method, which shows a number mathematically, high values indicate an increase in discomfort, while the limit value is determined as 6 Watts. In calculating the comfort level, getting data from the driver or passenger locations is more accurate. Therefore, acceleration data at the driver's location were taken for calculations. In order to extract acceleration values, first of all, analyses should be made in the multibody dynamics program. Then, with the acceleration values obtained from here, the absorbed power value can be transformed with the transfer functions specified by TARADCOM. Graphical programming and numeric computing environments were used simultaneously for this transformation, and the transfer functions provided are analyzed. Absorbed power values in all three global axes were obtained, and the results were transferred to tables and graphics. Another assessment criterion was ISO 2631. Examining the frequency-weighted RMS and vibration dose value (VDV) results, comments on ride quality were given. To determine these values, the signal processing application examined the acceleration data in all three axes.

While analyzing the ride quality, the vehicle travels 300 meters to make the assessment correctly. In addition, the speed values between 12.87 km/h and 32.19 km/h were examined to observe different speed conditions. In order to examine different phase conditions, analyses were made on the paths at 0, 45, 90, and 180-degree phase angles, and their effects were examined. Anti-roll bars with different torsional stiffness values have been investigated for their impact on ride quality.

According to the results obtained, the change in thickness did not have much effect on ride quality at a 0-degree phase angle due to the low effect of the anti-roll bar on the

system. It also allows the vehicle to remain more stable at this phase angle. With the increase of the phase angle, significant increases were observed in the absorbed power values of the lateral axis. It has been determined that this difference widens even more, especially in the 180-degree phase angle. The presence of an anti-roll bar significantly increases the discomfort in the lateral axis. In addition, it was determined that the highest absorbed power values were on the vertical axis. It has been observed that the anti-roll bar has a negative effect on the vertical axis at the 180-degree phase angle. As a result, it has been deduced that the anti-roll bar has negatively affected ride quality, especially with the increase in the phase angle. According to the results, the lowest absorbed power values were achieved in the vehicle's longitudinal axis (x-axis). When frequency-weighted RMS and vibration dose value (VDV) values were also analyzed, it was found that similar results were obtained with the absorbed power method.

In vehicle dynamics analyses such as cornering and double-lane change, it has been observed that vehicles with anti-roll bars in the front suspension have lower roll angle values. In addition, vehicles with and without an anti-roll bar were subjected to lane change analyses by the ISO 3888-2 standard. This analysis consists of an obstacle avoidance course and is based on double-lane change. According to ISO 3888-2 results, it has been observed that the vehicle with an anti-roll bar can complete the track at higher speeds.

When the results of the model with active anti-roll bars operating according to the specified algorithm are analyzed, it is observed that the ride quality values of the vehicles have improved considerably, especially at high phase angles. In the analyses where the handling is evaluated, it is observed that the vehicles with the active anti-roll bar system produced similar results to the standard vehicle and performed better than the vehicles without an anti-roll bar. As a result, while the ride quality of the vehicle with the active system is better, there is no change in handling characteristics.



AKTİF VİRAJ DEMİRİ SÜSPANSİYONU OLAN ARAZİ ARACINDA SÜRÜŞ KALİTESİ VE YOL TUTUŞU ÖZELLİKLERİ

ÖZET

Bir aracın süspansiyon sistemi, aracı yol titreşimlerinden izole ederek konforlu bir sürüş sağlar. Ayrıca, araçta yeterli yol tutuşunu sağlamasının yanı sıra aracın dengeli ve güvenilir bir şekilde sürülmesine yardımcı olur. Viraj denge çubuğu sistemi, viraj alma sırasında gövde yalpalamasını azaltarak güvenliğe önemli ölçüde katkıda bulunur. Ancak pasif olarak kullanıldığında bu sistem aracın sürüş kalitesini olumsuz etkileyebilmektedir. Aktif viraj denge çubuğu sistemleri, sertliği sürücünün girdisine veya yol koşullarına göre ayarlayabilir. Mevcut çalışmaların çoğu, aracın dengesini ve yol tutuşunu iyileştirmek için bu özelliğe odaklanmıştır. Ancak, askeri araçlarda sürüş kalitesi birçok nedenden dolayı önemlidir. Sürücü, kötü yol koşullarına maruz kaldığında dikkat kaybı, rahatsızlık ve sağlık sorunları gibi sorunların ortaya çıkması mümkündür.

Bu çalışmada viraj denge çubuğunun sistem üzerindeki etkisi incelenmek istenmiştir. Elde edilen sonuçlara göre daha iyi sürüş kalitesine sahip bir araç için kontrolcü geliştirilmesi hedeflenmektedir. Bu kapsamda, her bir sonucun net bir şekilde karşılaştırılabilmesi için aynı dalgalılık ve pürüzlülük değerlerine sahip yollarda analizler yapılmıştır. Ayrıca, aracın çeşitli koşullara verdiği tepkiyi gözlemlemek için farklı faz açılarına sahip yollarda analizler yapılmıştır.

Arazi aracı üzerinde analizler gerçekleştirirken sürüş kalitesini artırmanın yanı sıra yol tutuşu ve denge gibi temel parametreleri kontrol etmek önemli olacaktır. Sonuç olarak viraj alma ve şerit değiştirme gibi araç dinamikleri için elzem olan bu süspansiyon sistemlerinin analizleri de yapılmalıdır. Amaç, sürüş kalitesinin iyileştirilmesine katkı sağlarken yol tutuşunu ve dengesini olumsuz etkilemeyecek bir sistem oluşturmaktır.

Farklı viraj denge çubuğu kalınlıkları ile sürüş kalitesini incelemek için analizlerde dört farklı süspansiyon sistemi kurulmuştur. Bunlardan biri daha kalın bir viraj denge çubuğudur. Ön süspansiyon bu durumda daha yüksek burulma sertliği ve yük transferi

 retir. İkincisi, ara ta aktif olarak kullanılan viraj denge  ubu udur.          ise daha d     kalınlıkta bir viraj denge  ubu udur. B  ylece viraj denge  ubu unun etkisinin azaltıldı ı durumdaki sonu lar incelenmek istenmi tir. Son olarak viraj denge  ubu u s  spansiyon sisteminden  ıkarılmı tır. Kullanılan viraj denge  ubu u modeli karma ık bir yapıya sahip de ildir. Bu nedenle burulma sertli i hesaplanırken SAE (Society of Automotive Engineers) tarafından olu turulan hesaplama y ntemi kullanılmı tır.

S r   kalitesini de erlendirmek i in ISO 2631, BS 6841, absorbe edilen g   ve VDI 2057 gibi standartlar ve y ntemler bulunmaktadır. Bu  alı ma kapsamında s r   kalitesini de erlendirmek i in absorbe edilen g   y ntemi, frekans a ırlıklı RMS ve titre im doz de eri (VDV) kullanılmı tır. Absorbe edilen g  , rahatsızlık seviyesini    mek i in insan v cudunun ne kadar g   emdi ini temsil etmek  zere Watt cinsinden ifade edilir. Matematiksel olarak bir sayıyı g  steren bu y ntemde y ksek de erler rahatsızlı ın arttı ını g  sterirken sınır de er 6 Watt olarak belirlenmi tir. Konfor seviyesinin hesaplanmasında s r  c  veya yolcu konumlarından veri alınması daha do ru olmaktadır. Bu nedenle hesaplamalar i in s r  c  konumundaki ivme verileri alınmı tır. İvme de erlerinin  ıkarılması i in  ncelikle  oklu cisimler dinami i programında analizler yapılmalıdır. Daha sonra buradan elde edilen ivme de erleri ile absorbe edilen g   de eri TARADCOM tarafından belirtilen transfer fonksiyonları ile d  n    r lebilir. Bu d  n    m i in grafik programlama ve n merik hesaplama programı e  zamanlı olarak kullanılmı  ve sa lanan transfer fonksiyonları analiz edilmi tir. Her    global ekseninde absorbe edilen g   de erleri elde edilmi , sonu lar tablo ve grafiklere aktarılmı tır. Bir di er de erlendirme kriteri ise ISO 2631'dir. Frekans a ırlıklı RMS ve titre im doz de eri (VDV) sonu ları incelenerek s r   kalitesi hakkında yorumlara yer verilmi tir. Bu de erleri belirlemek i in sinyal i leme uygulaması ile her    eksenindeki ivme verileri incelenmi tir.

S r   kalitesi analiz edilirken de erlendirmenin do ru yapılabilmesi i in ara  her bir analizde 300 metre yol kat etmektedir. Ayrıca, farklı hız ko ullarını g  zlemlemek i in 12.87 km/sa ile 32.19 km/sa arasındaki hız de erleri incelenmi tir. Farklı faz ko ullarını incelemek i in 0, 45, 90 ve 180 derecelik faz a ılarındaki yollar  zerinde analizler yapılmı  ve etkileri incelenmi tir. Farklı burulma sertli i de erlerine sahip viraj denge  ubuklarının s r   kalitesi  zerindeki etkisi g  zlenlenmi tir.

Elde edilen sonu lara g  re, viraj denge  ubu unun sistem  zerindeki etkisinin d     olması nedeniyle kalınlıktaki de i imin 0 derecelik faz a ısında s r   kalitesi  zerinde

çok fazla etkisi olmamıştır. Ayrıca aracın bu faz açısında daha dengeli kalmasını sağlamaktadır. Faz açısının artmasıyla birlikte yanal eksenin absorbe edilen güç değerlerinde önemli artışlar gözlemlenmiştir. Özellikle 180 derecelik faz açısında bu farkın daha da açıldığı tespit edilmiştir. Viraj denge çubuğunun varlığı yanal eksenindeki sürücü rahatsızlığını önemli ölçüde artırmaktadır. Ayrıca en yüksek absorbe edilen güç değerlerinin dikey ekseninde olduğu tespit edilmiştir. Viraj denge çubuğunun 180 derecelik faz açısında dikey ekseninde olumsuz bir etkiye sahip olduğu gözlemlenmiştir. Sonuç olarak, viraj denge çubuğunun özellikle faz açısının artmasıyla birlikte sürüş kalitesini olumsuz etkilediği çıkarımına varılmıştır. Sonuçlara göre, en düşük absorbe edilen güç değerleri aracın boylamasına ekseninde (x eksen) elde edilmiştir. Frekans ağırlıklı RMS ve titreşim doz değeri (VDV) değerleri de incelendiğinde absorbe edilen güç metodu ile benzer sonuçların olduğu tespit edilmiştir.

Viraj alma ve çift şerit değiştirme gibi araç dinamiği analizlerinde, ön süspansiyonunda viraj denge çubuğu bulunan araçların daha düşük yuvarlanma açısı değerlerine sahip olduğu gözlemlenmiştir. Ayrıca ISO 3888-2 standardı ile viraj denge çubuğu olan ve olmayan araçlar şerit değiştirme analizine tabi tutulmuştur. Bu analiz bir engelden kaçınma parkurundan oluşmakta ve çift şerit değişimine dayanmaktadır. ISO 3888-2 sonuçlarına göre viraj denge çubuğuna sahip aracın parkuru daha yüksek hızlarda tamamlayabildiği görülmüştür.

Belirlenen algoritmaya göre çalışan aktif viraj denge çubuğuna sahip modelin sonuçları incelendiğinde araçların sürüş kalitesi değerlerinin özellikle yüksek faz açılarında oldukça iyileştiği gözlemlenmiştir. Yol tutuşun değerlendirildiği analizlerde ise aktif viraj denge çubuğu sistemine sahip araçların standart araçla benzer sonuçlar ürettiği ve viraj denge çubuğu olmayan araçlardan daha iyi performans sergilediği gözlemlenmiştir. Sonuç olarak, aktif sisteme sahip aracın sürüş kalitesi daha iyi olurken, yol tutuş özelliklerinde herhangi bir değişim yaşanmamıştır.



1. INTRODUCTION

Given the complexity of the subject matter, the study of vehicle dynamics plays a critical role in finding the right balance between driver feel, passenger comfort and safety, and fun. To improve vehicle dynamics, active suspension system studies become popular these days. The usage areas of active suspension systems vary from sports vehicles to off-road vehicles. These technologies improve ride comfort, vehicle stability and handling characteristics, making the ride more enjoyable and safer.

A vehicle's suspension system provides a smooth ride by isolating the vehicle from road disturbances. An active suspension system is a type of suspension that adjusts its settings in response to road conditions and driver inputs. The system uses various accelerometers and position sensors to monitor the vehicle's motion and adjust the suspension settings accordingly.

Active suspension systems can be classified into two categories: hydraulic and electromagnetic. Hydraulic systems use hydraulic actuators to adjust the suspension settings, while electromagnetic systems use electric actuators. Both types of systems have their advantages and disadvantages [1]. Semi-active suspension systems are mainly used in automotive applications, while active suspensions are rare.

An anti-roll bar, also called a stabilizer bar or anti-sway bar, is a component of a vehicle's suspension system that reduces body roll during cornering. Active anti-roll bars are a type of suspension system that adjusts the stiffness of the anti-roll bar depending on road conditions and driver input.

The active anti-roll bar uses sensors to measure the movement of the vehicle, and the control unit adjusts the stiffness of the anti-roll bar accordingly. The system can increase stiffness when cornering to reduce vehicle roll, improving vehicle stability and handling. When driving straight, the system can reduce stiffness for a more comfortable ride.

Hydraulic active suspension and anti-roll bar systems offer excellent performance, high accuracy, and fast response times. However, they are heavy, complex, and

expensive. Electromagnetic active suspension and anti-roll bar systems are lightweight, compact, and cost-effective. Still, they offer lower performance and slower response times than hydraulic systems [1].

Ride quality is important in a military vehicle for several reasons. First and foremost, a comfortable ride can help maintain the physical and mental health of vehicle occupants. Defense personnel may have to work in difficult and demanding environments, and a smooth and comfortable ride can help reduce fatigue and stress.

In addition, ride comfort is critical to the safety and effectiveness of defense operations. Vehicles with poor ride quality can lead to discomfort, distraction, and reduced situational awareness, affecting the driver's ability to perform his or her duties. Any distraction or discomfort can be life-threatening in combat situations where split-second decisions must be made.

In addition to the above factors, ride comfort can also affect vehicle performance and reliability. Poor ride quality can lead to mechanical stress and wear on vehicle components, increasing maintenance and repair costs. A comfortable ride can reduce stress on the vehicle's suspension, resulting in longer life and higher reliability.

Additionally, a comfortable ride can help maintain the readiness of defense personnel by reducing downtime due to injuries and medical problems caused by excessive vibration, shock, and rough terrain. A vehicle with excellent ride quality can help defense personnel operate efficiently and effectively for extended periods of time.

Gosselin-Brisson et al. (2007) compared the active anti-roll bar system using a hydraulic or pneumatic actuator with and without an anti-roll bar. It is observed that passive anti-roll bar boosted stability but lowered comfort. It is also mentioned a low-pass filter should be installed between the controller and the actuator to improve the performance of the active anti-roll bar system. Employing a high pass filter or including a control algorithm to prevent suspension bottoming is advised. [2]

Cronje and Els (2010) aimed to improve the vehicle's handling without significantly impacting comfort in its validated models with a double lane change. According to the steady-state results, a significant decrease in the rollover angle was obtained up to 0.4g lateral acceleration value in the active anti-roll bar system with an actuator in the middle. If it is above this value, it is aimed to show the driver that the road holding

limits have been reached for warning purposes. They also showed that the active anti-roll bar system on the Belgian pave road did not significantly affect comfort. [3]

Khalil (2018) intended to improve vehicle's rollover performance using fuzzy PID controller with active anti-roll bar system. To offset the weight shift on each axle, he employed DC motors in active anti-roll bars for the highest torque. Fishhook maneuver, double-lane change maneuver, and slowly increasing steer test were applied to the vehicle with three different setups: vehicle with and without anti-roll bar and active anti-roll bar. The fishhook maneuver, double-lane change maneuver, and slowly increasing steer test were all applied to the vehicle. With an active anti-roll bar system, it has been seen that the roll angle is decreased by over 95%, and the fuzzy PID controller almost completely prevents body weight transfer during maneuvers. [4]

Vu and Gaspar (2019) used an integrated model (four Electronic Servo-Valve Hydraulic (ESVH) actuators in a linear single unit heavy vehicle yaw-roll model) and the H_∞ controller architecture for the active anti-roll bar system. They noticed that the H_∞ active anti-roll bar control mechanism of the vehicle significantly increased roll stability. [5]

Hwang (2020) has applied active variable stiffness anti-roll bar to improve vehicle handling for a medium-sized electric bus. Multi-body dynamic studies of flexible bodies were discovered. Since the response of the chassis system was within 10 Hz, the noise was removed using a low pass filter set to 10 Hz. The technique is intended to strengthen the front rolling stiffness, decrease load shifting to the front outer wheel, and improve the grip of the front inner tires when significant oversteer occurs. The approach is to raise the rear rolling stiffness, decrease load shifting to the rear outer wheel, increase the traction of the rear inner tires, and increase the rear slip angle when the vehicle understeers excessively. The vehicle may respond to steering input with greater direction control and a quicker stabilization time, notably avoiding the risky unstable situation. [6]

In the active anti-roll bar studies carried out so far, anti-roll bar properties have been tried to be controlled using an actuator or motor. However, there are insufficient analyses in which the anti-roll bar is deactivated and its effect removed from the vehicle. In addition, the studies were generally carried out on specific paths, and the effect of phase angles was not examined. This study examines the effect of the anti-

roll bar at different phase angles and aims to improve the ride quality without observing a negative effect on handling.

This study investigated different road conditions, different anti-roll bar setups, and comfort conditions at different speeds of the FED-Alpha (Fuel Efficiency Demonstrator) military vehicle. In addition, the vehicle's handling levels were determined with different anti-roll bar thicknesses. The aim is to improve the ride quality by reducing the effect on the vehicle in cases where the comfort level deteriorates due to the anti-roll bar. Under these conditions, the vehicle is intended to provide better handling than the model without an anti-roll bar.

Analyses were performed with the FED-Alpha vehicle developed as a fuel efficiency demonstrator by Ricardo and validated by MSC Software. Vehicle dynamics analyses were carried out with the ADAMS Car program. At this stage, primarily anti-roll bar configurations that have a negative impact on driving comfort have been identified. In addition, the road-holding performance of the vehicle was examined according to the anti-roll bar thicknesses.

According to the results obtained, it was determined in which situations the anti-roll bar element should work. In addition, an algorithm that will provide both situations has been established, and analyses have been carried out.

2. THEORETICAL AND CONCEPTUAL BACKGROUND

2.1 Active and Semi-active Suspension Systems

2.1.1 Purpose of usage and advantages

Modern automobiles now include semi-active suspension systems, which enhance the convenience, safety, and control. Active and semi-active suspension systems use advanced sensors, computers, and actuators to change suspension settings in real time for a smoother, more steady ride than traditional suspension systems, which depend on inactive springs and dampers to buffer road impacts.

Electronic sensors, hydraulic or pneumatic actuators, and complex control methods are used in active suspension systems to rapidly and accurately change the rigidity, damping, and ride height of suspension components. As a result, the car can more effectively absorb bumps, keep equilibrium and traction when stopping and turning, and give occupants a smoother, more pleasant journey.

It is impossible to exaggerate the value of active suspension systems in contemporary cars. They provide a number of advantages, such as:

Increased comfort: Vehicle height and stiffness can be adjusted by active suspension systems to give occupants a softer, more pleasant journey.

Better ride: Active suspension systems can change the suspension parameters to improve the control and stability of the vehicle, particularly during sudden moves and high-speed turns.

Enhanced safety: By adjusting the suspension to account for road imperfections and abrupt shifts in orientation, active suspension systems can help avoid accidents.

Reduced wear: By ensuring a more constant ride height and lessening the effect of collisions on the car, active suspension systems can reduce wear on tires, brakes, and other components.

2.1.2 Types of active and semi-active suspension systems

A normal vehicle's suspension system is made up of shock absorbers and springs, which work in concert to safeguard the car's passengers from any unexpected wheel movement while it is moving. To ensure passenger convenience and protection as well as the long-term performance of the car's mechanical and electrical components, a suspension system needs to be properly adjusted.

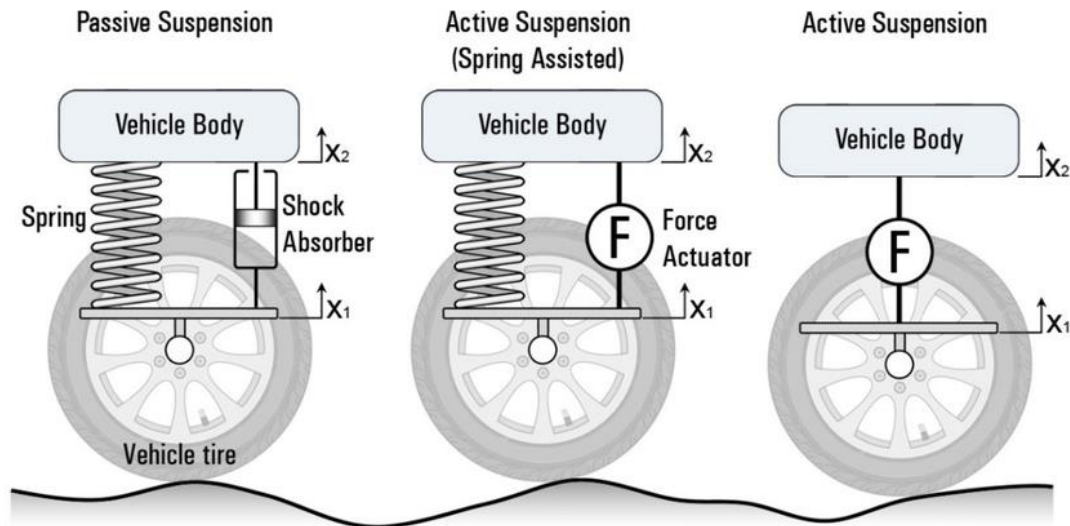


Figure 2.1 : Active suspension system concept [7].

Active suspension systems are designed to detect forces acting on the wheels and modify the links between the chassis and the wheels in real time to maintain a level chassis and absorb the energy of vertical wheel motion. In addition, advances in computer control have allowed the driver to set different suspension travel and response options while driving, including a "sport" mode that provides a more dynamic ride.

In modern times, semi-active suspension systems are utilized to address the limitations of passive suspension systems in terms of ride performance and reduce the high costs and energy consumption associated with fully active suspension systems. The hardware and control equipment necessary for an active suspension system is also required for a semi-active system. However, the actuator, hydraulic pumps, and accumulators are replaced with a controllable damper. Semi-active systems can provide more affordable suspension control solutions than active ones. The critical elements of a semi-active suspension system include the semi-active damper, acceleration sensors, ride height sensors, and controller [8].

2.1.2.1 Hydraulic active suspension system

This system uses hydraulic fluid to drive the suspension, which is controlled by a computer. The hydraulic system can adjust suspension settings in real time for a smoother, more stable ride.

Active hydraulic suspension systems use a hydraulic fluid to transfer energy between the various system components. The system consists of a pump, reservoir and hydraulic cylinders connected to the suspension components. [9]

When a bump occurs on the road, the suspension cylinder is compressed, forcing hydraulic fluid from the cylinder into the accumulator. The accumulator acts as a reservoir for the hydraulic fluid, allowing it to be stored under pressure for later use. When the vehicle encounters an incline, the hydraulic fluid is released from the accumulator, causing the suspension cylinder to extend and cushion the impact.

The hydraulic pump is responsible for pressurizing the fluid necessary to keep the active suspension system functioning. The system is controlled by a computer that can adjust the stiffness of the suspension, the damping and the ride height according to the driving conditions. [9]

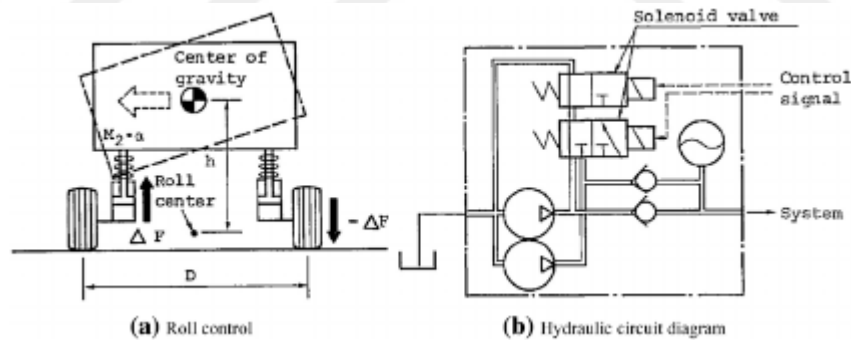


Figure 2.2 : Hydraulic active suspension system example [9].

An example hydraulic active suspension is shown in Figure 2.2. This is achieved by utilizing a power supply that delivers substantial energy, precise sensors, and highly responsive devices [9]. Active hydraulic suspension systems have the potential to offer a cost-effective and practical solution for improving ride comfort, handling control, and preventing rollover accidents. These systems can be particularly beneficial for heavy vehicles such as buses, trucks, SUVs, and similar vehicles. [10]

2.1.2.2 Electromechanical active suspension system

The electromechanical suspension system (EMS) for cars comprises a spring element and an electromechanical actuator, illustrated in Figure 2.3. The electromechanical actuator involves a DC motor and a ball screw mechanism, transforming the linear vibration motion into motor input torque.

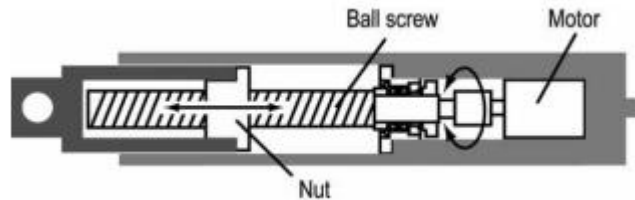


Figure 2.3 : Concept of electro-mechanical actuator [11].

When used passively, the electro-mechanical actuator utilizes electro-mechanical force as the damping force, which can be controlled by regulating the induced current. The proposed actuator has three crucial characteristics: high controllability achieved by tunable damping force, active suspension, and energy regeneration. Previous studies have employed the proposed actuator as passive suspension, exhibiting almost linear damping characteristics. However, this paper uses the electro-mechanical actuator as an active suspension, controlling it to produce output force calculated from sprung and unsprung masses' velocities. Unlike traditional shock absorbers that convert vibration energy to heat, the electro-mechanical suspension comprises an electric motor/generator and a spring that converts vibration energy into electric energy. The damping control by the active suspension enables energy-saving control, as the amount of absorbed energy depends on the suspension system's damping. The control that considers energy consumption with the electro-mechanical suspension can enhance vehicle maneuverability, which includes vehicle attitude and road-holding ability of tire balancing energy consumption [11].

2.1.2.3 Magneto-rheological semi-active suspension system

Semi-active suspension systems can be achieved through technical solutions such as controllable fluids, hydraulics, compressible fluids, and friction-based systems. The most promising technology among these solutions is the semi-active suspension systems that use magneto-rheological (MR) fluids. MR fluids contain suspended ferromagnetic particles and exhibit changes in rheological properties, including

viscosity when exposed to a magnetic field. This allows them to rapidly change from viscous liquids to semi-solids in milliseconds when a magnetic field is applied, making MR fluids attractive for many applications. This gives a rapid response interface between electronic controls and mechanical systems, making MR technology attractive for many applications, including dampers. MR fluids have been successfully installed in some upper-class cars [12].

Magneto-rheological (MR) active suspension systems use a special magneto-rheological fluid in the shock absorbers. This fluid contains tiny magnetic particles suspended in a base fluid. When a magnetic field is applied to the fluid, the particles align and form chains, making the fluid more viscous and resistant to flow. The result is a more rigid and reactive suspension. The magnetic field is controlled by an electromagnet that can be adjusted according to road conditions and vehicle dynamics, allowing the suspension system to be adjusted in real time. The result is a suspension that can adapt to changing road conditions, providing a smoother and more stable ride.

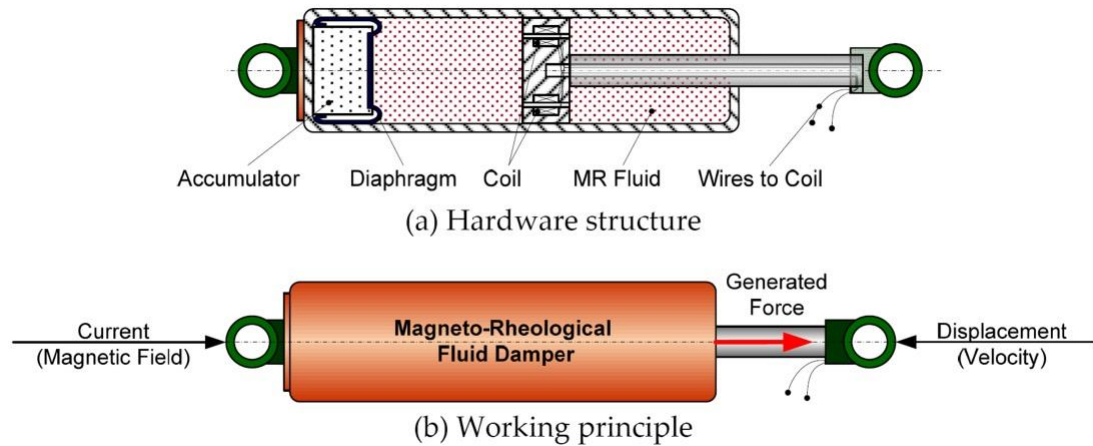


Figure 2.4 : Working principle of magneto-rheological damper [13].

The use of MR dampers can help in managing hysteresis. Moreover, it can also help address other intricate nonlinearities, like those that occur due to deadband and backlash, without linearization. The stability of the adaptive system is verified by transforming the intrinsic hysteresis of MR dampers into a nonlinearity without memory [14].

2.1.2.4 Electromagnetic active suspension system

Electromagnetic active suspension system (EMAS) is a type of active suspension system that uses electromagnets to provide the necessary damping and spring forces for the suspension system. The EMAS system typically includes sensors to measure the road surface and vehicle motion, a controller to process this information and determine the required electromagnetic force, and an actuator to generate the force using the electromagnets. The system is designed to provide a comfortable and stable ride by adjusting the suspension forces in real time according to the current road conditions, speed and maneuvering of the vehicle.

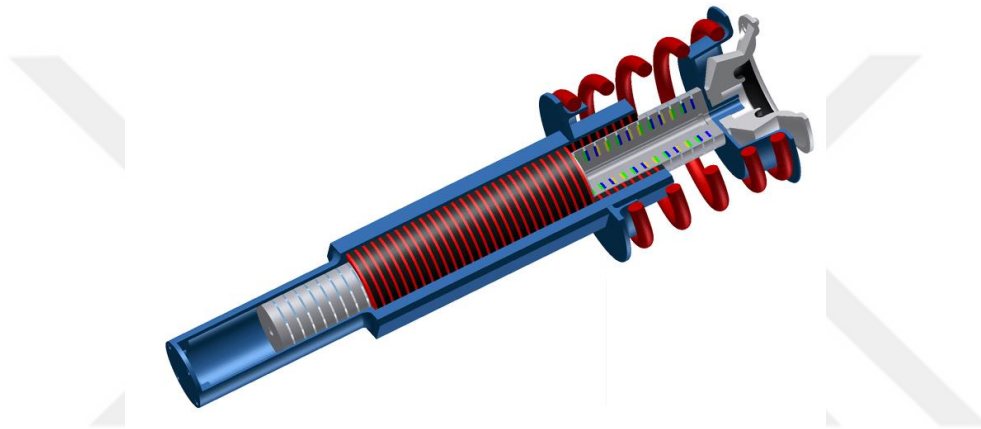


Figure 2.5 : Electromagnetic active suspension system example [15].

The electromagnetic active suspension (EMAS) system utilizes two types of actuators: the rotary motor and the linear motor. The rotary motor is paired with mechanical structures such as gear racks and ball screws to transform reciprocating movement into unidirectional rotation, but this results in larger installation dimensions and moment of inertia that is detrimental to light-weight design and energy recovery efficiency. Typically, rotary dampers have the ability to produce greater power and achieve a higher damping coefficient within a given space [16]. In contrast, the linear motor has a simpler structure and faster response in high frequency domains, making it a preferred actuator for active suspension systems. The linear motor is capable of accurately exporting electromagnetic force in power-driven mode, recovering energy through electromagnetic induction, and providing damping force based on motor characteristics in power-regenerated mode. Control algorithms are necessary to switch between these modes and design an appropriate control strategy, which is crucial for achieving the excellent performance of EMAS systems [17].

2.1.3 Applications of active and semi-active suspension systems

Active and semi-active suspension systems have a wide range of applications.

Automotive industry: Active and semi-active suspension systems are used in many different vehicle types. They enhance ride quality, handling, stability, and performance during rides. [18]

Military: To increase mobility, stability, and protection, military trucks use active and semi-active suspension systems. They lessen vibration in the car and enhance ride quality while helping to keep control and equilibrium on rocky ground.

Aerospace industry: Aviation landing gears use active and semi-active suspension devices to increase safety during takeoff and touchdown. They lessen vibration in the airplane, cushion touchdown shock, and enhance comfort.

Railroads: To increase travel comfort, stability, and safety, railroads use active and semi-active suspension systems. They lessen car vibrations, buffer the impacts of track irregularities, and enhance total ride quality.

Industrial machinery: Active and semi-active suspension systems are used in heavy machinery to improve stability, safety and ride comfort. They lessen driver fatigue, enhance handling, and reduce car vibrations.

Medical industry: Medical apparatus, including hospital beds, scooters, and surgical tables, uses active and semi-active suspension mechanisms. They lessen disturbance, enhance patient satisfaction, and guard against accidents.

Sports industry: Sports equipment, including bicycles, motorbikes, and snowmobiles, use active and semi-active suspension systems. They enhance steering, control, and steadiness while minimizing vibration and impact to the user.

2.2 Ride Quality

2.2.1 Content of ride quality

The ability of a vehicle to prevent damage to its passengers from bumps in the road surface is referred to as ride quality. A car provides a comfortable journey for the driver and occupants.

2.2.1.1 Significance

In addition to minimizing cargo damage and increasing passenger comfort, a smooth ride minimizes driver weariness during long trips. This is important because road disruptions can impair the driver's ability to manage the car [19].

Because vehicles with firm suspension can result in greater control of bodily motions and faster responses, suspension design is frequently a compromise between ride quality and vehicle handling [20]. Similarly, a lower center of gravity improves handling but restricts suspension movement and necessitates stiffer springs due to low ground clearance.

2.2.1.2 Influencing variables for ride quality

The state of the road pavement can significantly impact the ride characteristics. Bumpy, uneven, or potholed roadways can cause uncomfortable vibrations and jarring motions in the car.

Ride quality can be significantly influenced by the car's architecture, including the suspension system, tires, and seats. While supportive seats can help passengers feel less worn out and comfortable, a well-designed suspension system can help mitigate shocks and noises from the road [21].

The driver's driving style can impact the vehicle's travel quality. Passengers may feel uneasy due to sudden moves and changes in pace caused by erratic or aggressive driving.

As a result of reduced visibility and a more dangerous road surface, unfavorable weather conditions like strong winds, persistent rain or snow, and ice can affect the quality of the ride.

An unpleasant ride can be caused by excessive noise from the engine, tires, or other sources. These noises can also be irritating and distracting. Poor vehicle upkeep, including old tires, worn or broken suspension parts, or unbalanced wheels, can also harm travel quality [18].

Ride quality can be impacted by the number of passengers and the automobile's cargo quantity. A car can become unsteady from overloading, affecting the suspension system's capacity to dampen shocks and vibrations.

2.2.2 Evaluation methods

A variety of assessment techniques are employed to rate ride quality. Subjective and objective measures are the two general groups into which these techniques can be split.

2.2.2.1 Subjective evaluation methods

Occupant feedback about their encounters with the journey is the basis for subjective evaluation techniques for ride quality. These techniques seek to record the passengers' subjective perceptions and emotions, which can offer insightful information about the quality of the journey and point out areas for development.

Road maintenance plans are developed by agencies using subjective roughness evaluations from user groups for a very long time. But the commoditization of technology spurred the creation of more impartial, automatic, and economical measurement technologies.

Ratings of ride quality are used by transportation organizations to help create policies for maintaining the road network, from resurfacing to replacement. Prior to the development of concrete metrics like the international roughness index (IRI), organizations relied on subjective evaluations of trip quality provided by users who were thought to be experts.

Researchers have lately started looking into the feasibility of combining data from connected cars or cellphones to discover more affordable options. However, due to changes in instrument sensitivity, orientation, and positioning, such methods can result in significant measurement variations. By implementing standards and calibration techniques, future linked cars offer the potential to decrease such measurement variations. Agencies will still need to understand how those measurements pertain to roughness levels experienced by human users even when future connected cars produce objective roughness measurements that are standardized [22].

The subjective panel evaluation, which is known to differ from region to region and nation to country, is thus a unanchored scale that is affected by the observers' preconceptions of acceptable ride comfort [23]. Many experts have studied objective evaluation methods to avoid these differences.

2.2.2.2 Objective evaluation methods

Physical metrics that evaluate the ride's quality are considered objective evaluation methods. These methods offer quantifiable information that can be used to evaluate the experience and pinpoint areas in need of development.

In computer simulations, objective evaluation methods for trip quality usually make use of simulations and statistical models to rate the ride's quality. These methods can be used to assess the ride quality under various circumstances and situations and are frequently used in the construction and testing of vehicles.

Earlier vehicles are frequently the basis for the fundamental travel comfort design. For these, the objective suspension parameters (such as the characteristics of the suspension springs and dampers or the unsprung mass) are well known, and subjective evaluations can be used to determine whether the vehicle is acceptable before manufacturing begins. Given this context, the new generation vehicle is initially planned rather haphazardly. Then, different suspension parts with their corresponding parameters are tuned iteratively in consecutive cycles, with the shock absorber system, including the damper top mounts, receiving particular attention [24].

The objective ride quality evaluation methods were developed using various techniques developed by various experts. Limit values were set by contrasting the numerical results with the travelers' feedback. An example methodology is shown in Figure 2.6.

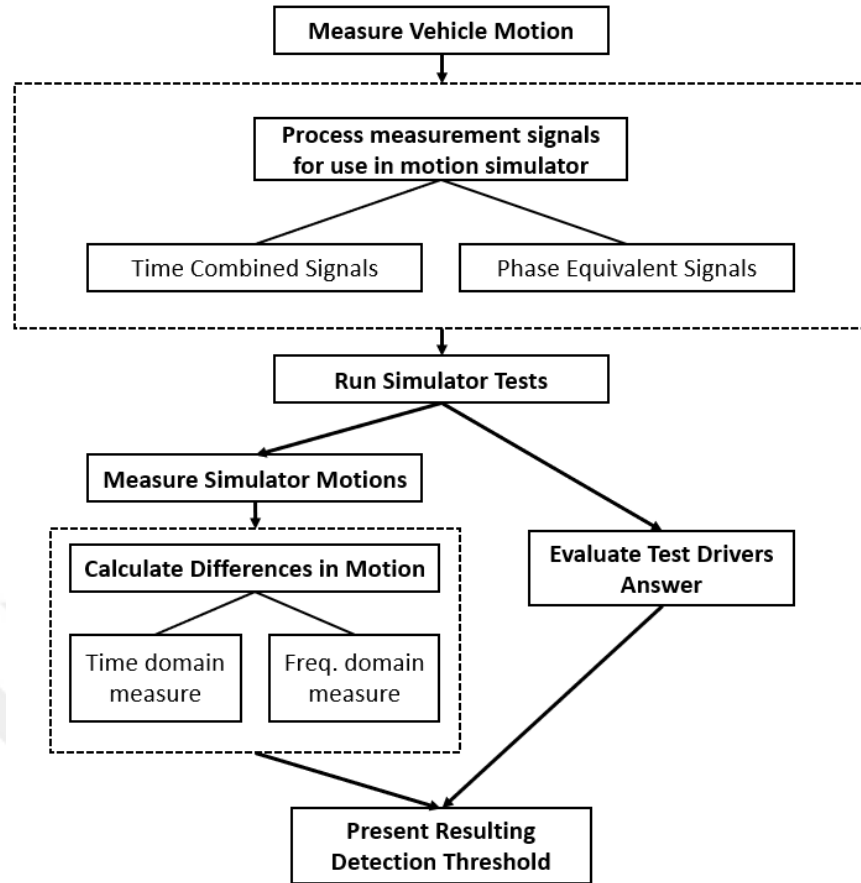


Figure 2.6 : An example methodology for measuring test drivers sensitivity to changes in vehicle response [25].

The definition of the vehicle comfort occurrence that will be studied should be the first step in this process. There is a lot of possibility in focusing on one or more elements as opposed to rating comfort, as the following chapter demonstrates. In the latter scenario, it would be much more challenging to determine what actions are required to increase comfort because each ride comfort criterion has unique relationships with the fundamental physical process, which is still not fully known. The change of the properties (component or subsystem particular) will become more target-oriented for maximizing ride satisfaction as our knowledge of these interrelations grows. However, since there might be minor variations in how those terms are described, it seems suitable to start by establishing a shared understanding for all parties involved in the process [24].

It is crucial to decide on the ranking and labeling criteria precisely and plainly for those performing the subjective evaluation. Some of these techniques include scoring between 0 and 10 or moving forward with designations like very excellent or average.

The continued creation of sophisticated driving models has made it possible to simulate actual road data in a lab setting. Thus, contemporary driving simulators provide a compromise for the disputes that occur for both field tests and test rigs between reproducibility and test validity, as well as between high-test variation and expense and time usage. However, it is essential to note the high acquisition prices of these systems as a drawback. Finding the appropriate objective measures that best represent the subjective expression is the primary component of the correlative method after all the data has been collected. The quality of the subjective data can be evaluated using a variety of descriptive or explorative statistics techniques.

On the presumption that there is a causal relationship between the driver's perception and an objective value, the goal of accurately describing the subjective impression of ride satisfaction is predicated.

The most well-known calculation method to determine ride quality has emerged with ISO 2631. "Frequency-weighted root mean square acceleration values," also known as "rms values," are a collection of weighted acceleration values that can be used by ISO 2631 to assess the pain brought on by whole-body vibration [26]. These numbers are determined by monitoring the acceleration at various frequencies and weighting the results based on how they are thought to affect people's wellbeing.

The standard establishes a collection of rms numbers based on the sensitivity of various bodily parts to vibration for various frequency bands. The "vibration dose value," or VDV, is then calculated by adding these numbers together using a mathematical algorithm.

2.2.3 Absorbed power

The Absorbed Power Method was created to provide a numerical standard for gauging pain. The resulting number is represented in the measure "Watt," which reflects the fundamental notion that the human body receives a certain quantity of electricity during a vehicle journey. Higher received power values correspond to more significant pain. The upper limit for an extended vehicle journey over uneven terrain is 6 Watts absorbed power for typical individuals [27].

A time-dependent acceleration indication from the base of the driver's seat is required to determine the amount of power that is received. The frequency-weighting of the acceleration impulses using unique filters is the following stage. For weighting of longitudinal (fore-aft), lateral (side-to-side), and vertical seat motion, various categories are available. The transfer functions for fore-aft, side-to-side, and vertical motion can be found in Equations 2.1, 2.2, and 2.3, respectively.

$$H(s) = \frac{230s}{(s + 15)(s^2 + 24s + 750)} \quad (2.1)$$

$$H(s) = \frac{525s(s + 9.47)(s + 8.8)}{(s + 125)(s^2 + 8.8s + 142)(s^2 + 3.39s + 14.2)} \quad (2.1)$$

$$H(s) = \frac{14.2s(s + 44)}{(s + 63)(s^2 + 24s + 750)} \quad (2.3)$$

$$s = j\omega = j2\pi f \quad (2.4)$$

f: frequency, Hz.

These formulae specify the electronic weighting circuits' frequency properties. To determine the absorbed power for each action, the result needs to be squared and summed.

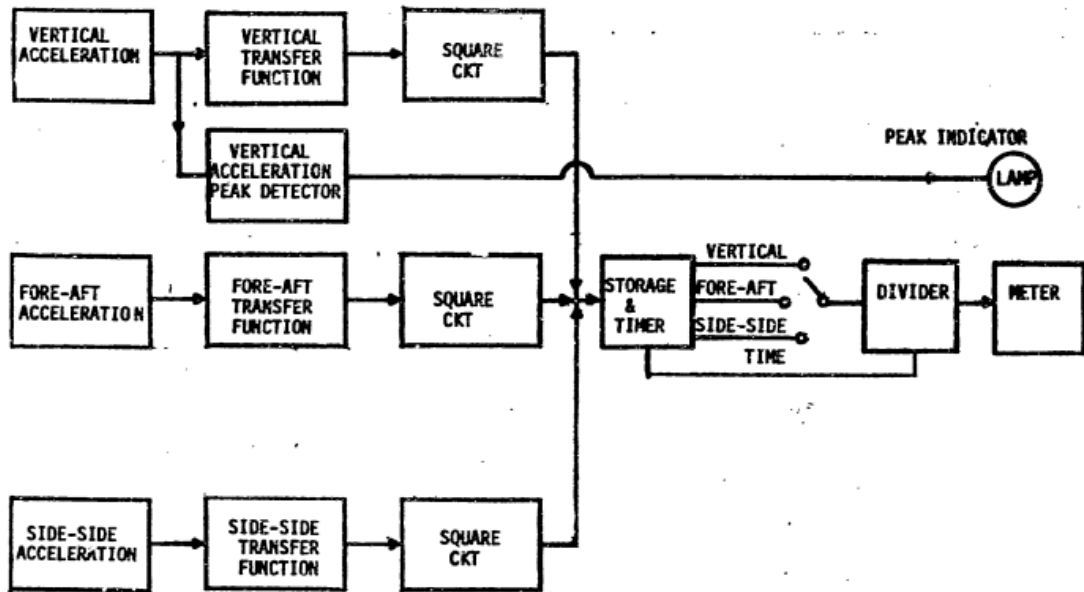


Figure 2.7 : Absorbed power computing process [28].

A block schematic of the interconnection of the different circuits is shown in Figure 2.7. A triaxial accelerometer is used to measure motion. The accelerations are fed through the frequency weighting circuits. Each action has a separate circuit for rounding and storing data [28].

2.2.4 Frequency-weighted rms

The rms weighted acceleration technique, outlined in ISO 2631-1, is another approach that determines the ride quality and the maximum amount of vibration a human body can endure.

A measure of whole-body vibration in ISO 2631, known as frequency-weighted root mean square (rms), considers the human body's sensitivity to various vibrational frequencies. The ISO 2631 standard offers recommendations for assessing and quantifying human exposure to whole-body vibration, which can happen in various circumstances, including operating heavy machinery or traveling in a vehicle. To take into consideration the fact that the human body reacts differently to vibrations at various frequencies, the standard suggests utilizing frequency weighting. The human body's sensitivity to vibration at various frequencies is the foundation for the frequency weighting employed in ISO 2631. The frequency weighting function is first applied to the vibration signal, and the root mean square of the resultant values is used to determine the frequency-weighted rms value. This gives a measurement of the energy content of the vibration that accounts for the body's sensitivity to vibration at various frequencies [26].

The weighted rms acceleration could be determined using Equations 2.5.

$$a_w = \left[\frac{1}{T} \int_0^T a_w^2(t) dt \right] \quad (2.5)$$

$a_w(t)$: the weighted acceleration (translational or rotational) as a function of time (time history), expressed in meters per second squared (m/s^2) or radians per second squared (rad/s^2), respectively

T: the measurement period in seconds.

The Equation 2.6 calculates the total vibration value of weighted rms acceleration, which is derived from vibration in orthogonal coordinates and incorporates vibrations in several directions.

$$a_v = \sqrt{k_x^2 a_{wx}^2 + k_y^2 a_{wy}^2 + k_z^2 a_{wz}^2} \quad (2.6)$$

a_{wx} , a_{wy} , a_{wz} : the weighted rms accelerations with respect to the orthogonal axes x, y, z, respectively

k_x , k_y , k_z : multiplying factors.

In this study, k_x , k_y , and k_z are all taken as 1.4, 1.4, and 1.0, respectively.

The criteria for how rms weighted acceleration values feel to people are shared in the ISO 2361-1 standard. The definition of "extremely uncomfortable" is greater than 2 m/s², whereas "not uncomfortable" is less than 0.315 m/s². In Table 2.1, a distinct range is designated as a pleasant level between "extremely uncomfortable" and not uncomfortable [26].

Table 2.1 : ISO 2631-1 discomfort evaluation levels.

a_v (m/s ²) Range	Discomfort Level
Less than 0.315 m/s ²	not uncomfortable
0.315 m/s ² to 0.63 m/s ²	a little uncomfortable
0.5 m/s ² to 1 m/s ²	fairly uncomfortable
0.8 m/s ² to 1.6 m/s ²	uncomfortable
1.25 m/s ² to 2.5 m/s ²	very uncomfortable
Greater than 2 m/s ²	extremely uncomfortable

2.2.5 Vibration dose value

A whole-body vibration measurement called the vibration dose value (VDV) accounts for both the time and the intensity of vibration exposure. It is a single value that sums up the total amount of vibration exposure a person has endured.

The root mean square (rms) acceleration of the vibration signal is first determined, and then a weighting factor that considers the strength and duration of the vibration is applied. A measure of cumulative exposure is then produced by integrating the resultant number through time. High vibration dose values will result in excruciating

pain, discomfort, and damage and, generally speaking, reflect the intensity of the vibration exposures that led to them [29].

The vibration dose value is calculated using Equation 2.7.

$$VDV = \left\{ \int_0^T [a_w(t)]^4 dt \right\}^{1/4} \quad (2.7)$$

The terms "exposure action" and "limit value" describe the daily vibration exposure to whole-body vibration for any worker that should be reduced or not exceeded. Table 2.2 displays the exposure action and daily VDV limit values [29].

Table 2.2 : Exposure action and limit values of daily VDV ($\text{m/s}^{1.75}$).

Level of daily vibration exposure	European Union Directive 2002/44/EC	ISO 2631-1	Probability of adverse health effect
Exposure action value	9.1	9.1	Moderate
Exposure limit value	21	14.8	High

2.2.6 Components affecting ride quality

Several vehicle parts can significantly affect the ride quality and comfortable driving. The suspension system, tires, wheels, steering system, brakes, and chassis all play a vital role in ensuring a seamless and pleasant journey. To deliver a better driving experience, every component—from impact absorption to traction and stability—must be improved. In this way, knowing how each of these parts works can assist drivers in choosing a vehicle or enhancing the ride quality of their current vehicle.

The suspension system supports the vehicle's weight and is in charge of buffering road impacts and vibrations. Shock absorbers, springs, struts, control arms, and anti-roll bars are part of the suspension system [30].

By giving the vehicle traction, cushioning shocks, and lowering noise and disturbance, tires play a crucial part in ride quality. The tread pattern, tire pressure, and tire variety are some variables that can impact tire effectiveness. (e.g., all-season, winter, summer).

By transferring shocks and movements from the road to the suspension system, the wheels can also impact ride quality. Size, material, and shape are some elements that can influence how well a tire performs.

By giving the driver input and assisting in maintaining the vehicle's stability on the road, the steering system can also impact ride quality [31]. The steering wheel, steering column, steering rack, and steering actuator are all parts of the steering mechanism.

The brakes are essential for halting the vehicle but can compromise the journey. The brake cushion type, the brake rotor size, and the brake caliper design are all variables that can impact brake effectiveness.

The vehicle's structure, or chassis, provides structural support and aids in vibration and impact absorption from the road [32]. Materials used in building and design elements like reinforced regions can impact how well a chassis performs.

2.3 Passive Anti-roll Bar

An anti-roll bar, a sway bar, a stabilizer bar, or an anti-sway bar is part of a vehicle's suspension that helps lessen body roll when the vehicle turns or corners.

Body roll is the side-to-side or lateral movement of a vehicle's body when it is moving, especially at high speeds or during severe bends. This motion may force the vehicle to tilt to one side, impairing its stability, traction, and handling.

A metal bar that is installed across the vehicle's chassis, often close to the front and rear axles, is an anti-roll bar. The bar is attached to the suspension on each side of the car through links or bushings.

When a vehicle turns or corners, the vehicle's weight exerts a force that may cause the suspension on one side of the vehicle to compress while the suspension on the other side stretches. This may result in the vehicle leaning to one side, affecting its stability and traction.

The stabilizer bar distributes compression force applied to one wheel's upward movement to the other, causing that wheel's suspension to compress. This enhances the vehicle's handling, traction, and overall stability by keeping it level and stable during cornering. Anti-roll bars, which connect the left and right wheels, can reduce

the vehicle's roll angles at strong lateral accelerations and manage the weight distribution between the tires during cornering for road-holding purposes [33].

Anti-roll bars are frequently used in various vehicles, such as trucks, sports cars, and passenger cars, and they are frequently combined with other suspension parts to create a balanced and efficient suspension system.

2.3.1 Working principle

An anti-roll bar's operation is based on minimizing body roll during cornering or turning.

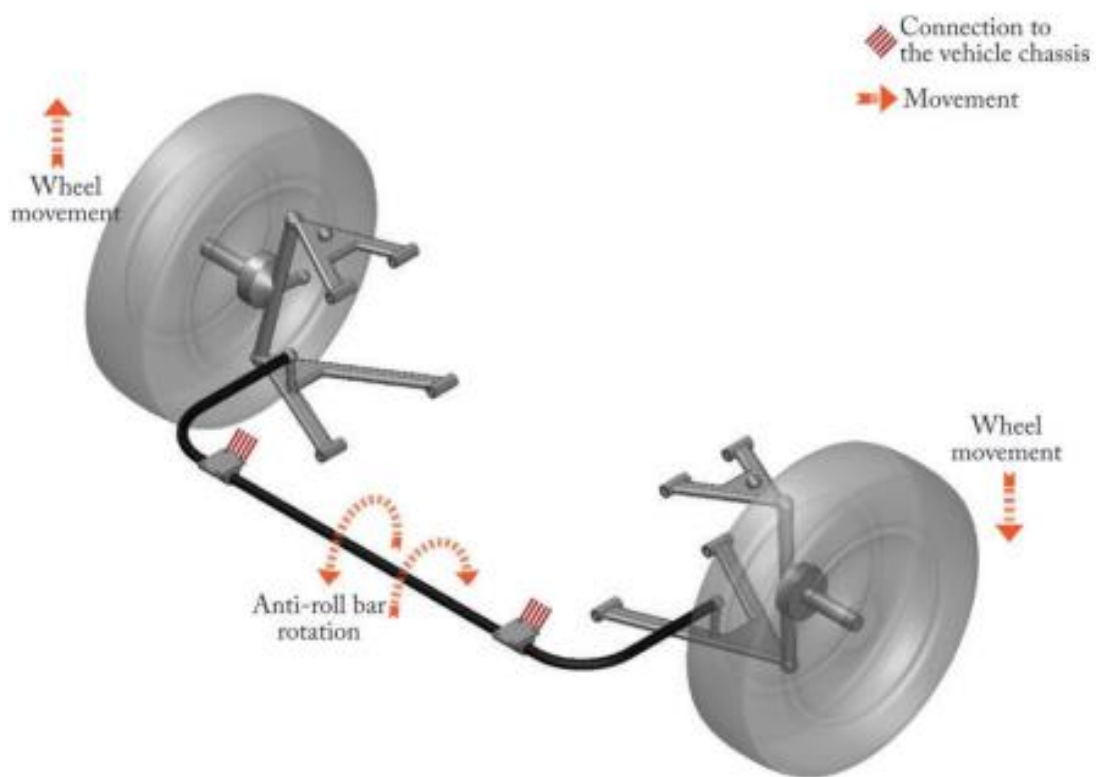


Figure 2.8 : An example anti-roll bar installation [34].

When a vehicle turns, its weight transfer force may cause the suspension on one side of the vehicle to compress while the suspension on the other side expands. As a result, the car leans to one side, which might affect its stability and traction.

By transferring part of the force produced by the weight shift from the compressed suspension to the stretched suspension, an anti-roll bar aids in preventing this body roll. The anti-roll bar accomplishes this by linking the left and right suspension systems via a metal bar across the car's chassis, often close to the front and rear axles.

The anti-roll bar twists, and the suspension on one side of the car contracts as it rotates. The anti-roll bar's twisting produces a force that inhibits body roll and shifts some weight from the compressed suspension to the stretched suspension on the other side of the car. This enhances the car's handling, traction, and overall stability by keeping it level and stable during cornering [34].

2.3.2 Effects on vehicle

Depending on the precise design and application of the bar, anti-roll bars can positively or negatively impact the vehicle's dynamics and ride quality.

Anti-roll bars can enhance a vehicle's handling and stability when turning or cornering by decreasing body roll. Particularly while traveling at high speed or when making forceful maneuvers, this can assist in lowering the chance of rollover or loss of control. Statistics show that installing an anti-roll bar reduces the likelihood of a rollover by between 60% and 80% [35].

When turning or cornering, anti-roll bars can assist in maintaining better tire-road contact, increasing traction, and lowering the danger of sliding or skidding [35].

Anti-roll bars are designed to lessen swaying during corner turns. As a result of how they affect steering and keep the wheel firmly planted on the ground, they can also improve vehicle handling and stability.

Anti-roll bars may make the suspension system more rigid, giving the ride a harsher feel. This is especially obvious on roads with uneven or rough surfaces.

Anti-roll bars can worsen the vehicle's noise, vibration, and harshness (NVH) levels, particularly at higher speeds or during forceful maneuvers. Increased vibration or noise coming from the chassis or suspension parts may be the cause of this. Research has found that even the mounting position of the anti-roll bar reduces vibration and indoor noise [36].

2.3.3 Calculation of torsional stiffness of the anti-roll bar

Anti-roll bars are torsion springs mounted laterally against the vertical displacement of the wheels relative to each other. Attachments to the frame and suspension components are usually rubber-insulated [37].

There are different types of anti-roll bars. These can be classified as solid anti-roll bars, tubular anti-roll bars, splined anti-roll bars, and active anti-roll bars. It is known that the changing profile of the anti-roll bar section changes the stiffness by increasing the momentum of inertia without the need to add weight [38]. This study studies a solid anti-roll bar, and calculations are carried out using SAE roll rate calculation for anti-roll bars.

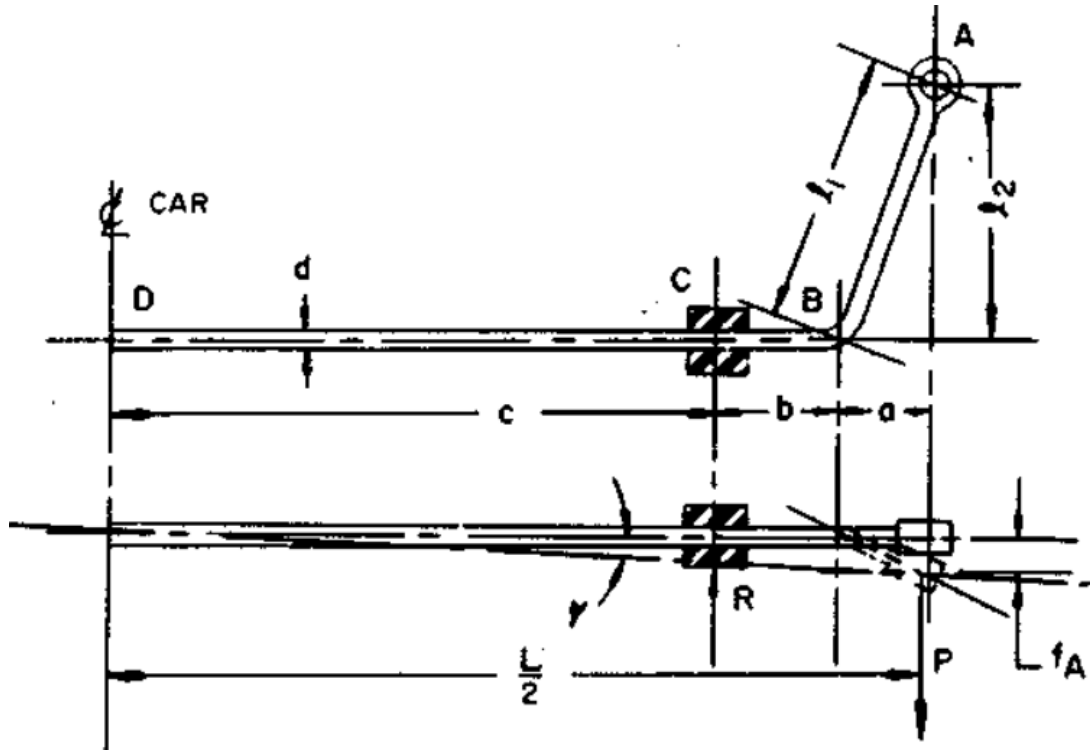


Figure 2.9 : Anti-roll bar parameters for calculation [37].

A and C represent the point of load application and support cushion, respectively. The required rod rolling speed can be obtained by multiplying the desired effective rolling speed at the wheels by the square of the ratio of wheel movement to rod movement at the rod end and the square of the rod length ratio [37].

$$k_R = k_{RW} \times \left(\frac{f_W}{f_A} \right)^2 \times \left(\frac{L}{T} \right)^2 \quad (2.8)$$

k_R : Bar roll rate, Nmm/rad

k_{RW} : Effective roll rate at wheels, Nmm/rad

f_W : Wheel travel

f_A : Deflection at bar end "A"

L: Bar length

T: Wheel tread.

The deflection "A" of the rod ends can be determined by the unit load method:

$$f_A = \int \frac{M \cdot m}{E \cdot I} ds + \int \frac{T \cdot t}{G \cdot J} ds \quad (2.9)$$

M: Bending moment

T: Torsional moment

m: Bending moment equations due to a force of one Newton acting at the bar end "A"

t: Torsional moment equations due to a force of one Newton acting at the bar end "A"

E: Elastic modulus

I: Moment of inertia

G: Modulus of rigidity

J: Polar moment inertia.

The deflection of "A":

$$f_A = \frac{P}{3EI} \left[l_1^3 - a^3 + \frac{L}{2} (a + b)^2 + 4l_2^2(b + c) \right] \quad (2.10)$$

Bar roll rate:

$$k_R = \frac{PL^2}{2f_A} \quad (2.11)$$

$$k_R = \frac{3EIL^2}{2 \left[l_1^3 - a^3 + \frac{L}{2} (a + b)^2 + 4l_2^2(b + c) \right]} \quad (2.12)$$

P: Applied load at bar end, N

E: Elastic modulus, MPa

I: Moment of inertia, mm⁴

k_R: Bar roll rate, Nmm/rad.

As a result of Equation 2.12 obtained, comments can be made about the parameters that can be changed to obtain a soft or stiff anti-roll bar. As seen from this equation, it is possible to change the stiffness of the anti-roll bar without changing the attachment or other important positions.

In this study, anti-roll bar setups with different thicknesses/radii were used to change the stiffness value of the anti-roll bar. Calculations were carried out based on Equation 2.12.

2.4 Active Anti-roll Bar

A vehicle's active anti-roll bar is a part of the suspension system that helps to lessen body roll when turning or doing other lateral actions. Shifting part of the lateral pressures from the outside to the inside wheels, which helps keep the vehicle more stable and balanced, is intended to enhance the vehicle's handling and stability.

An active anti-roll bar is a dynamic system that may dynamically modify the bar's stiffness depending on the driving circumstances, in contrast to a standard passive anti-roll bar, which is a static component that joins the left and right sides of the suspension together [2]. This is commonly done by using hydraulic or electrical actuators, which may exert changing forces on the bar based on the vehicle's speed, steering angle, and other characteristics.

An active system can improve handling and stability when the vehicle is pushed to its limits during high-speed cornering or other maneuvers while delivering a smoother and more comfortable ride under usual driving conditions. Traditionally, this is done by dynamically adjusting the stiffness of the anti-roll bar.

3. MODEL AND SIMULATIONS

3.1 FED-Alpha Vehicle Model

The simulations feature the FED-Alpha armored personnel carrier developed as a fuel efficiency demonstrator. The FED-alpha vehicle combines improved aerodynamics, lightweight materials, and an innovative hybrid engine to achieve performance and efficiency. It is intended to be a lightweight, high-performance off-road vehicle. A 28-volt generator that can provide 20 kW of electrical power for communications and other equipment and low-friction differentials and wheel hubs are also added to the vehicle. Additionally, an indicator that purports to provide driving tips is included. [39]



Figure 3.1 : Photo of FED-Alpha vehicle.

Table 3.1 : Properties of FED-Alpha.

Parameter	Unit	Value
Vehicle mass	kg	5157
Sprung mass	kg	4093
Unsprung mass	kg	1064
Height	mm	2088
Length	mm	5156
Width	mm	2288
Frontal Area	m ²	3.8
Weight distribution	%	46.27
Front corner load	N	14115
Rear corner load	N	16389
Wheelbase	mm	3302
Front axle track width	mm	1953
Rear axle track width	mm	1959
Ground clearance	mm	337.8
COG position x	mm	2831
COG position y	mm	5.1
COG position z	mm	966.5
Approach angle	°	54
Departure angle	°	57
Tire	-	335 / 65 R 22.5
Tire pressure	psi	35 / 60
Engine	-	Cummins 4.5 L Diesel
Transmission	-	6-Speed Automatic
Final drive gear ratio	-	4.88
Max. speed	km/h	132

In Table 3.1, there are two different tire pressure values. The reason is that tires with 35 psi pressure are used in ride quality analyses, while 60 psi pressure tires are used in handling analyses.

The FED-Alpha has several fuel-saving features, including:

- Low rolling resistance tires created especially for the FED
- Optimized I4 engine expressly for tactical vehicle fuel economy
- Lightweight aluminum framework, armor, and underbody blast shield that also helps to lighten the vehicle
- An enhanced driveline using non-geared hubs
- Isotropic gears with a super-finished surface to decrease friction and boost fuel economy
- Drivers are instructed on using the vehicle to generate the optimum fuel efficiency via the accelerator feedback pedal and the fuel economy display.

Titanium coil springs with air springs are utilized in the suspension system, which is important to determine spring characteristics, along with dampers with varied responses at low and high frequencies. The right and left wheels can move thanks independently to the double-wishbone axle design. The suspension also permits 180 mm of rebound and jounce movement. The air springs and the ride height sensor, built into the axles, maintain the vehicle's height despite shifting weight distributions and load combinations. According to the suspension speed, the air springs' stiffness value and the vehicle's spring characteristics alter. Only titanium springs determine the vehicle's spring properties in an empty state. The stiffness value of the suspension has a stiffer curve as the load on the suspension increases, as does the air pressure within the air springs.

The damper from FED-Alpha features two distinct dampening characteristics: a high frequency and a low frequency. The damper characteristics are chosen based on the frequency that is being impacted. Lower damping force impacts the high-frequency loads depending on the vehicle's speed. On the other hand, the low-frequency curve climbs with a more significant trend as the vehicle's speed rises. In this instance, the vehicle's motion is dampened more in the low-frequency region.

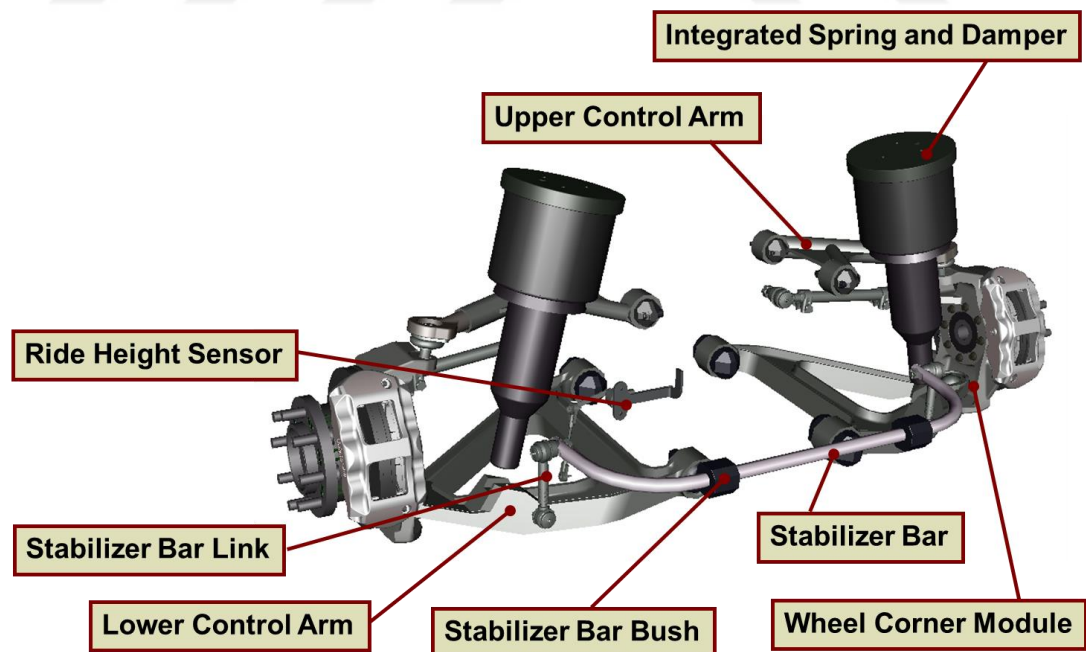


Figure 3.2 : CAD model of FED-Alpha vehicle suspension system.

The suspension system includes an anti-roll bar to lessen the rolling movement of the vehicle. Only the front axles have anti-roll bars; the rear axles do not. It, therefore, impacts handling, particularly on roads where two wheels move substantially.

3.2 Anti-roll Bar Model

The FED-Alpha vehicle only has a front anti-roll bar and has a solid type anti-roll bar. The presence of an anti-roll bar in the front directly affects parameters such as body roll, handling balance, and ride quality.

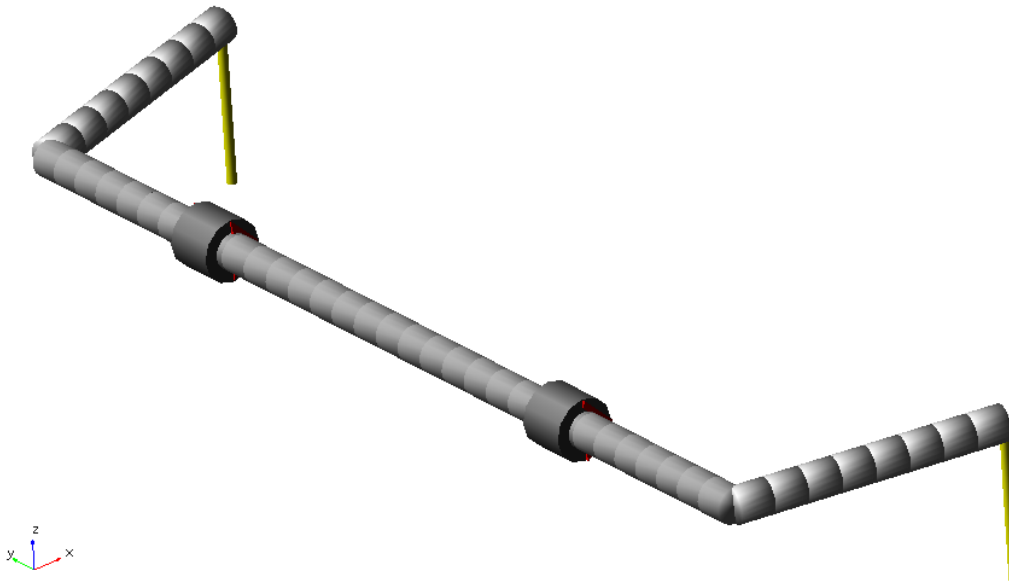


Figure 3.3 : Beam model of FED-Alpha anti-roll bar system.

There was a total of five different suspension systems employed in this investigation. The anti-roll bar was turned off in the first of them, and the system continued to function without it. Second, an anti-roll bar with a roll rate 50% lower than the one currently in use is employed. Third, the FED-Alpha vehicle's existing standard anti-roll bar is utilized. Fourth, an anti-roll bar is employed, but it has a roll rate 75% greater than the anti-roll bar in the vehicle. Finally, it was turned on by incorporating a controller within the anti-roll bar that was already in situ. Brief naming of these systems is as in Table 3.2.

Table 3.2 : The naming of vehicles with different anti-roll bars.

Vehicle	Naming
Vehicle without an anti-roll bar	Without ARB
Vehicle with %50 softer anti-roll bar	Less Stiff ARB
Standard vehicle setup	Standard ARB
Vehicle with %75 stiffer anti-roll bar	Stiffer ARB
Vehicle with active anti-roll bar	Algorithm

While changing the stiffness values for the anti-roll bar, calculations were made considering the effect of the change in thickness. While the calculations are being made, progress has been made in proportioning since the only variable is the radius. In this case, where only the hand calculation is made, the opposite wheel travel analyses were made while obtaining the actual thickness values. A representative example of this type of analysis is shown in Figure 3.4.

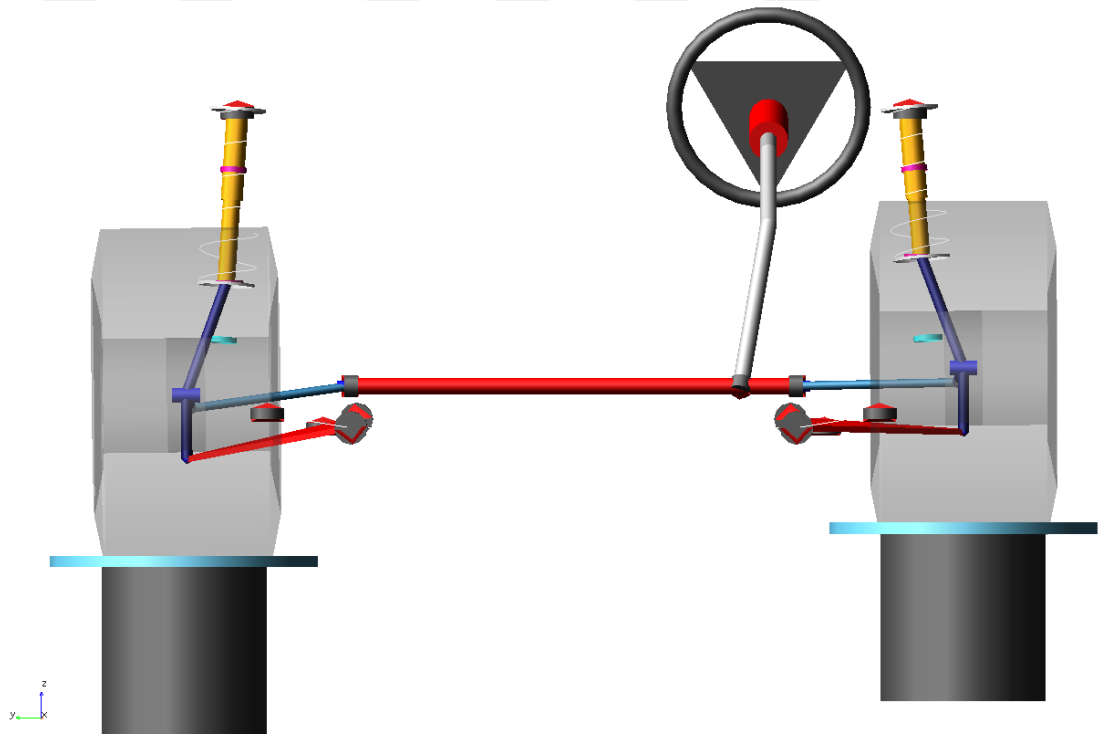


Figure 3.4 : Representative opposite wheel travel analysis.

The SAE method was taken as a reference when determining the final anti-roll bar thicknesses [37].

$$k_{bar} = \frac{PL^2}{2f_A} \quad (3.1)$$

P: Force at point “A”, N

L: Length of the anti-roll bar, mm

f_A : Displacement of point “A”, mm

k_{bar} : Roll rate of the anti-roll bar, Nmm/rad.

Point “A” is indicated in Figure 2.9.

The values obtained as a result of the calculations are as in Table 3.3.

Table 3.3 : The radius values of anti-roll bars.

Anti-roll bar	Radius
%50 softer	15.576mm
Standard	19.135mm
%75 stiffer	23.319mm

Anti-roll bars with the thickness values specified in Table 3.3 were used in the analyses.

3.3 Algorithm

Off-road vehicle vibration is frequently regarded as among the worst and most likely to result in significant discomfort or damage. Vibration intensity may noticeably increase when an off-road vehicle's speed rises [40]. A pleasant ride can preserve in-vehicle passengers' physical and mental health. Poor ride quality can make occupants feel uncomfortable, distracted, and less attentive to their surroundings, impairing their ability to execute their duty as a driver. Because of these reasons, it is crucial to have a vehicle with good ride quality.

In order to give occupants a reasonable level of comfort, a typical off-road vehicle suspension is reasonably soft and has substantial suspension displacements. The active anti-roll bar may make off-road vehicles safer by enhancing comfort and stability. Passive anti-roll bar use improves stability but reduces the comfort, particularly on rough roads [2].

In this study, analyses were performed on roads with different phase angles but with the same waviness and roughness values. According to the results obtained with passive system analysis, it has been observed that the anti-roll bar has a negative effect on comfort, especially on roads with high phase angles. It is aimed to eliminate this negativity on roads with high phase angles. However, another critical parameter to be

considered in the algorithm to be established is vehicle dynamics. The reason for this is to improve the ride quality of the vehicle while not compromising on safety. The contribution of the anti-roll bar to safety is known, especially in cornering situations. The primary purpose of the algorithm is to increase the ride quality while not adversely affecting other vehicle dynamics parameters.

It has been estimated that one of the most noticeable features that can be observed in the vehicle on high-phase roads will be in the direction of rollover. To solve this, roll rate values were examined to prevent the negative ride quality obtained in high phases. In studies based on roll rate in the algorithm, sufficient improvements in ride quality have been observed. However, it has been observed that these algorithms do not work adequately during cornering and adversely affect safety. After the roll rate, studies were carried out with roll acceleration, but it was also determined that these values were insufficient. With the studies done with the roll jerk (deg/s^3) with a threshold, it has been determined that the vehicle can provide sufficient comfort.

Two critical parameters that can be observed during the vehicle's cornering are the steering wheel angle and yaw rate. The primary purpose of this parameter is to understand that the vehicle is cornering and to ensure that the anti-roll bar is operational in this case. In poor road conditions, changes in the steering wheel angle may be observed due to the vehicle's continuous bumping or falling into a pothole, so it was not used as the determining parameter in this study. It has been decided to use this parameter as the determining value, where the change in the yaw rate value is related to the vehicle's route. Since the studies were carried out in rough road conditions, the yaw rate values obtained with the existing anti-roll bar were noted, and a threshold suitable for the algorithm was determined.

The algorithm created aligns with the specified information as in Figure 3.5.

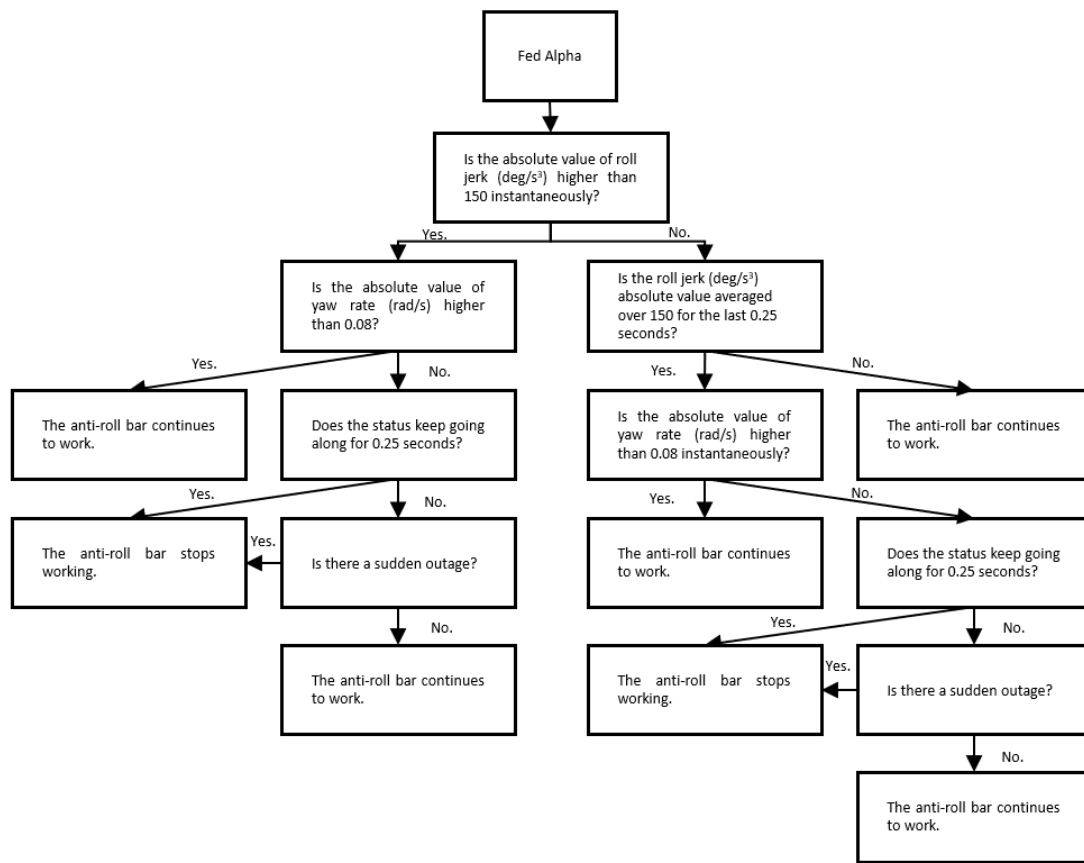


Figure 3.5 : The algorithm used in the study.

To explain the algorithm shown in Figure 3.5, roll rate and yaw rate values are obtained through the sensor located in the vehicle's center of gravity. Although there could be spikes that do not show the reality in these obtained values, the system will not stop working or become operational again when it encounters a single value. It is recommended that the sample rate of the data to be obtained with the sensor should be around 200 Hz. The roll rate value is integrated twice over time.

The first value to be observed is the roll jerk, which will indicate that the vehicle is in rough road conditions. As a result of the analysis, it was determined that if the roll jerk (deg/s^3) value exceeds 150 continuously, rough road conditions are entered. In addition, the yaw rate (rad/s) was monitored to ensure that the vehicle was not cornering if it exceeded 0.08. In cases where the roll jerk value exceeds 150 deg/s^3 , and the yaw rate value does not exceed 0.08 rad/s , the anti-roll bar will stop working. In order to prevent momentary interruptions, the average of both values in the last 0.25 seconds is tracked. If the average values meet the mentioned conditions, the anti-roll bar will continue not to work. If these conditions are not met, the anti-roll bar will remain in operation continuously. In addition, the conditions specified to prevent the

system from turning on and off continuously are realized in 0.25 seconds. For example, in order for the anti-roll bar not to work, it must continue for 0.25 seconds when conditions are triggered. In this way, it is aimed to obtain a more stable system by avoiding instant on-off operations.

3.3.1 Active anti-roll bar mechanism

There are two essential parameters for the previously mentioned algorithm to work. These values are yaw rate and roll rate. Yaw rate and roll rate values are read by the sensor located in the vehicle's center of gravity and transferred to the electronic control unit. After the data is transferred, there are two critical parts.

The yaw rate value is checked first when the anti-roll bar is integrated with the body. The limit value for this controlled value is 0.08 rad/s. If the read yaw rate value exceeds this limit, no action is taken, and the system continues to work as a passive anti-roll bar. If the read yaw rate value is lower than this limit value, the second important parameter, the roll jerk value, is started to be checked. The limit value for this parameter is 150 deg/s³. If it is in the initial state, data entry starts by assigning 1 to the empty values. It is checked whether the average of the roll jerk value is higher than the limit value in the specified time. If this value is lower than 150, the system exits passively and takes no action. However, if this value is higher than 150, the mechanism created to release the anti-roll bar is triggered. The system continues to release the anti-roll bar for the prescribed time.

If the anti-roll bar is free, the yaw rate value is checked first, as in the previous scenario. If this value exceeds the specified limit value, the anti-roll bar mechanism is triggered to return to its original position. If the yaw rate value is lower than the limit value, it is checked whether the average of the roll jerk value is high within the specified time interval. If the reading is lower than the limit value, the anti-roll bar system returns to its position within the prescribed time and acts as a passive anti-roll bar. If this value is higher than 150, no action is taken, and the anti-roll bar remains free in the system.

It is essential to transfer the value read from the ECU to the electric motor driver to release and recover the anti-roll bar. According to the value read from the output pin, the steel rope connected to the electric motor is released or retracted. Since the system works synchronously, the anti-roll bar's left and right body connection can be

controlled with a single motor. Thanks to the rotation torque of the motor, mechanical transfer to the rollers is provided. In this part, since the rope allows rotation and movement in every axis, it ensures that the anti-roll bar does not fulfill its primary task. The rope at the other end of the pulley is connected with a pin to the profile part of the bush, where the anti-roll bar is connected to the body. Thus, it can be ensured that this rope does not break from the body connection of the anti-roll bar and allows movement.

The length of the free rope is chosen not to prevent the movement of the anti-roll bar. In the event of the motor rotation, whose winding amount is determined, it will pull the rope enough to collect it. The engine should be selected with features that can perform the specified rotation within the stipulated time.

3.4 Simulation Types

3.4.1 Ride quality

The term "ride quality" describes how well a vehicle can isolate its occupants from the shocks, jounces, and vibrations that might enter the cabin through the suspension system. While a vehicle with poor ride quality may feel harsh or jarring to passengers, a vehicle with high ride quality will offer a smooth and pleasant ride, especially on bumpy or uneven roads.

The purpose of these analyzes is to determine how much the driver is affected by poor road conditions by determining the ride quality. The roughness and waviness values of the roads are the same and are 5.08 cm, and 2, respectively. Although four different road profiles were used, it was studied on roads with phase angles of 0° , 45° , 90° , and 180° .

The study's primary aim is to find out how much the driver is affected by different road conditions with different anti-roll bar setups. The FED-Alpha vehicle with four different anti-roll bars was used at this stage. The vehicle without an anti-roll bar, with %50 softer, standard, and %75 stiffer anti-roll bars are studied during these analyses.

NATO specifications determine the length of the road created within the scope of the study and are 1000 ft. (≈ 305 m.) long. In order to obtain comfort values, the vehicle was simulated on roads of this length and at different phase angles. For the accuracy

of the comparison, it is of great importance that the roads of the same lengths are completed at constant speeds.

In comfort analyses, tire pressure values are set at 35 psi.

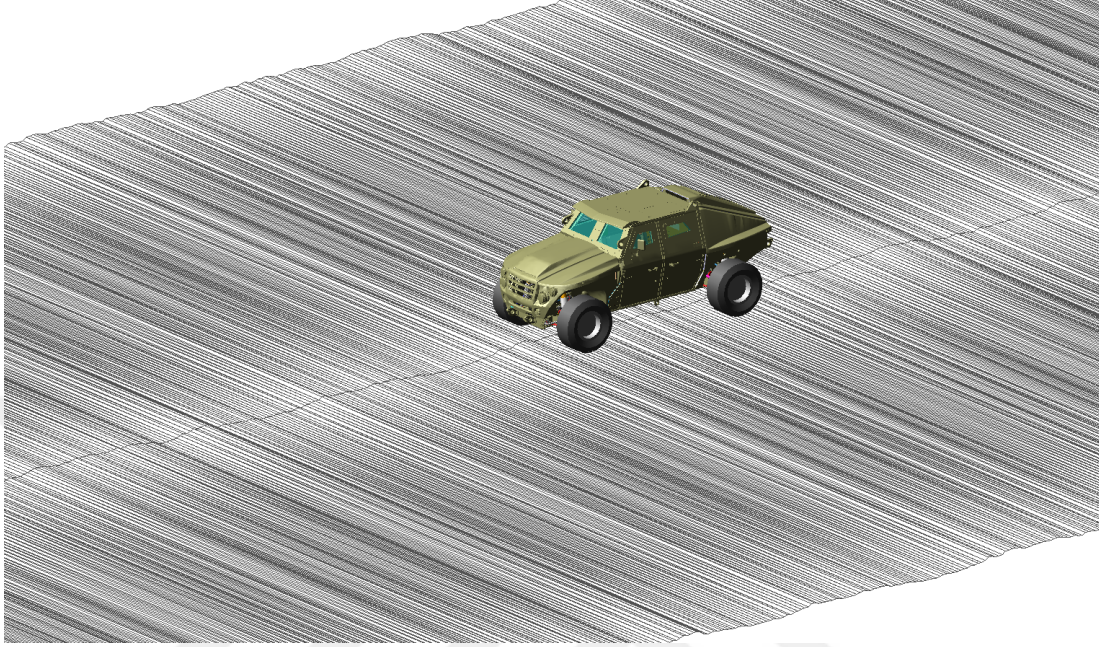


Figure 3.6 : Photo of 0° phase angle ride quality analysis.

A visual example of an analysis made on a 0° phase angle road is shown in Figure 3.6 in the analysis for ride quality.

Three different methods were used to determine ride quality. The first of these is the absorbed power method. This method finds a constant value for each axis using the acceleration values taken from the driver's seat in three different axes and the transfer functions as explained before in Equations 2.1, 2.2, and 2.3. If the power value exceeds 6 watts on any axis, it can be understood that the ride quality values are at an undesirable level for the specified axis. The second method is the frequency-weighted rms method. This method has been applied as specified in the ISO 2631 standard. While Equations 2.5 and 2.6 were used during the calculations, the k_x , k_y , and k_z coefficients were chosen as 1.4, 1.4, and 1.0, respectively, following the standard. The third method is for calculating the vibration dose value. Equation 2.7 was used while making these calculations.

In the analyzes made in the study, the speed values that the driver will be significantly affected by the road conditions were determined, and improvement studies were made for this. Obtained results are shown in graphs, making comparison easier.

While obtaining the values, a software program that facilitates these calculations and data processing software were used.

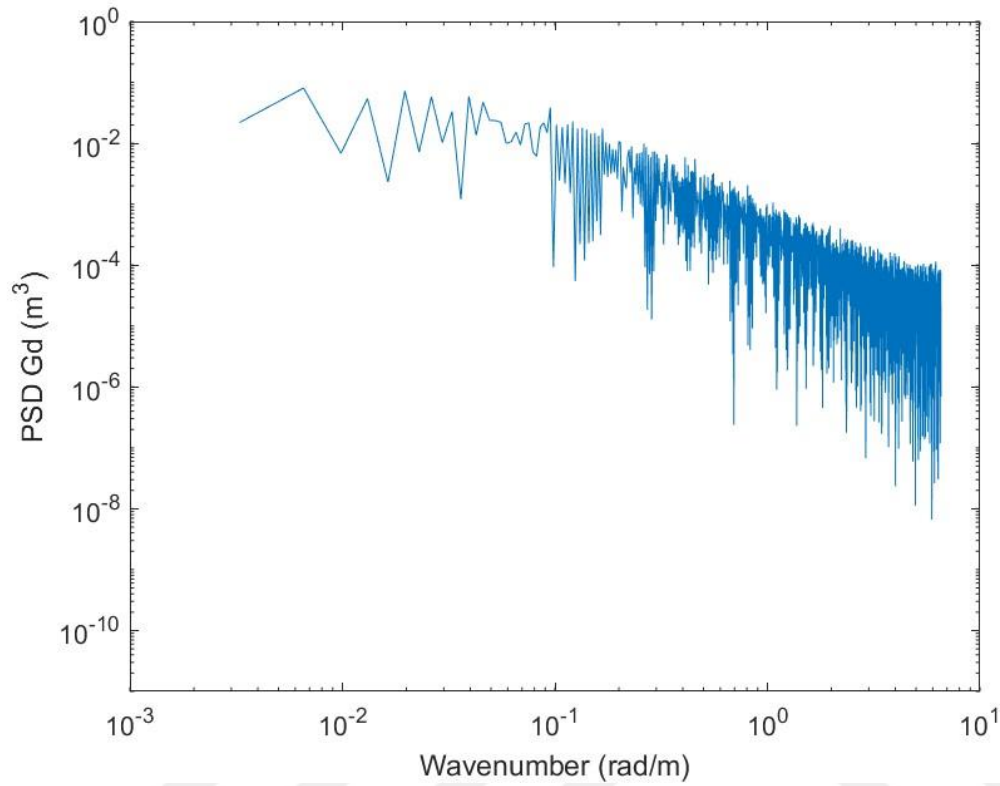


Figure 3.7 : PSD road graph as a function of wavenumber.

According to ISO 8608, a signal's limiting mean-square value per unit of frequency bandwidth is referred to as its PSD [41]. The road surface is identified and the properties of the road are understood using the power spectral density (PSD) approach.

3.4.2 ISO 3888-2 double-lane change maneuver

The test track dimensions for a closed-loop, extreme lane-change maneuver test to evaluate a vehicle's performance in avoiding obstacles, which is a particular aspect of vehicle dynamics and road-holding abilities, are specified in ISO 3888-2:2011. It assesses a vehicle's ability to avoid unexpected obstacles.

Figure 3.8 depicts the test method visually.

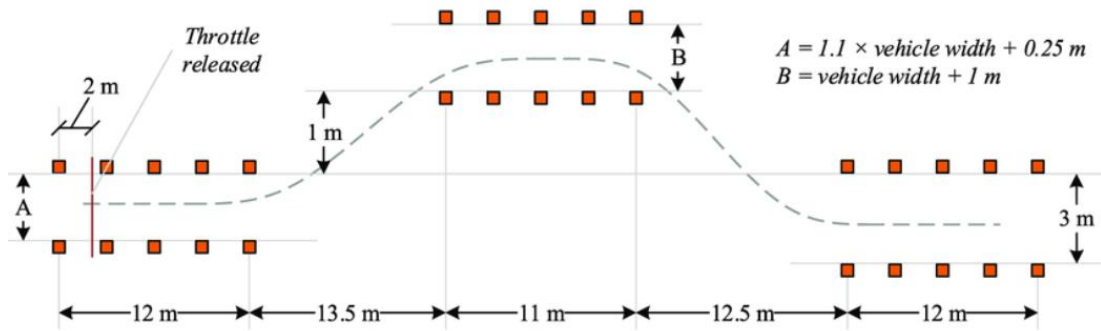


Figure 3.8 : The track specified in ISO 3888-2.

It has a 12 m-long entering lane at the beginning and a 12 m-long departure lane at the conclusion. A so-called side lane has a 1 m lateral offset between these two. The longitudinal space between the entry and side lanes is 13.5 meters, while the distance between the side and exit lanes is 12.5 meters. The test further stipulates that the throttle must be removed 2 meters after approaching the scene, and the remaining portions of the test must be completed only with steering without using the throttle or brakes.

In these analyses, three vehicles were used: vehicles without anti-roll bars, vehicles with standard anti-roll bars, and vehicles with active anti-roll bars. Tire pressure values are set to 60 psi.



Figure 3.9 : The trajectory example of an ISO 3888-2 analysis.

An example visual from the analysis of an ISO 3888-2 maneuver that has been made is in Figure 3.9.

The requirements that a vehicle must satisfy in order to pass the test are as follows. The vehicle must successfully complete the test without any cones being struck or leaving the authorized test area. Throughout the test, the vehicle must have a steady and predictable reaction to steering inputs. For example, the vehicle's lateral acceleration must fall within a given range, and its yaw rate must not exceed a predetermined threshold. The time required for the test, which varies based on the

vehicle's speed and design, must be met by the vehicle. The test must be carried out per the guidelines and specifications stated in the standard, including the use of the appropriate cone count and placement, vehicle speed, test setup, and the appropriate measurement and analysis methods.

These analyses noted the final speed values that all three vehicle types could complete without touching a cone. However, it was also checked whether other criteria mentioned above corresponding to this situation were met.

3.4.3 Fishhook maneuver

A driving test, known as the fishhook maneuver, is designed to gauge a vehicle's handling and stability in emergencies, such as when a driver must quickly change directions to escape an obstacle or an accident. A "fishhook" shape is made on the road by the vehicle making a move while moving straight ahead at a certain speed, then abruptly veering sharply to one side and then going back again.

Several characteristics are evaluated and recorded during the maneuver to assess the vehicle's handling and stability, including lateral acceleration, steering angle, and yaw rate. The test's objectives are to evaluate the vehicle's capacity to maintain stability and control in emergencies and to spot any handling or stability faults that might need fixing.

Table 3.4 : Analysis inputs for fishhook maneuvers.

Parameter	Unit	Value
Vehicle speed	km/h	60
Gear position	-	4
Duration	s	5
Steering	-	Variable
Tire pressures	psi	60

The values shown in Table 3.4 are the same for all vehicles and do not change. Figure 3.10 shows a graph showing the change in steering wheel angle per second.

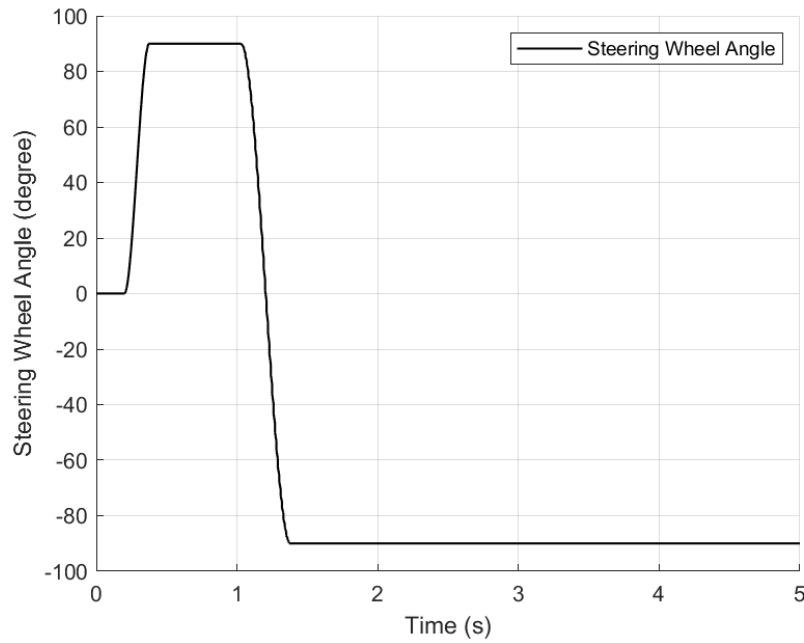


Figure 3.10 : The steering wheel angle over time in the fishhook maneuver.

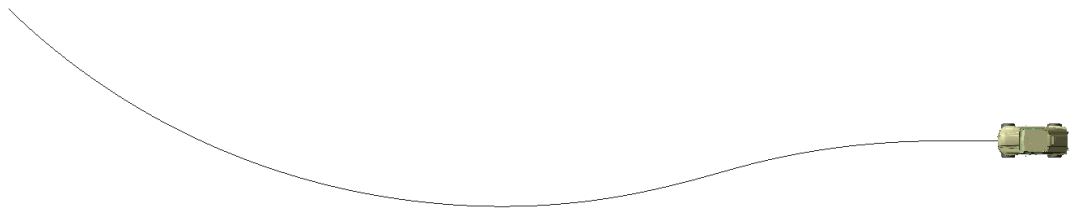


Figure 3.11 : The trajectory example of a fishhook maneuver analysis.

An example visual from the analysis of a fishhook maneuver that has been made is in Figure 3.11.

3.4.4 J-turn maneuver

The "J" shape of the vehicle's route gives the maneuver its name. The J-turn maneuver involves the driver approaching a T-shaped crossroads quickly before suddenly turning to the side and turning the vehicle 180 degrees to face the other way. The motorist then immediately accelerates in the other direction to leave the danger zone.

The J-turn maneuver calls for a high level of proficiency, expertise, and specific training and tools. High-performance engines give the required power and control to carry out the move safely and successfully, so rear-wheel-drive vehicles are usually used to complete it.

Table 3.5 : Analysis inputs for J-turn maneuvers.

Parameter	Unit	Value
Vehicle speed	km/h	60
Gear position	-	4
Duration	s	10
Start time of test	s	0.2
Turning direction	-	Left
Steering wheel angle	°	90
Steer scale factor	-	1.0
Steer rate	°/s	512
Tire pressures	psi	60

The values shown in Table 3.5 are the same for all vehicles and do not change.

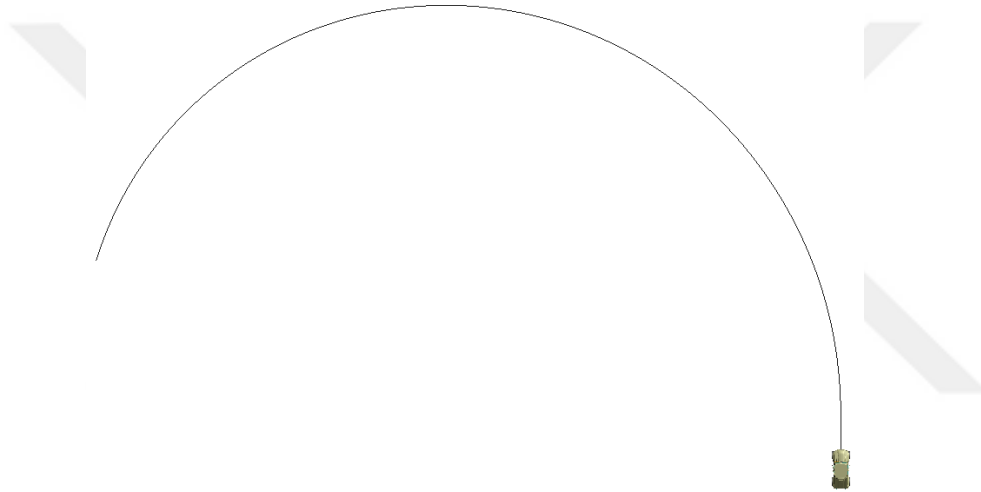


Figure 3.12 : The trajectory example of a J-turn maneuver analysis.

An example visual from the analysis of a J-turn maneuver that has been made is in Figure 3.12.

3.4.5 Single-lane change maneuver

A driving technique called the single lane change maneuver is used to assess a vehicle's stability and handling when changing lanes fast. In order to pass the test, the driver must control the vehicle through a succession of cones or markers spaced out in a single lane, simulating an evasive movement to avoid an obstruction.

The test involves driving the vehicle steadily while the driver makes a quick maneuver to the side to avoid an obstruction. At the end of the maneuver, the vehicle moves into the left lane and maintains this lane parallel to the previous lane with a constant speed. In order to evaluate the vehicle's performance under various circumstances, the test is often run repeatedly in both directions. Like the double-lane change maneuver, the single-lane change maneuver has a smaller turning radius and a more straightforward path.

Table 3.6 : Analysis inputs for single-lane change maneuvers.

Parameter	Unit	Value
Vehicle speed	km/h	60
Gear position	-	4
Duration	s	10
Steering	-	Variable
Turning direction	-	Left
Tire pressures	psi	60

The values shown in Table 3.6 are the same for all vehicles and do not change.



Figure 3.13 : The trajectory example of a single-lane change maneuver analysis.

An example visual from the analysis of a single-lane change maneuver that has been made is in Figure 3.13. Change in steering wheel angle per second is also shown in Figure 3.14.

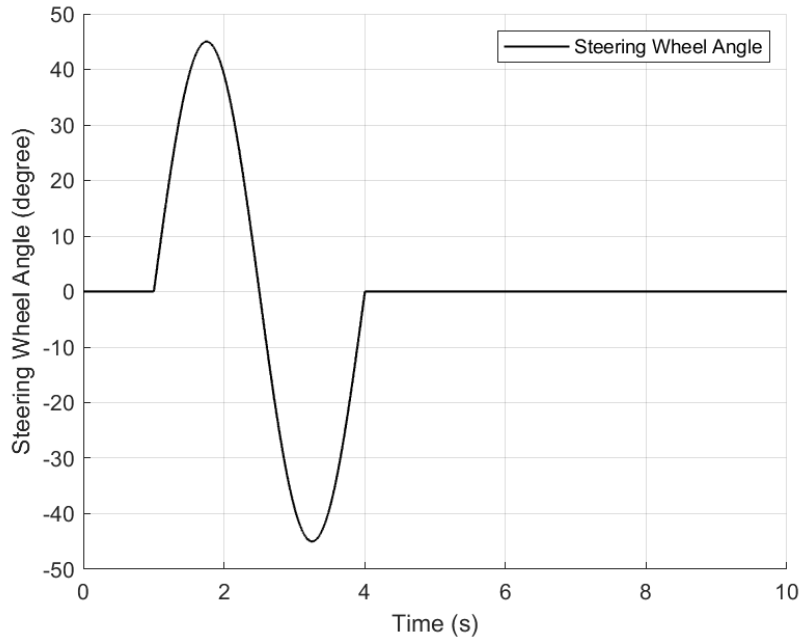


Figure 3.14 : The steering wheel angle over time in a single-lane change maneuver analysis.

3.4.6 Constant radius cornering

Constant radius cornering is a driving maneuver to assess a vehicle's stability and handling while navigating a course with a constant radius curve. In order to make this move, the vehicle must be driven around a circle at a constant speed with a constant turning radius. Usually, the turn's radius is selected to mimic a real-world driving situation, such as a motorway off-ramp or a winding road. However, the selected parameter for these analyses is lateral acceleration.

The constant radius cornering maneuver is a typical test that automakers, safety groups, and regulatory agencies use to judge how well passenger vehicles and light trucks handle and are stable and how well they adhere to set industry norms and rules. The test evaluates the vehicle's lateral acceleration, yaw rate, and other dynamic handling characteristics.

Table 3.7 : Analysis inputs for constant radius cornering maneuvers.

Parameter	Unit	Value
Vehicle speed	km/h	54
Gear position	-	4
Duration	s	30
Lateral acceleration	g	0.2, 0.3, 0.4, 0.5
Steering	-	Changes over vehicle and scenario
Turning direction	-	Left
Maneuver start distance	m	10
Tire pressures	psi	60

The values shown in Table 3.7 are the same for all vehicles except for steering wheel angle and lateral acceleration.

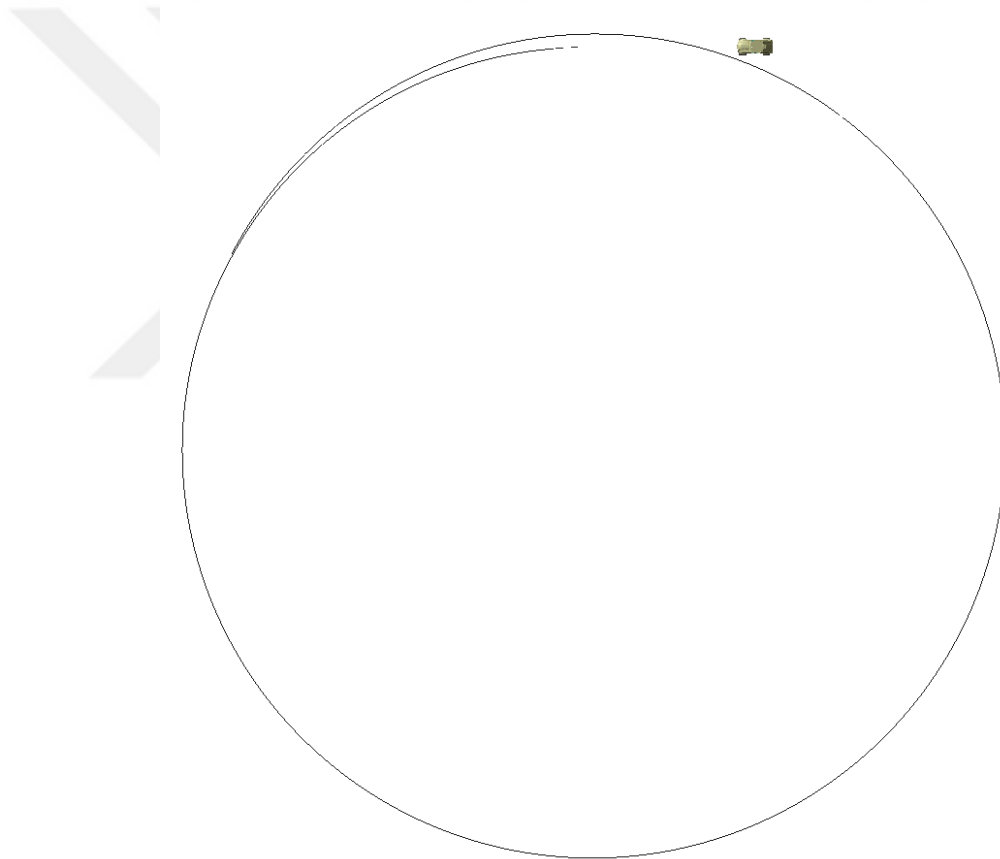


Figure 3.15 : The trajectory example of a constant radius cornering analysis.

An example visual from the analysis of a single-lane change maneuver that has been made is in Figure 3.15.



4. SIMULATION RESULTS

How the results of the analysis specified in this section are drawn, and scenarios of these analyses are explained in the previous sections.

4.1 Ride Quality Analysis Results

4.1.1 Absorbed power

The absorbed power values are calculated separately according to the axes in the absorbed power calculations. Thus, it has been determined which axis is affected more by the phase angle of the driver.

4.1.1.1 0° phase angle

Absorbed power values for all three axes at the 0-degree phase angle are shown in Figures 4.1, 4.2, and 4.3.

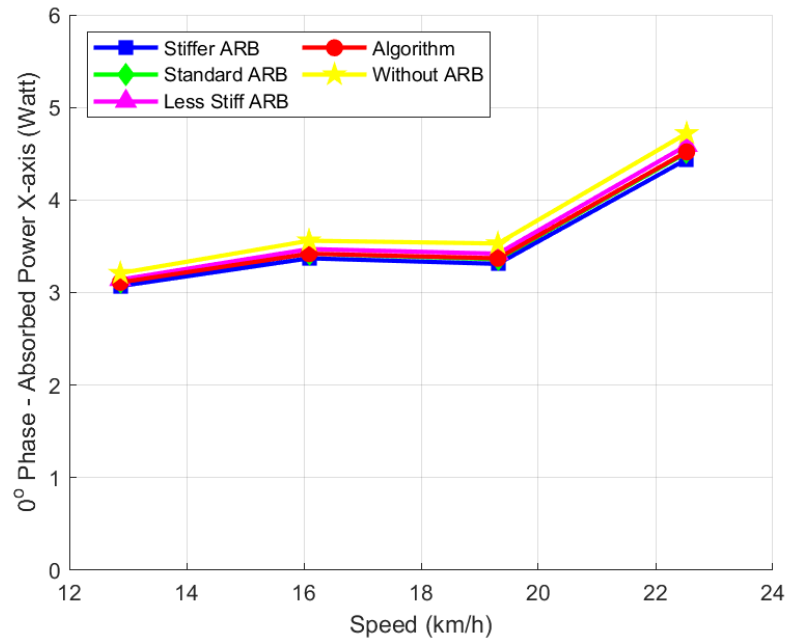


Figure 4.1 : The driver seat base's absorbed power measurements as a function of vehicle speed in the x-axis at the 0° phase angle.

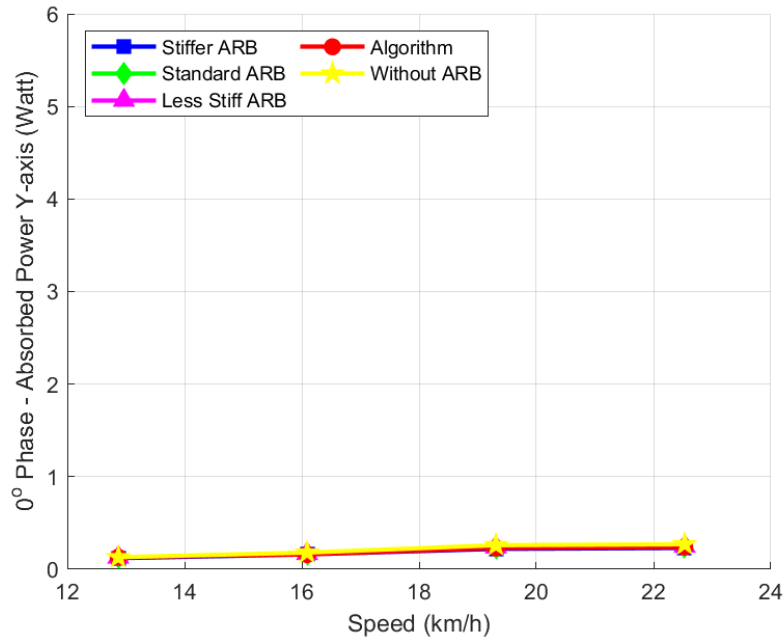


Figure 4.2 : The driver seat base's absorbed power measurements as a function of vehicle speed in the y-axis at the 0° phase angle.

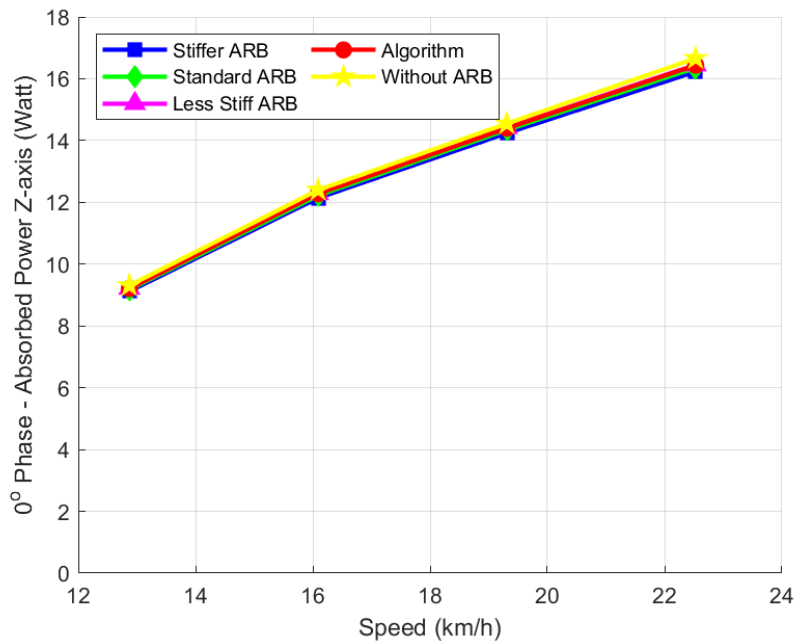


Figure 4.3 : The driver seat base's absorbed power measurements as a function of vehicle speed in the z-axis at the 0° phase angle.

Table 4.1 : 6-watt speed values at the 0° phase angle.

Vehicle	6-watt speed
Vehicle without an ARB	9.9 kph
Vehicle with softer ARB	9.9 kph
Vehicle with standard ARB	10.0 kph
Vehicle with stiff ARB	10.0 kph
Vehicle with active ARB	10.0 kph

Since Fed-Alpha vehicle is designed as a fuel efficiency demonstrator, 6-watt speed values are significantly lower than a typical off-road military vehicle. In order to elaborate the effect of anti-roll bar characteristics on ride quality, simulations were performed at higher speeds than the 6-watt speeds.

When looking at the 6-watt speed values at a 0-degree phase angle, it is seen that the differences are small. At this phase angle, the effect of the anti-roll bar on the 6-watt speed is small.

Although the absorbed power values obtained on the x-axis are higher than those on the y-axis, they are lower than those on the z-axis at the 0-degree phase angle. The values are pretty close to each other, but it can be seen that the model's comfort level with a more rigid anti-roll bar is good on this axis, albeit at deficient levels.

The absorbed power values obtained in the y-axis at the 0-degree phase angle are significantly lower than the other two axes. It is observed that the values vary between 0 and 0.5 watts. It can be deduced that the anti-roll bar installations at this phase angle do not significantly affect the driver on the y-axis.

The absorbed power values obtained in the z-axis at the 0-degree phase angle are high. As mentioned earlier, studies are evaluated for situations where the driver is exposed to poor road conditions. Although the differences are small, the model's ride quality with a more rigid anti-roll bar is observed to be better at deficient levels.

The active anti-roll bar system is close to the standard anti-roll bar system because the algorithm does not meet the criteria specified in Figure 3.5 in this phase angle. It can be observed for each axis that the value increases with increasing speed.

4.1.1.2 45° phase angle

In this part, the comparison of the values according to the 0-degree phase angle is also made. According to the results obtained, the changes in the absorbed power values according to the axes are also noted.

Absorbed power values for all three axes at the 45-degree phase angle are shown in Figures 4.4, 4.5, and 4.6.

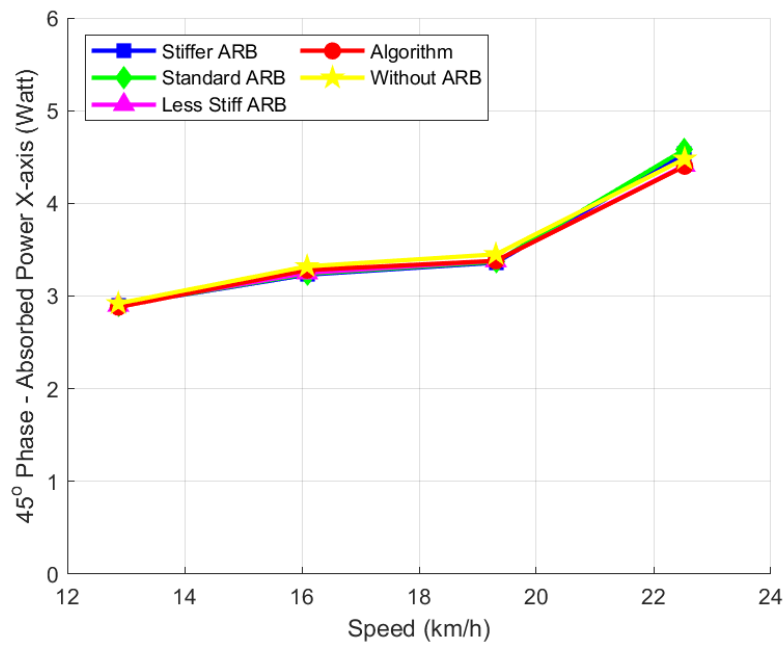


Figure 4.4 : The driver seat base's absorbed power measurements as a function of vehicle speed in the x-axis at the 45° phase angle.

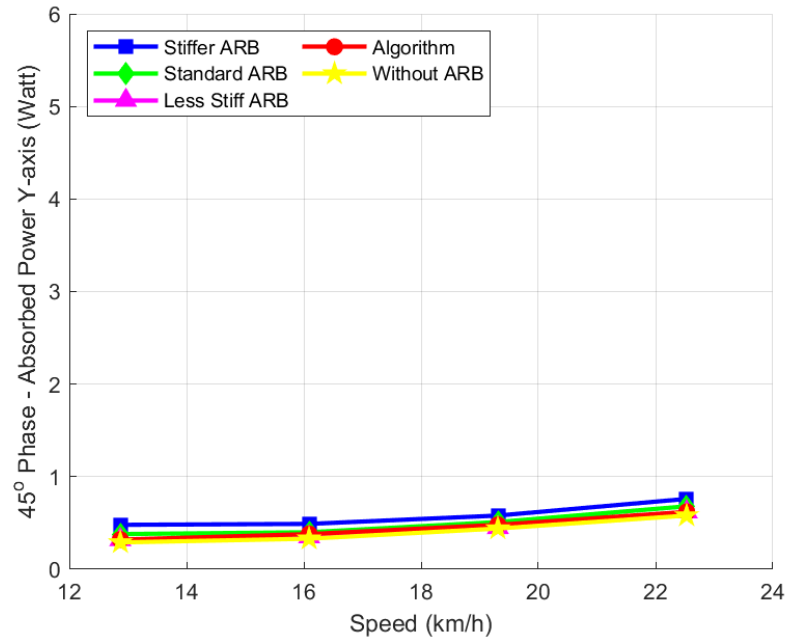


Figure 4.5 : The driver seat base's absorbed power measurements as a function of vehicle speed in the y-axis at the 45° phase angle.

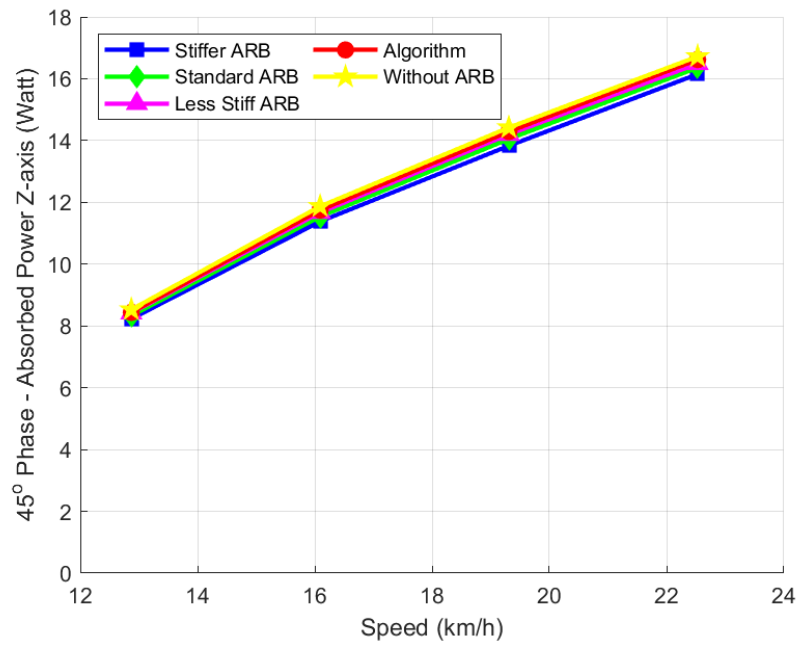


Figure 4.6 : The driver seat base's absorbed power measurements as a function of vehicle speed in the z-axis at the 45° phase angle.

Table 4.2 : 6-watt speed values at the 45° phase angle.

Vehicle	6-watt speed
Vehicle without an ARB	10.6 kph
Vehicle with softer ARB	10.7 kph
Vehicle with standard ARB	10.7 kph
Vehicle with stiff ARB	10.8 kph
Vehicle with active ARB	10.7 kph

The differences are slight when looking at the 6-watt speed values at a 45-degree phase angle. It can be seen from the results that the anti-roll bar did not have a significant effect on this phase.

Although the absorbed power values obtained on the x-axis are higher than those on the y-axis, they are lower than those on the z-axis in the 45-degree phase angle. Since the values are very close, it is tough to compare on this axis. However, it was observed that the absorbed power values of the stiffer anti-roll bars increased with the increase in speed.

The absorbed power values obtained in the y-axis at the 45-degree phase angle are significantly lower than the other two axes. It is observed that the values vary between 0 and 1 watt. It can be deduced that the anti-roll bar installations at this phase angle do not significantly affect the driver on the y-axis. With the increase in the torsional stiffness of the anti-roll bar, which can be observed from Figure 4.5, the ride quality was adversely affected. In the analyses made at this phase angle, the effect of the anti-roll bar on the ride quality on the y-axis is negative.

The absorbed power values obtained in the z-axis at the 45-degree phase angle are high. Although the differences are small, it is observed that the model's ride quality level with a more rigid anti-roll bar is better, albeit at deficient levels.

At this phase angle, the active system approaches the model without the anti-roll bar in terms of watts.

It can be seen for each axis that the absorbed power value increases with increasing speed. According to the results obtained, there is a decrease in the x-axis, a significant increase in the y-axis, and a decrease in the z-axis.

4.1.1.3 90° phase angle

In this part, the comparison of the values according to the 0 and 45-degree phase angles is also made. According to the results obtained, the changes in the absorbed power values according to the axes are also noted.

Absorbed power values for all three axes at the 90-degree phase angle are shown in Figures 4.7, 4.8, and 4.9.

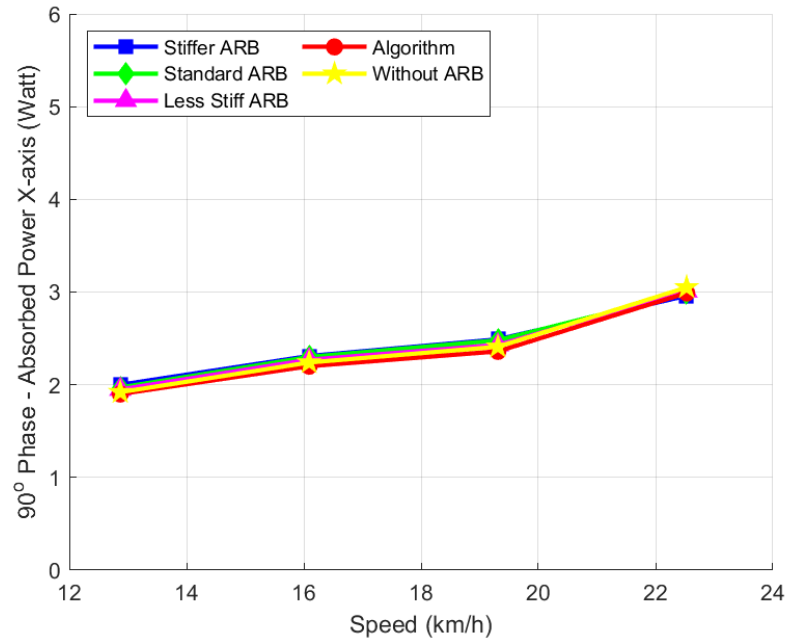


Figure 4.7 : The driver seat base's absorbed power measurements as a function of vehicle speed in the x-axis at the 90° phase angle.

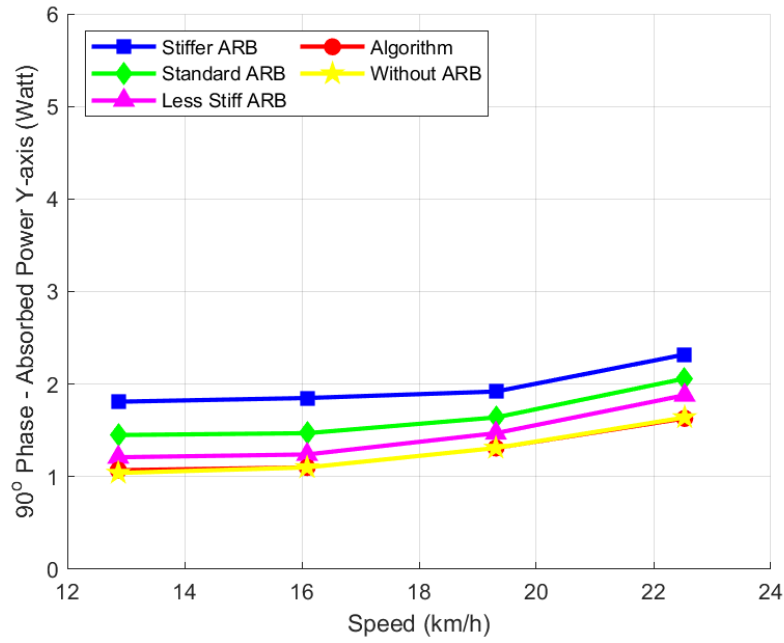


Figure 4.8 : The driver seat base's absorbed power measurements as a function of vehicle speed in the y-axis at the 90° phase angle.

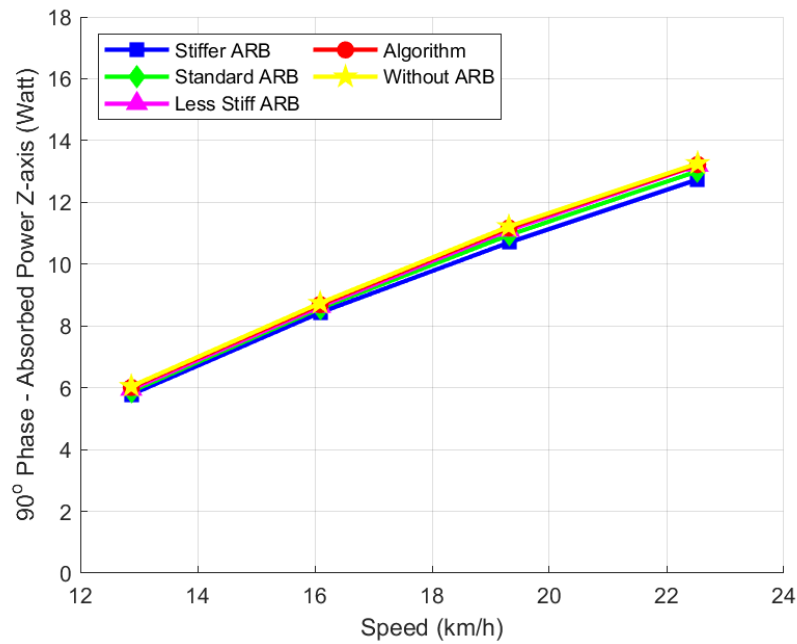


Figure 4.9 : The driver seat base's absorbed power measurements as a function of vehicle speed in the z-axis at the 90° phase angle.

Table 4.3 : 6-watt speed values at the 90° phase angle.

Vehicle	6-watt speed
Vehicle without an ARB	12.8 kph
Vehicle with softer ARB	12.9 kph
Vehicle with standard ARB	13.0 kph
Vehicle with stiff ARB	13.1 kph
Vehicle with active ARB	12.9 kph

When we look at the 6-watt speed values at 90 degrees phase angle, it is seen that the differences are low in percentage. The differences are slightly more pronounced than those obtained at other phase angles.

Although the absorbed power values obtained in the x-axis at the 90-degree phase angle are higher than the values obtained at the y-axis, it is observed that this difference is relatively low compared to 0 and 45-degree phase angles. Although the differences in the results obtained are close to each other, it is possible to make comparisons. The model without an anti-roll bar up to 21 km/h has been observed to provide the best ride quality in the studied speed region. However, when exceeding this speed, the ride quality of the stiffer anti-roll bar has become better than the others.

The absorbed power values obtained in the y-axis at the 90-degree phase angle are lower than the other two axes. However, it is observed that this difference decreases considerably compared to 0 and 45-degree phase angles. It is observed that the values vary between 1 and 2.5 watts. In this phase angle, it can be seen that the effect of anti-roll bar installations on the driver in the y-axis increases compared to the analyzes made at lower phases. The ride quality was negatively affected by the increase in the torsional stiffness of the anti-roll bar, which can be observed in Figure 4.8. In the analyses made at this phase angle, the effect of the anti-roll bar on the ride quality on the y-axis is negative, as is the case at lower phase angles.

The absorbed power values obtained in the z-axis at the phase angle of 90 degree are high. With the increase in speed in vehicles with different anti-roll bar setups, the difference between different setups has become more apparent.

At this phase angle, the active system is almost the same in watts as the model without an anti-roll bar. This is because the criteria stated in Figure 3.5 are constantly being met.

It can be seen for each axis that the absorbed power value increases with increasing speed. The differences in the results became more evident compared to the 0 and 45-degree phase angles. It is observed that the values in the x-axis decrease by approximately 35% compared to the 0-degree phase angle values. It is observed that the values in the y-axis increase by approximately 700% compared to the 0-degree phase angle values. It has been observed that the values in the z-axis decrease by approximately 25% according to the 0-degree phase angle.

4.1.1.4 180° phase angle

In this part, the comparison of the values according to the 0, 45, and 90-degree phase angles is also made. According to the results obtained, the changes in the absorbed power values according to the axes are also noted.

Absorbed power values for all three axes at the 180-degree phase angle are shown in Figures 4.10, 4.11, and 4.12.

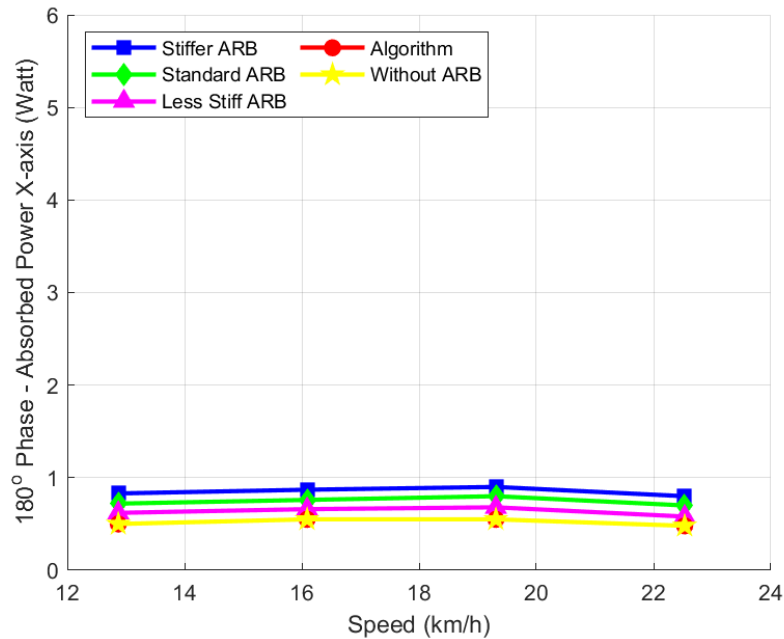


Figure 4.10 : The driver seat base's absorbed power measurements as a function of vehicle speed in the x-axis at the 180° phase angle.

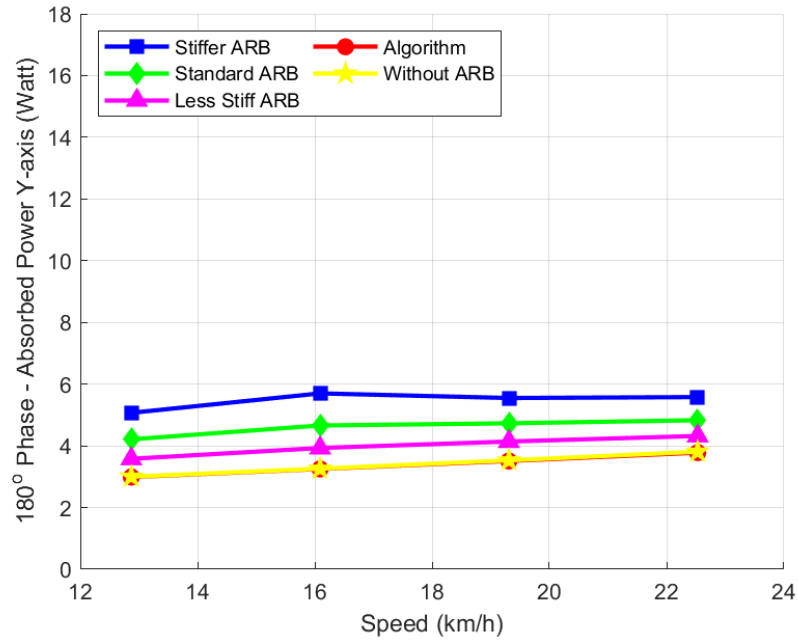


Figure 4.11 : The driver seat base's absorbed power measurements as a function of vehicle speed in the y-axis at the 180° phase angle.

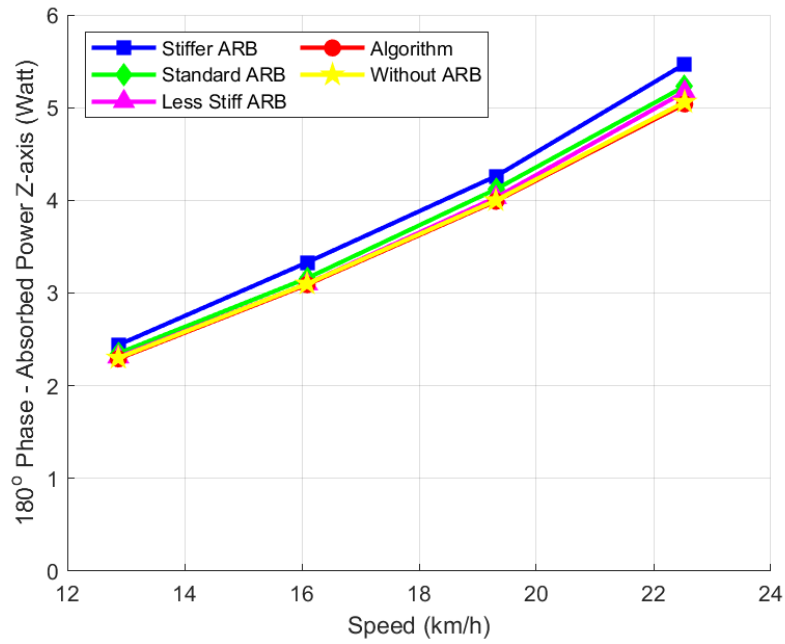


Figure 4.12 : The driver seat base's absorbed power measurements as a function of vehicle speed in the z-axis at the 180° phase angle.

Table 4.4 : 6-watt speed values at the 180° phase angle.

Vehicle	6-watt speed
Vehicle without an ARB	25.2 kph
Vehicle with softer ARB	24.7 kph
Vehicle with standard ARB	24.5 kph
Vehicle with stiff ARB	23.9 kph
Vehicle with active ARB	25.2 kph

The differences are apparent at the 6-watt speed values at 180 degrees phase angle. Vehicles that do not have an anti-roll bar and an active anti-roll bar have higher 6-watt speed values than other installations. This symbolizes that the threshold value is reached later in the 180-degree phase angle, and the ride quality is better than others.

The absorbed power values obtained in the x-axis at the 180-degree phase angle are significantly low compared to other axes. There was a significant reduction in the results obtained, and the values ranged from 0 to 1 watt. It can be observed that the absorbed power values are higher in the system with the anti-roll bar with the highest torsional stiffness value.

It is observed that the absorbed power values obtained in the y-axis at the 180-degree phase angle are higher compared to the other axes. However, it is also observed that the absorbed power values in the y-axis are less affected by the velocity than in the z-axis. The absorbed power values at the y-axis vary between 2 and 6 watts depending on the anti-roll bar setup. In this phase angle, it can be seen that the effect of anti-roll bar installations on the driver in the y-axis increases considerably compared to the analyses made at lower phases. The ride quality was negatively affected by the increase in the stiffness of the anti-roll bar, which can be observed in Figure 4.11. In the analyses made at this phase angle, the effect of the anti-roll bar on the ride quality on the y-axis is negative, as is the case at lower phase angles.

A significant decrease was observed in the absorbed power values obtained in the z-axis at the 180-degree phase angle compared to the lower phase angles. With the increase in speed in vehicles with different anti-roll bar setups, the difference between different setups has become more apparent. In addition, it was determined that the vehicle, which has a more rigid anti-roll bar, has a noticeably worse ride quality than the others.

At this phase angle, the active system is the same in watts as the model without the anti-roll bar. This is because the criteria specified in Figure 3.5 are constantly met.

While the increase in velocity does not have a significant effect on the x-axis, the increase in absorbed value with the increase in the velocity on the y-axis is meager. The axial differences concerning the 0, 45, and 90-degree phase angles are sharp. It is observed that the values in the x-axis decrease by approximately 80% compared to the 0-degree phase angle values. It is observed that the values in the y-axis increase by approximately 900% compared to the 0-degree phase angle values. It has been observed that the values in the z-axis decrease by approximately 70% according to the 0-degree phase angle.

At this phase angle, it can be deduced that the anti-roll bar has a negative effect on the ride quality on each axis. It has been observed that the system with an active anti-roll bar has the same values as the vehicle model without an anti-roll bar. From an axial point of view, based on absorbed power values, the improvement in ride comfort on the x-axis is 30%, the improvement in the y-axis is approximately 30%, and the improvement in the z-axis is around 5%. Since the results in this phase angle are considered, the effect of the y-axis on the driver is more, and a significant improvement in ride quality has been achieved with the active anti-roll bar system.

4.1.2 Frequency-weighted rms

The total vibration values of weighted rms acceleration are calculated according to the calculations mentioned before.

4.1.2.1 0° phase angle

The total vibration values of weighted rms acceleration at the 0-degree phase angle are shown in Figure 4.13, 4.14, 4.15, and 4.16.

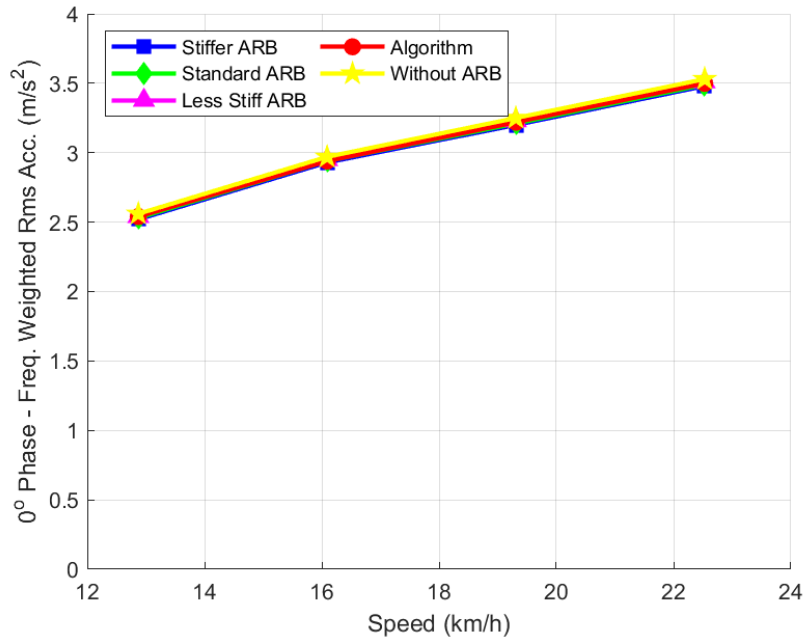


Figure 4.13 : Weighted rms vibration values of driver seat base as a combination of all axes as a function of vehicle speed at 0° phase angle.

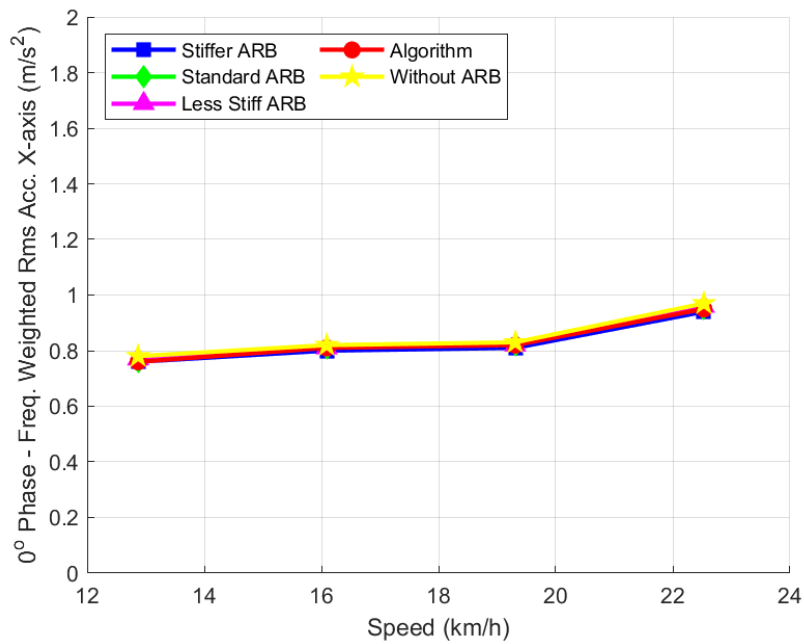


Figure 4.14 : The driver seat base's weighted rms vibration values as a function of vehicle speed in the x-axis at the 0° phase angle.

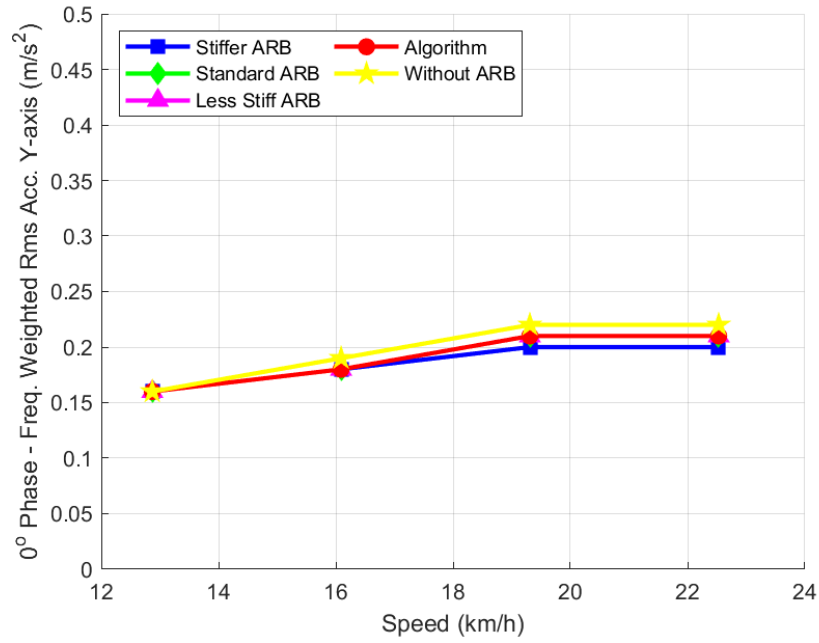


Figure 4.15 : The driver seat base's weighted rms vibration values as a function of vehicle speed in the y-axis at the 0° phase angle.

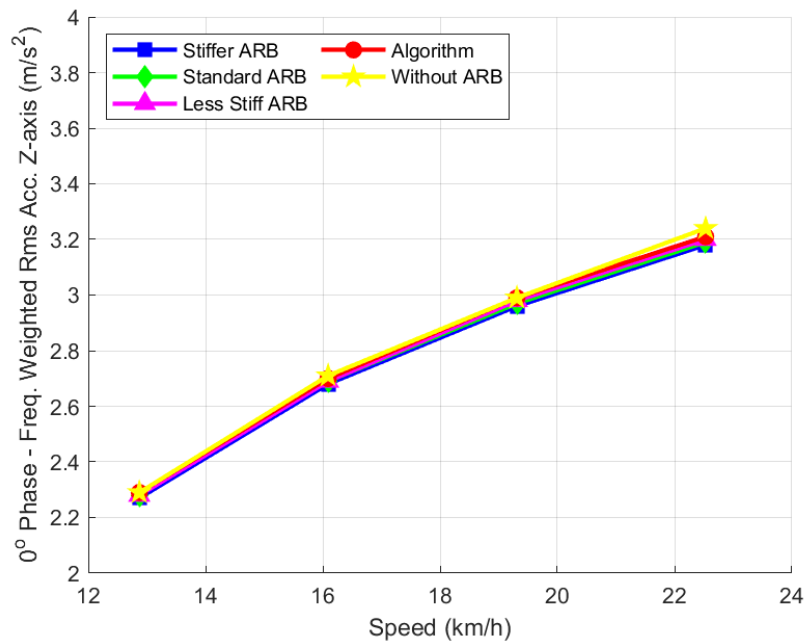


Figure 4.16 : The driver seat base's weighted rms vibration values as a function of vehicle speed in the z-axis at the 0° phase angle.

When the total vibration value of weighted rms is examined at the 0-degree phase angle, it is observed that the differences are small. As mentioned earlier, analyzes were conducted to examine the worst cases. Despite tiny differences, it is seen that the

system with the active anti-roll bar has the same values as the system with the standard anti-roll bar.

The resulting values are in similar synergy with the absorbed power values, and the axis differences in this phase angle are very low.

In this case, the active anti-roll bar system works like a standard anti-roll bar since the algorithm mentioned earlier does not meet the required criteria.

4.1.2.2 45° phase angle

In this part, the comparison of the values according to the 0-degree phase angle is also made. The total vibration values of weighted rms acceleration at the 45-degree phase angle are shown in Figure 4.17, 4.18, 4.19, and 4.20.

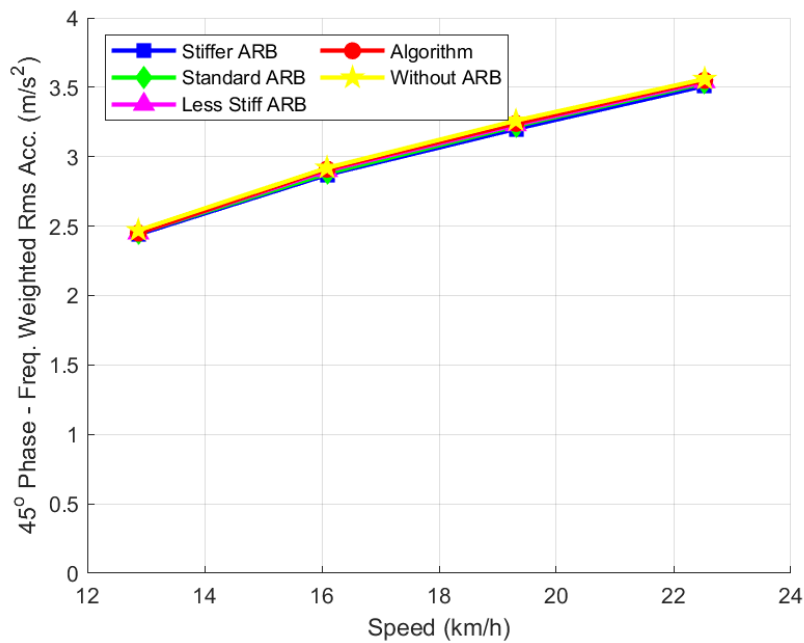


Figure 4.17 : Weighted rms vibration values of driver seat base as a combination of all axes as a function of vehicle speed at 45° phase angle.

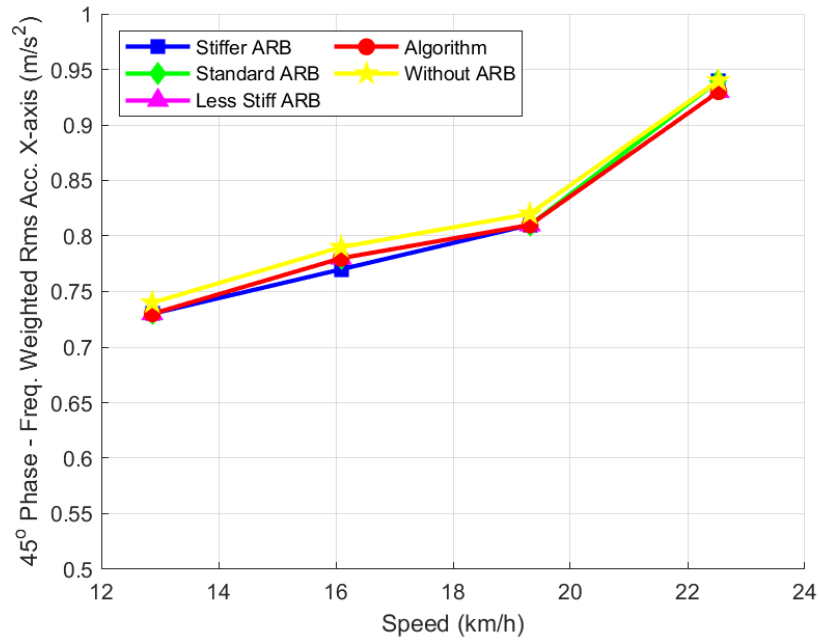


Figure 4.18 : The driver seat base's weighted rms vibration values as a function of vehicle speed in the y-axis at the 45° phase angle.

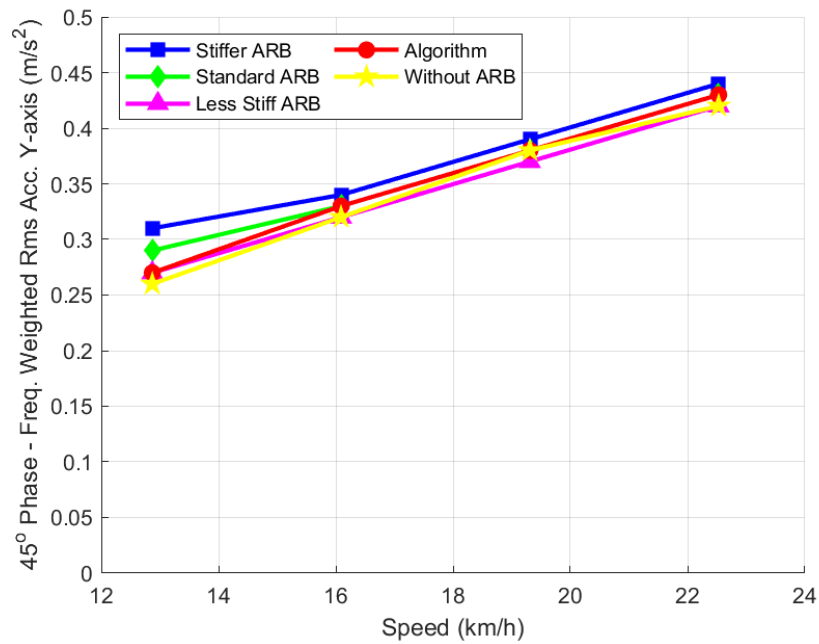


Figure 4.19 : The driver seat base's weighted rms vibration values as a function of vehicle speed in the y-axis at the 45° phase angle.

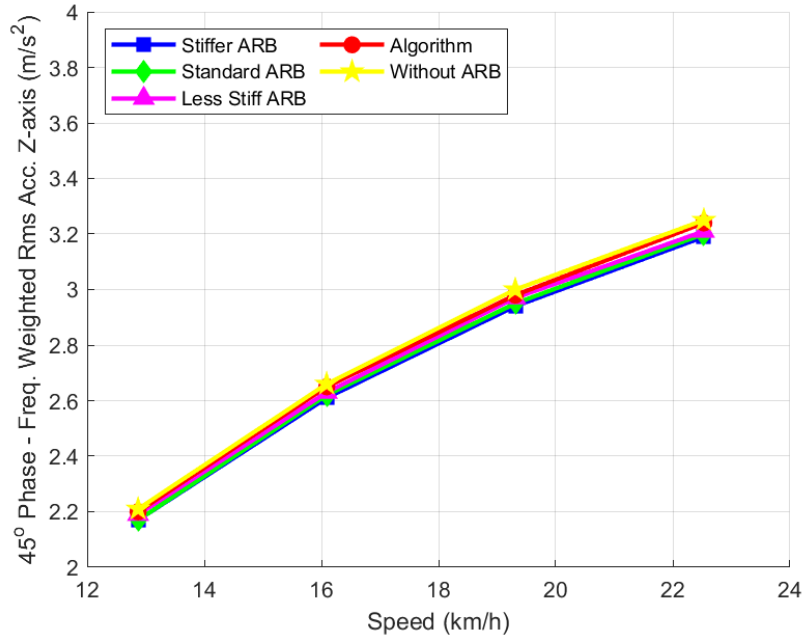


Figure 4.20 : The driver seat base's weighted rms vibration values as a function of vehicle speed in the z-axis at the 45° phase angle.

When the total vibration value of weighted rms is examined at the 45-degree phase angle, it is observed that the differences are small. With the increase in speed, the rms values increase, and this has a negative effect on the ride quality.

When compared with the 0-degree phase angle, it can be observed that the differences are small. Although the difference between these phase angles is meager, it is observed that the ride quality is better at the 45-degree phase angle up to 21 km/h speed, but at the values above it, the ride quality at the 0-degree phase angle is better. In this phase angle, the effect of the anti-roll bar on the ride quality is small compared to the passive systems.

The resulting values are in similar synergy with the absorbed power values, and the axis differences in this phase angle are very low.

The active system approaches the model without the anti-roll bar regarding rms values at this phase angle. Although the conditions shown in Figure 3.5 are occasionally offered, they are seldom provided throughout the simulation.

4.1.2.3 90° phase angle

The total vibration values of weighted rms acceleration at the 90-degree phase angle are shown in Figure 4.21, 4.22, 4.23, and 4.24.

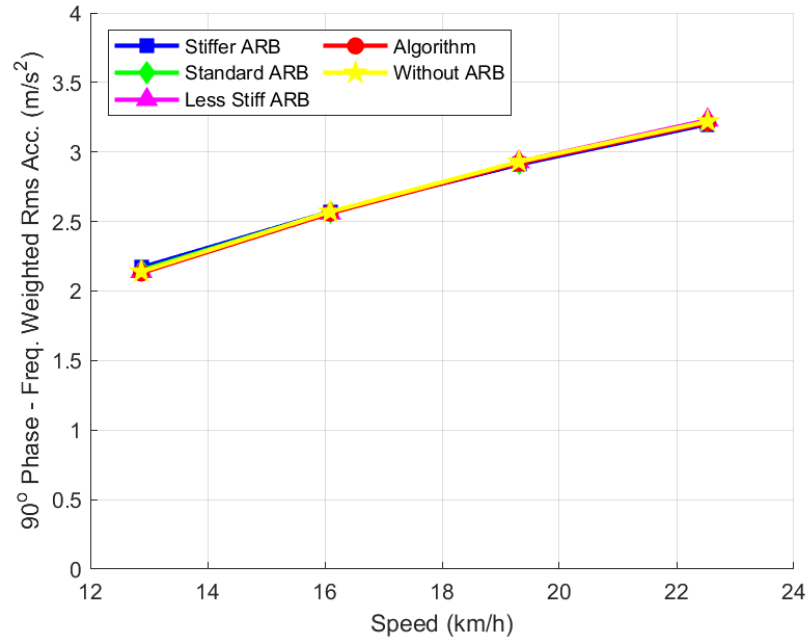


Figure 4.21 : Weighted rms vibration values of driver seat base as a combination of all axes as a function of vehicle speed at 90° phase angle.

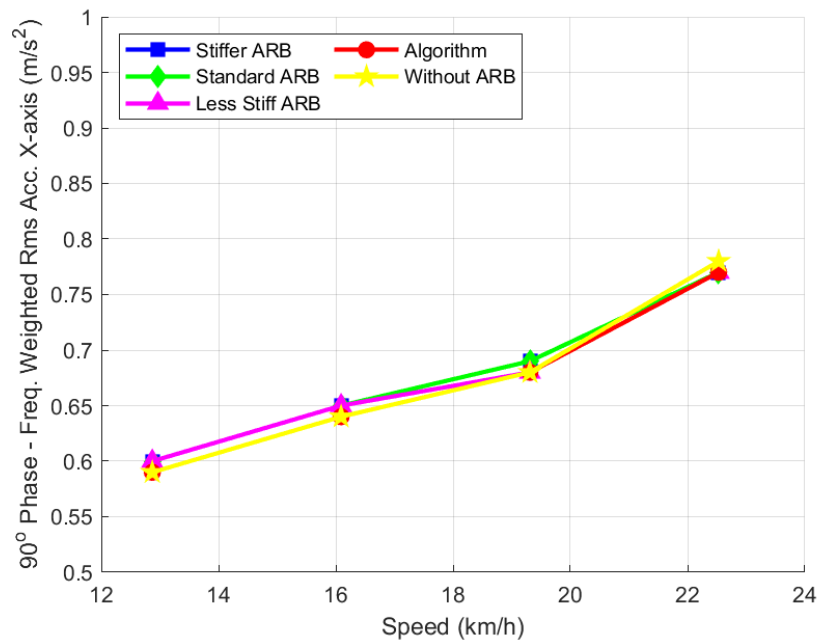


Figure 4.22 : The driver seat base's weighted rms vibration values as a function of vehicle speed in the x-axis at the 90° phase angle.

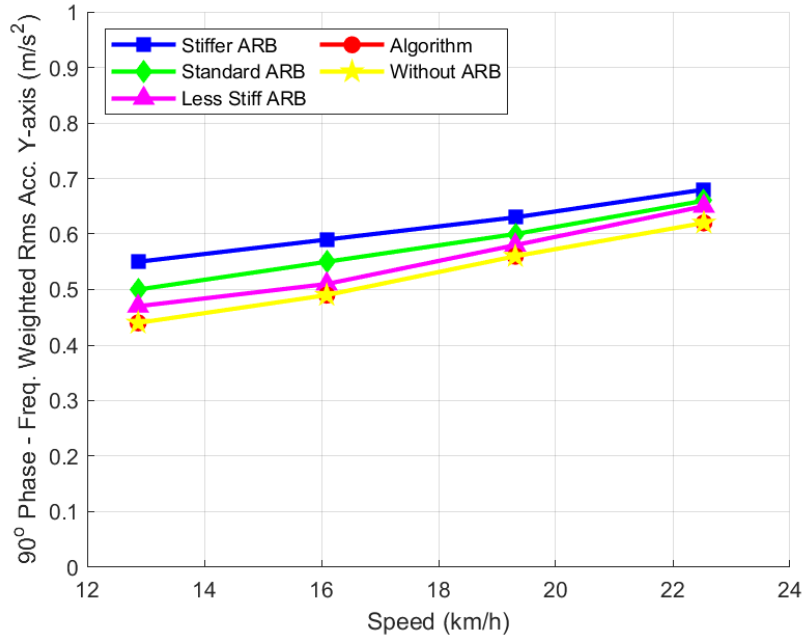


Figure 4.23 : The driver seat base's weighted rms vibration values as a function of vehicle speed in the y-axis at the 90° phase angle.

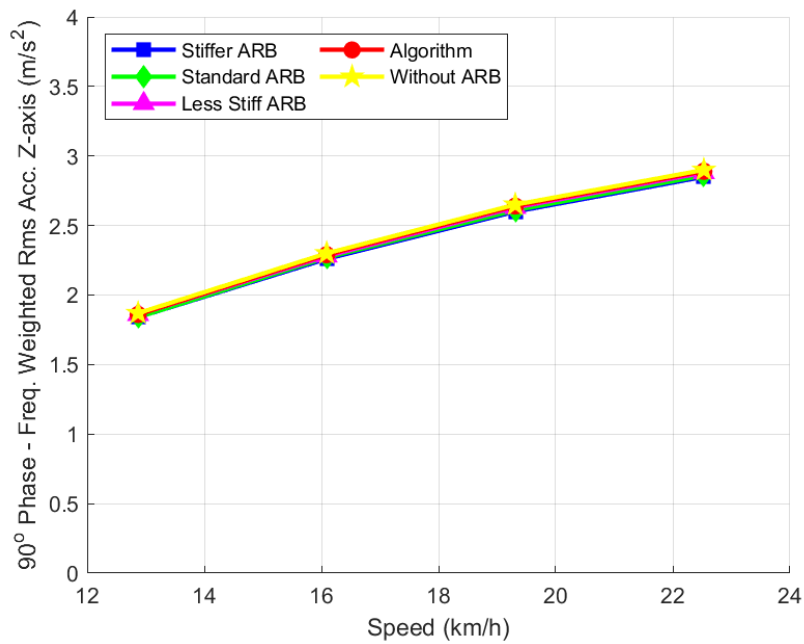


Figure 4.24 : The driver seat base's weighted rms vibration values as a function of vehicle speed in the z-axis at the 90° phase angle.

When the total vibration value of weighted rms is examined at the 90-degree phase angle, it is observed that the differences become apparent. With the increase in speed, the rms values increase, which is negative for the ride quality. However, it has been observed that the model without the anti-roll bar up to 16 km/h has a better ride quality.

The differences are more pronounced when compared with 0 and 45-degree phase angles. There is an approximate 15% decrease in these values obtained at the 90-degree phase angle compared to the 0-degree phase angle. In this phase angle, it can be concluded that the effect of the anti-roll bar on the ride quality becomes more critical with the increase in the phase angle compared to the passive systems.

The resulting values are in similar synergy with the absorbed power values. Although the differences in the longitudinal and vertical axis are low, the negative effect of the anti-roll bar on the ride quality can be observed in the lateral axis. The active system's wattage is nearly identical to the type without an anti-roll bar at this phase angle. This is due to the continual fulfillment of the requirements listed in Figure 3.5.

4.1.2.4 180° phase angle

In this part, the comparison of the values according to the 0, 45 and 90-degree phase angles is also made. The total vibration values of weighted rms acceleration at the 180-degree phase angle are shown in Figure 4.25, 4.26, 4.27, and 4.28.

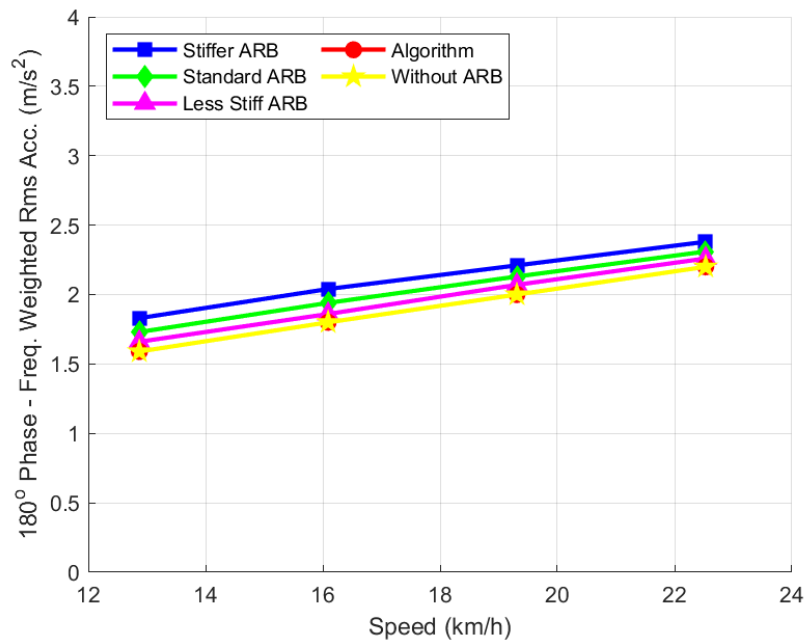


Figure 4.25 : Weighted rms vibration values of driver seat base as a combination of all axes as a function of vehicle speed at 180° phase angle.

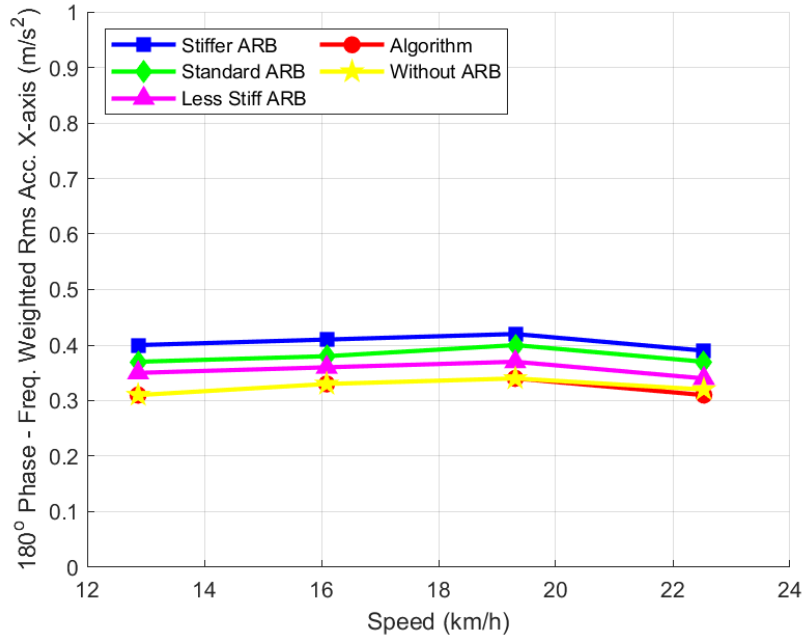


Figure 4.26 : The driver seat base's weighted rms vibration values as a function of vehicle speed in the x-axis at the 180° phase angle.

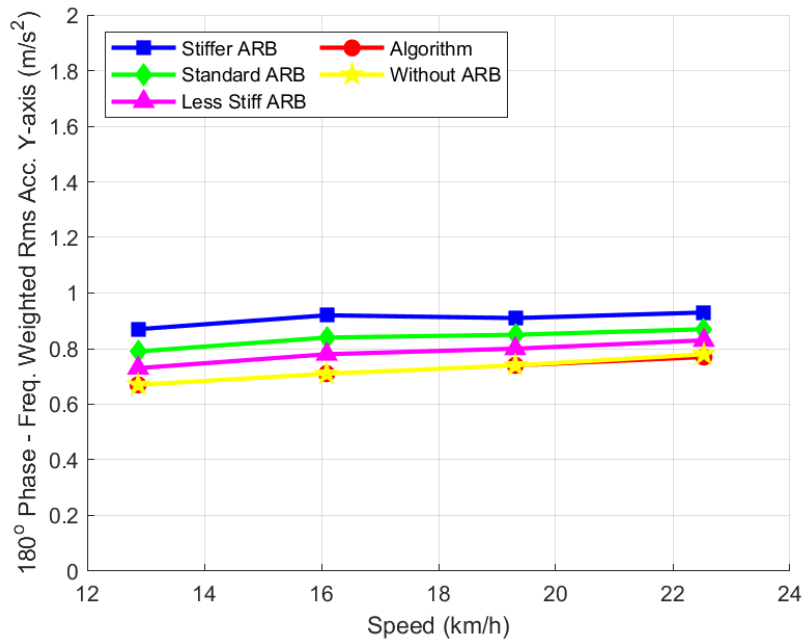


Figure 4.27 : The driver seat base's weighted rms vibration values as a function of vehicle speed in the y-axis at the 180° phase angle.

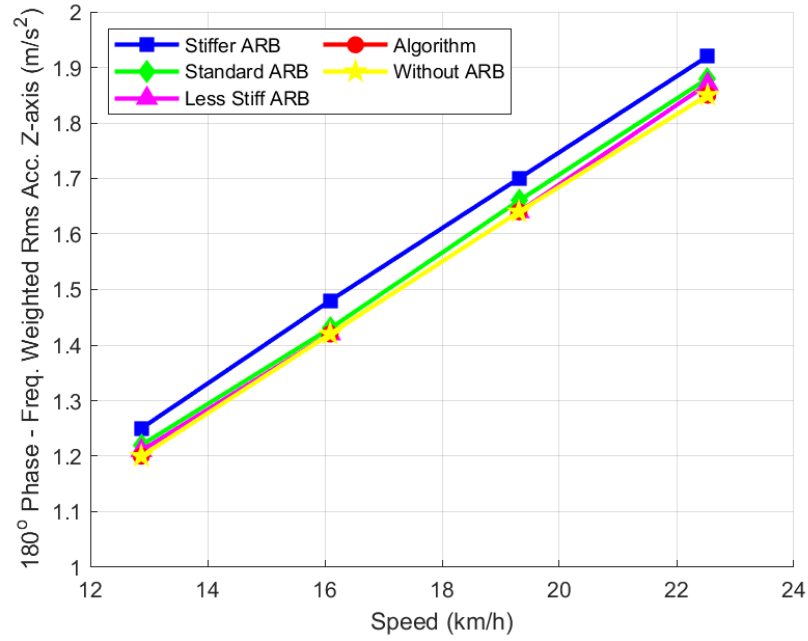


Figure 4.28 : The driver seat base's weighted rms vibration values as a function of vehicle speed in the z-axis at the 180° phase angle.

The differences are evident when the total vibration value of weighted rms is examined at the 180-degree phase angle. With the increase in speed, the rms values increase, as in other phase angles, adversely affecting the ride quality.

The differences can be clearly observed when compared with 0, 45, and 90-degree phase angles. There is a decrease of approximately 45% in these values obtained at the 180-degree phase angle compared to the 0-degree phase angle. In this phase angle, the effect of the anti-roll bar on the ride quality is more critical with the increase in the phase angle compared to the passive systems.

The resulting values are in a similar order to the absorbed power values. It can be seen that the anti-roll bar has a negative effect on every axis. The negative effect on the longitudinal and lateral axis is relatively high.

At this phase angle, the active system is the same in rms values as the model without the anti-roll bar. This is because the criteria specified in Figure 3.5 are constantly met. With this behavior of the active anti-roll bar system, a significant improvement in driving comfort has been observed compared to the passive anti-roll bar system.

4.1.3 Vibration dose value

The total vibration dose values are calculated according to the calculations mentioned before.

4.1.3.1 0° phase angle

The total vibration dose values at the 0-degree phase angle are shown in Figure 4.29, 4.30, 4.31, and 4.32.

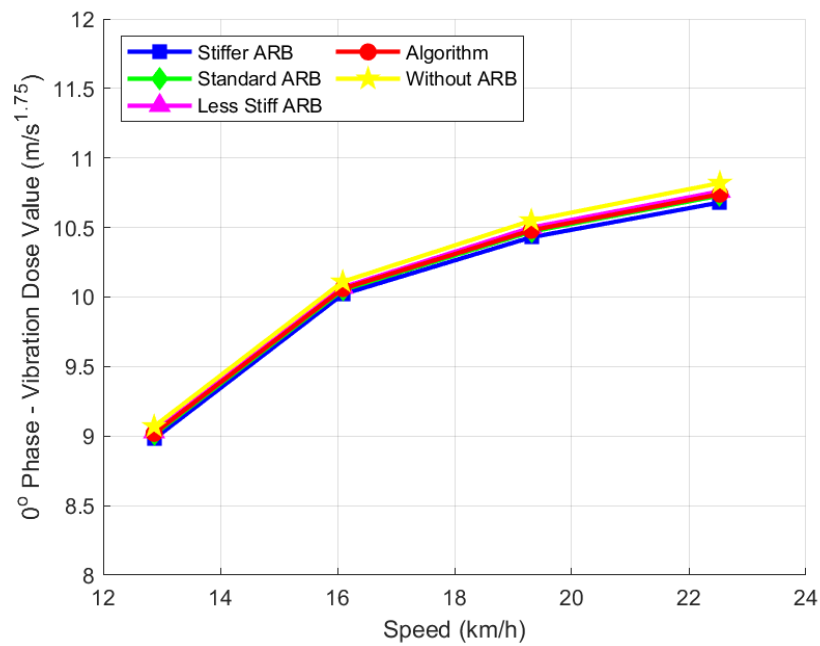


Figure 4.29 : Vibration dose values of driver seat base as a combination of all axes as a function of vehicle speed at 0° phase angle.

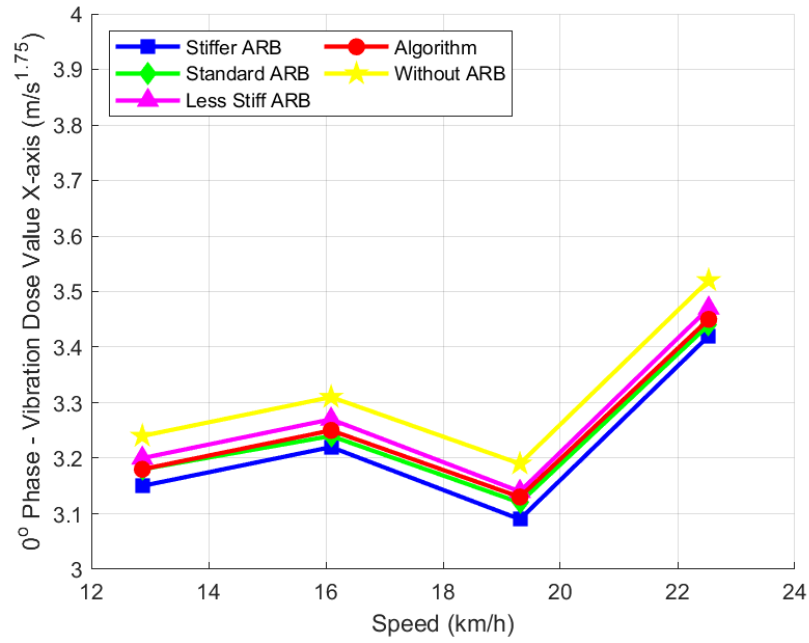


Figure 4.30 : The driver seat base's vibration dose values as a function of vehicle speed in the x-axis at the 0° phase angle.

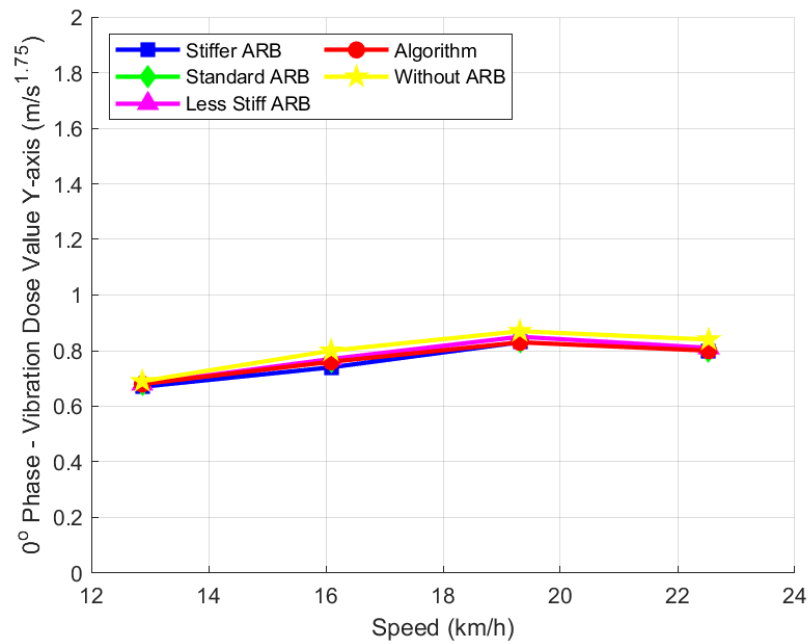


Figure 4.31 : The driver seat base's vibration dose values as a function of vehicle speed in the y-axis at the 0° phase angle.

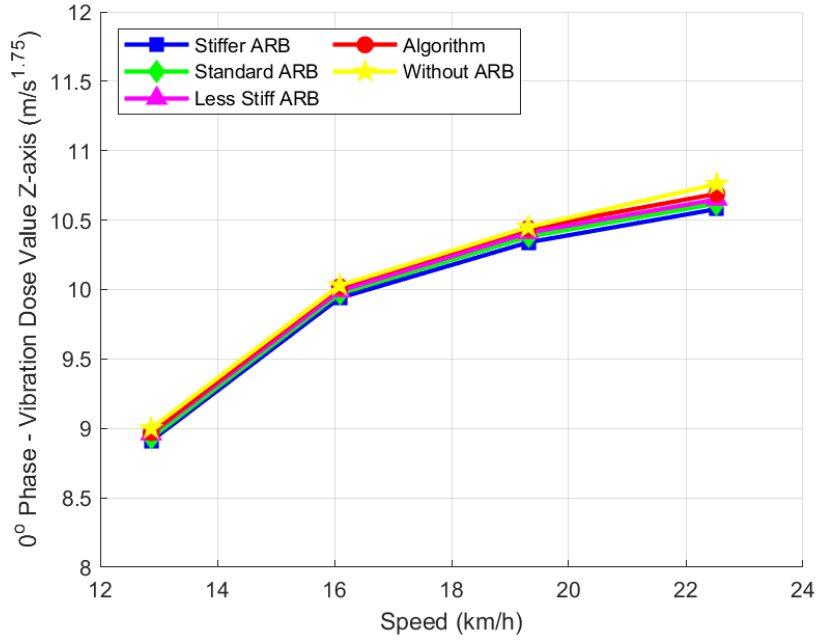


Figure 4.32 : The driver seat base's vibration dose values as a function of vehicle speed in the z-axis at the 0° phase angle.

When the total vibration dose values are examined at the 0-degree phase angle, it is observed that the differences are small. As mentioned earlier, analyses were conducted to examine the worst cases. The vehicle model with a stiffer anti-roll bar has quite better ride quality.

The resulting values are in a similar order to the absorbed power and weighted rms values. The anti-roll bar has no significant axial effect in this phase angle. As with low-phase angles, the axis that the driver is most affected by is the vertical axis.

Despite tiny differences, it is seen that the system with the active anti-roll bar has the same values as the system with the standard anti-roll bar. In this case, the active anti-roll bar system works like a standard anti-roll bar since the algorithm mentioned earlier does not meet the required criteria.

4.1.3.2 45° phase angle

In this part, the comparison of the values according to the 0-degree phase angle is also made. The total vibration dose values at the 45-degree phase angle are shown in Figure 4.32, 4.33, 4.34, and 4.35.

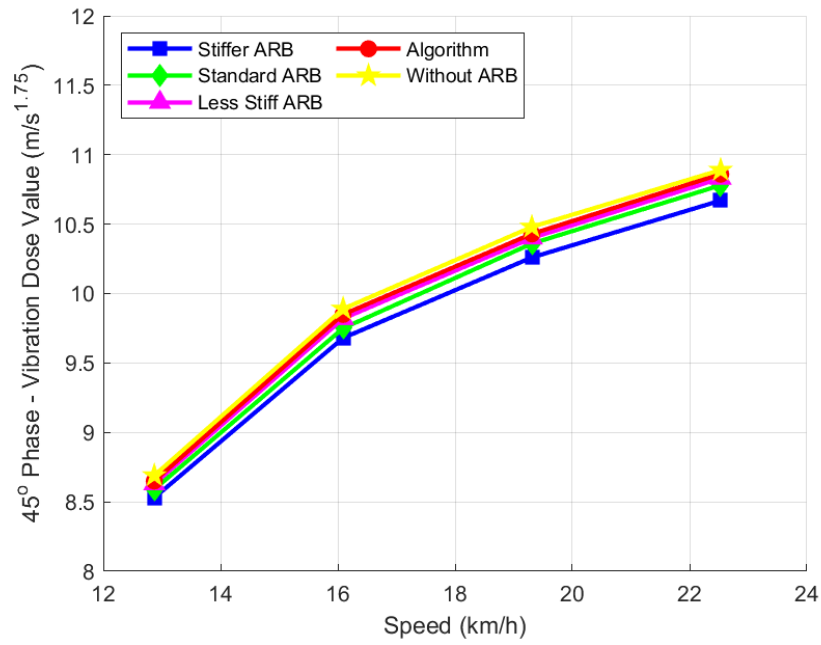


Figure 4.33 : Vibration dose values of driver seat base as a combination of all axes as a function of vehicle speed at 45° phase angle.

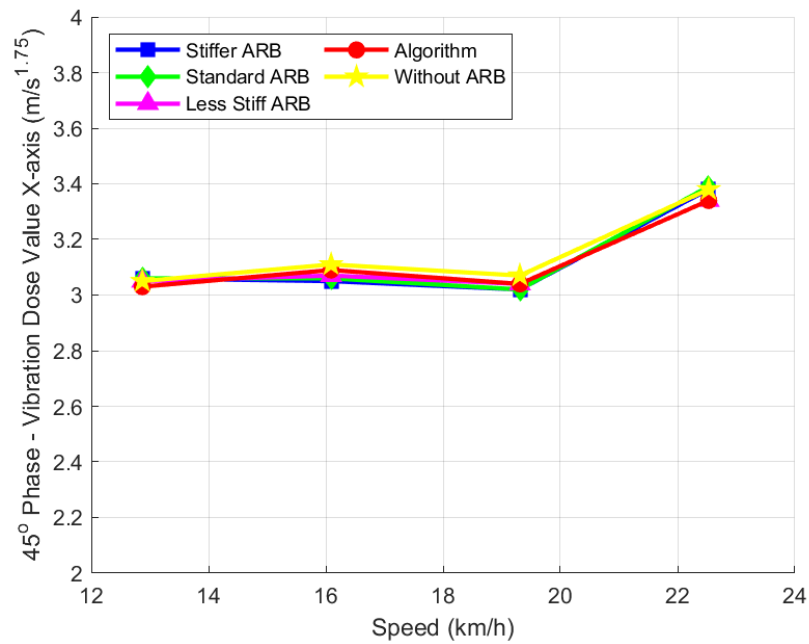


Figure 4.34 : The driver seat base's vibration dose values as a function of vehicle speed in the x-axis at the 45° phase angle.

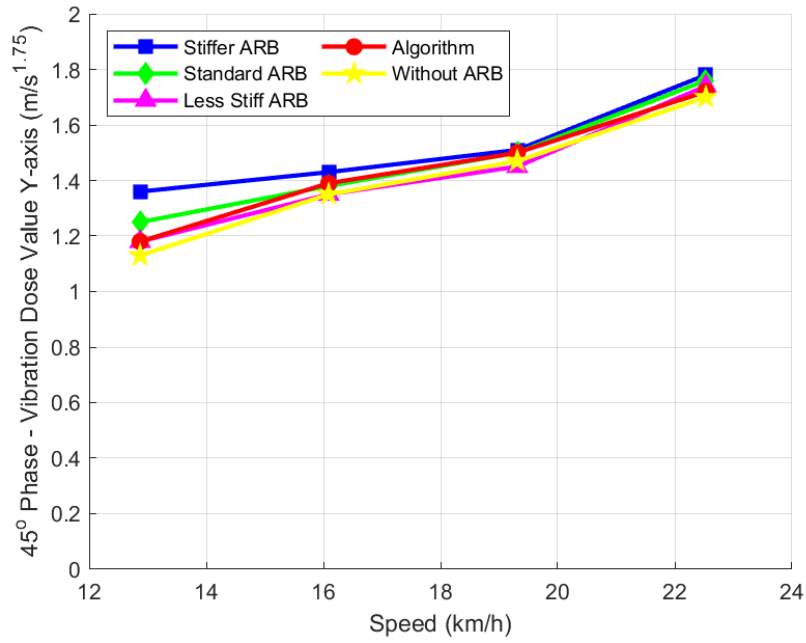


Figure 4.35 : The driver seat base's vibration dose values as a function of vehicle speed in the y-axis at the 45° phase angle.

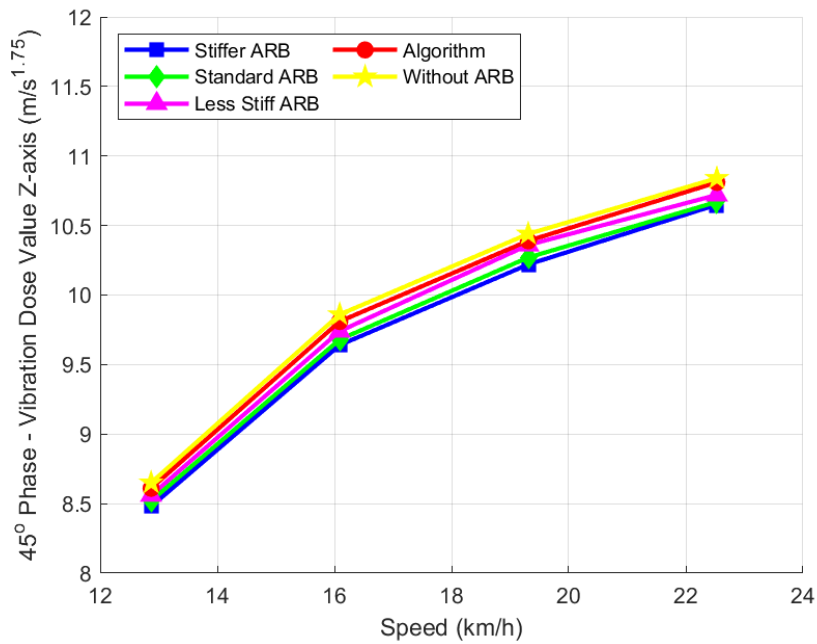


Figure 4.36 : The driver seat base's vibration dose values as a function of vehicle speed in the z-axis at the 45° phase angle.

Differences can be seen when the total vibration dose values are examined at the 45-degree phase angle. With the increase in speed, the rms values increase, negatively affecting the ride quality.

When compared with the 0-degree phase angle, it can be observed that the differences are small. Although the difference between these phase angles is meager, it is observed that the ride quality is better at the 45-degree phase angle up to 20 km/h speed, but at the values above it, vibration dose values are almost the same for the ride. In this phase angle, the anti-roll bar has quite a positive influence on ride quality.

The resulting values are in a similar order to the absorbed power and weighted rms values. While the anti-roll bar has a positive effect on the vertical axis, albeit at deficient levels, it can be seen that it has a negative effect on the lateral axis.

The active system approaches the model without the anti-roll bar regarding vibration dose values at this phase angle. Although the conditions shown in Figure 3.5 are occasionally offered, they are seldom provided throughout the simulation.

4.1.3.3 90° phase angle

In this part, the comparison of the values according to the 0 and 45-degree phase angles is also made. The total vibration dose values at the 90-degree phase angle are shown in Figure 4.37, 4.38, 4.39, and 4.40.

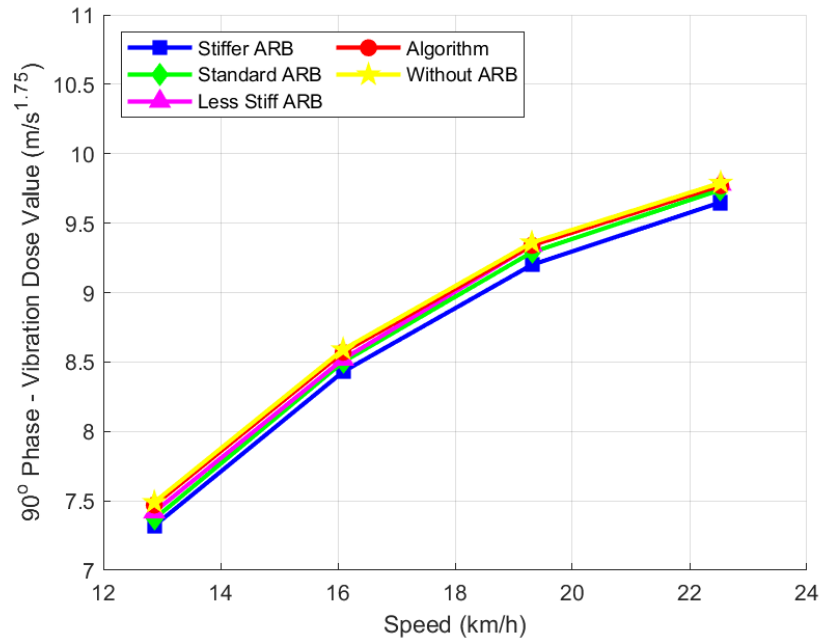


Figure 4.37 : Vibration dose values of driver seat base as a combination of all axes as a function of vehicle speed at 90° phase angle.

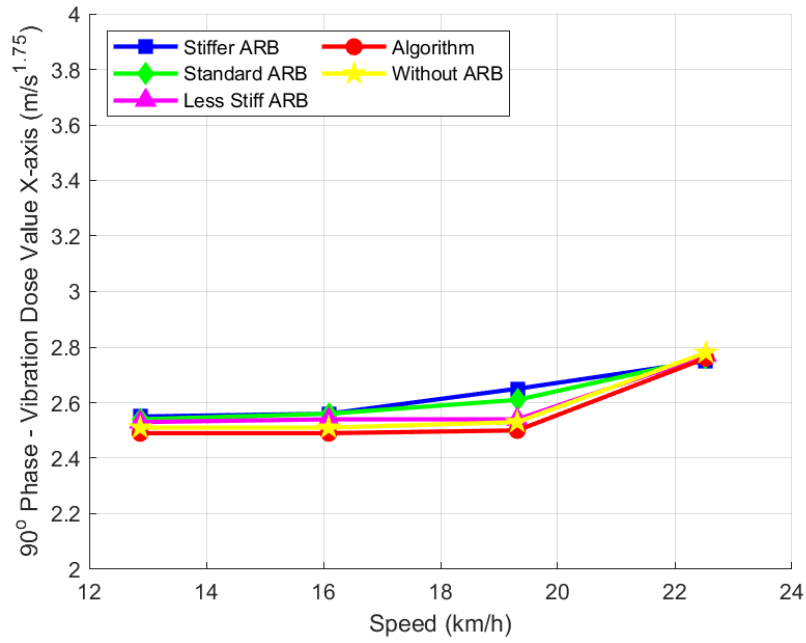


Figure 4.38 : The driver seat base's vibration dose values as a function of vehicle speed in the x-axis at the 90° phase angle.

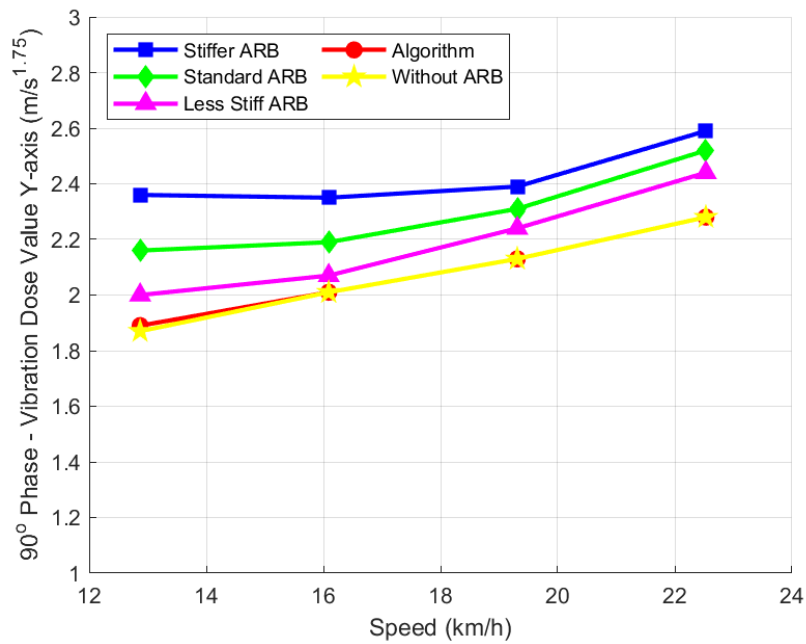


Figure 4.39 : The driver seat base's vibration dose values as a function of vehicle speed in the y-axis at the 90° phase angle.

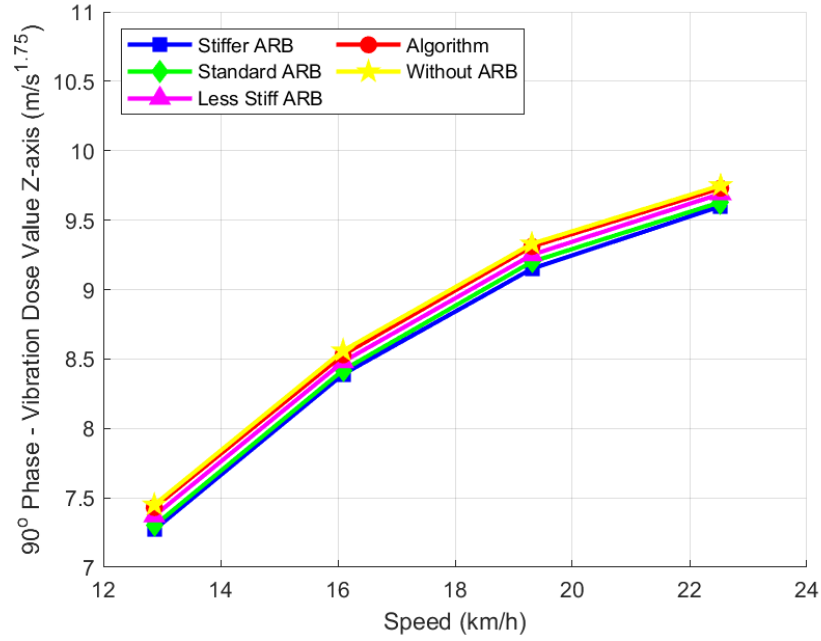


Figure 4.40 : The driver seat base's vibration dose values as a function of vehicle speed in the z-axis at the 90° phase angle.

When the total vibration dose values are examined at the 90-degree phase angle, it is observed that the differences are not apparent compared to absorbed power and frequency-weighted rms methods. With the increase in speed, the rms values increase like other phase angle analyses, which is negative for the ride quality.

The differences are more pronounced when compared with 0 and 45-degree phase angles. There is an approximate 15% decrease in these values obtained at the 90-degree phase angle compared to the 0-degree phase angle. In this phase angle, the effect of the anti-roll bar on the ride quality is not much for vibration dose value measurements.

The resulting values are in a similar order to the absorbed power and weighted rms values. The negative effect of the anti-roll bar can be seen in the longitudinal and lateral axis. Especially its negative effect on the lateral axis became more pronounced than at lower phase angles.

The active system's behavior is nearly identical to the type without an anti-roll bar at this phase angle. This is due to the continual fulfillment of the criteria listed in Figure 3.5.

4.1.3.4 180° phase angle

In this part, the comparison of the values according to the 0, 45 and 90-degree phase angles is also made. The total vibration dose values at the 180-degree phase angle are shown in Figure 4.41, 4.42, 4.43, and 4.44.

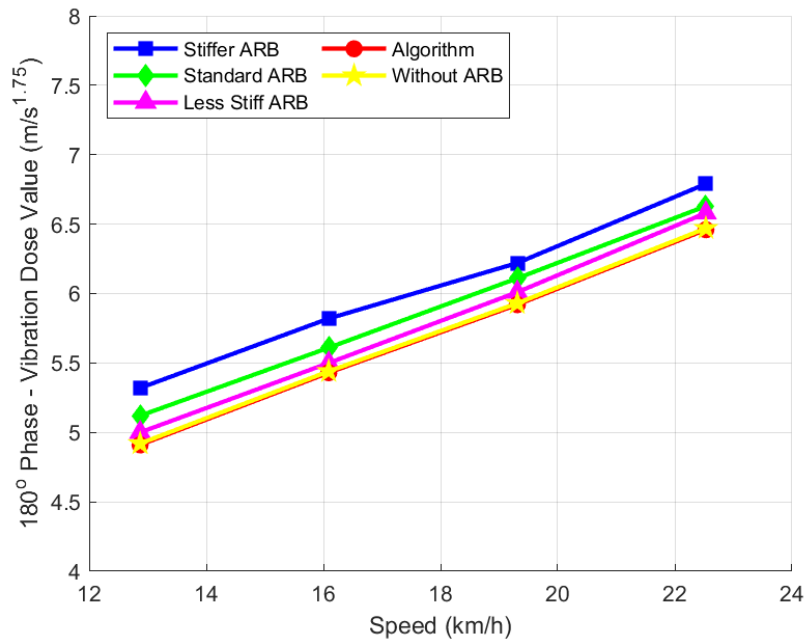


Figure 4.41 : Vibration dose values of driver seat base as a combination of all axes as a function of vehicle speed at 180° phase angle.

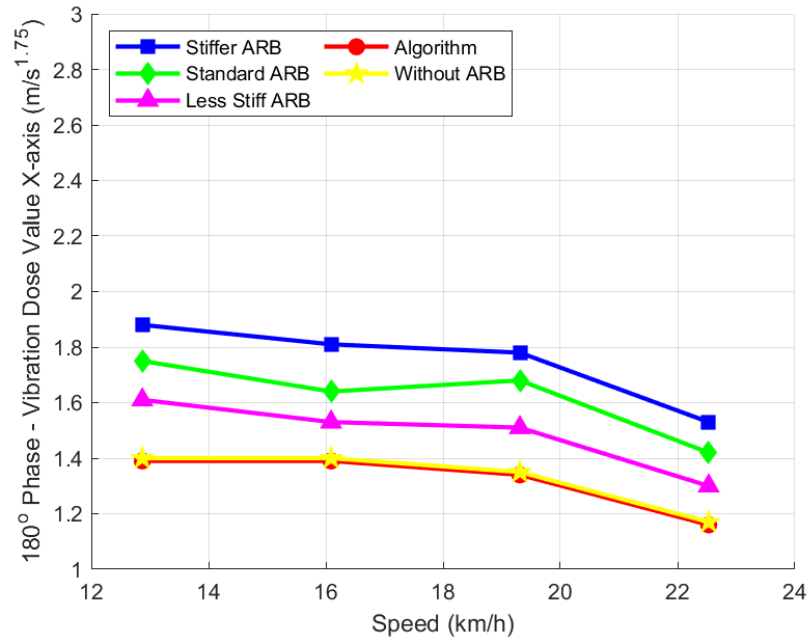


Figure 4.42 : The driver seat base's vibration dose values as a function of vehicle speed in the x-axis at the 180° phase angle.

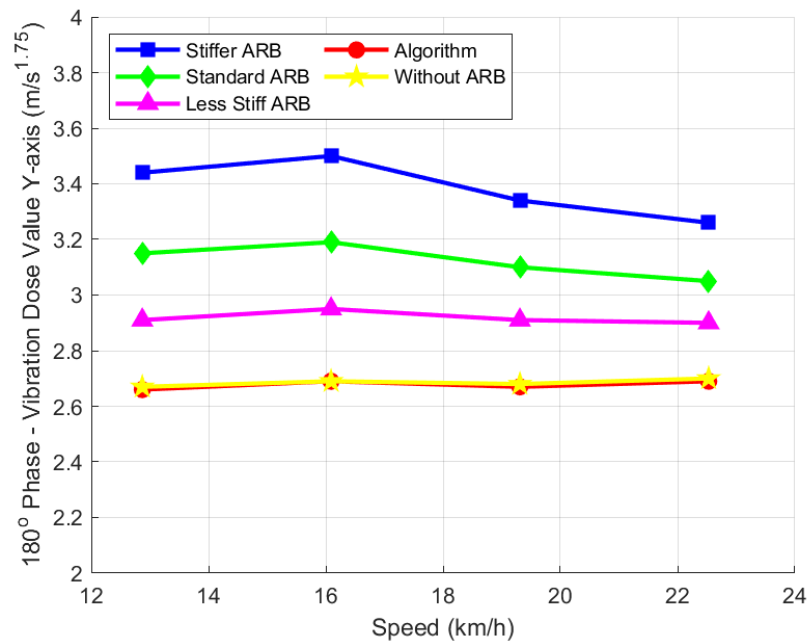


Figure 4.43 : The driver seat base's vibration dose values as a function of vehicle speed in the y-axis at the 180° phase angle.

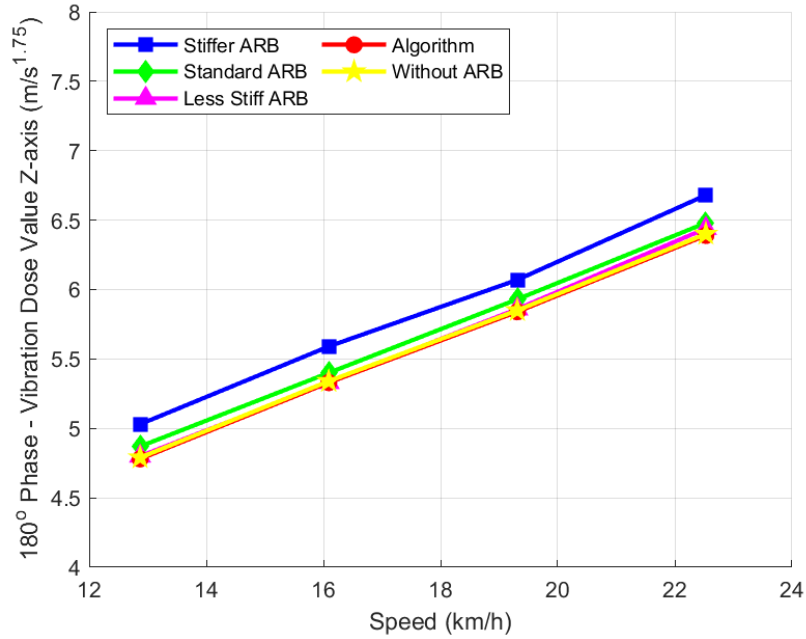


Figure 4.44 : The driver seat base's vibration dose values as a function of vehicle speed in the z-axis at the 180° phase angle.

The differences are evident when the total vibration dose values are examined at the 180-degree phase angle. With increased speed, the rms values, such as 0, 45, and 90-degree phase angles, adversely affect the ride quality.

The differences can be clearly observed when compared with 0, 45, and 90-degree phase angles. There is a decrease of approximately 35% in these values obtained at the 180-degree phase angle compared to the 0-degree phase angle. In this phase angle, the effect of the anti-roll bar on the ride quality is identical to the increase in the phase angle compared to the passive systems.

The resulting values are in a similar order to the absorbed power and weighted rms values. It can be seen that the anti-roll bar has a direct effect on every axis. It is observed that there is a very high negative effect on the ride quality on each axis.

At this phase angle, the active system acts like a vehicle without an anti-roll bar in terms of vibration dose values. This is because the criteria specified in Figure 3.5 are constantly met. It can be observed that the vehicle with an active anti-roll bar has the same values on each axis as the vehicle without an anti-roll bar. With this behavior of the active anti-roll bar system, a significant improvement in ride comfort has been observed compared to the passive anti-roll bar system.

4.2 ISO 3888-2 Double-lane Change Analysis Results

Deterioration of the vehicle's stability, going off the track, and touching the cone are among the reasons that cause the test to be considered invalid. In this study, contact with the cone was examined as the primary criterion, but it was checked whether other conditions were met.

When the specified conditions are evaluated, the final speed values of the vehicles before hitting any cone are as in Table 4.5.

Table 4.5 : Analysis results of double-lane change maneuvers.

Vehicle	Speed without touching a cone (km/h)
Vehicle without an anti-roll bar	54
Vehicle with a standard anti-roll bar	60
Vehicle with an active anti-roll bar	60

As can be deduced from Table 4.5, the model track without an anti-roll bar was completed at a much lower speed than the vehicle models with standard and active anti-roll bars. For this reason, the analysis of the vehicle without an anti-roll bar at the final speed values that the vehicle model with active and standard anti-roll bar can complete has not been performed. The active anti-roll bar system acted as a passive system in this analysis.

Scenario information is given in the previous chapter.

4.2.1 Vehicle speed of 54 km/h

The roll angle, lateral acceleration, trajectory, and front and rear vertical wheel forces for each model included in the analysis performed at 54 km/h are shown in Figures 4.45, 4.46, 4.47, 4.48, and 4.49.

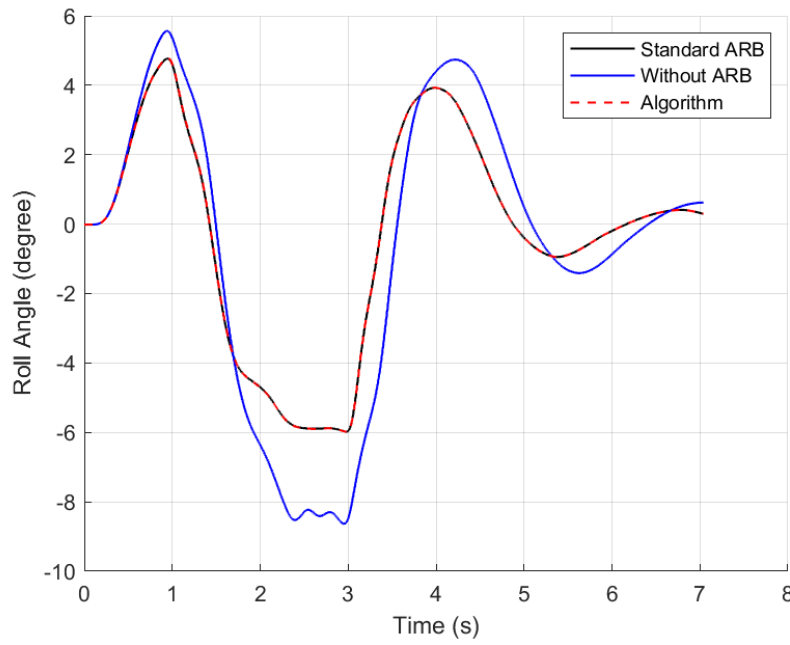


Figure 4.45 : Roll angle results in ISO 3888-2 double-lane change maneuver for 54 km/h speed.

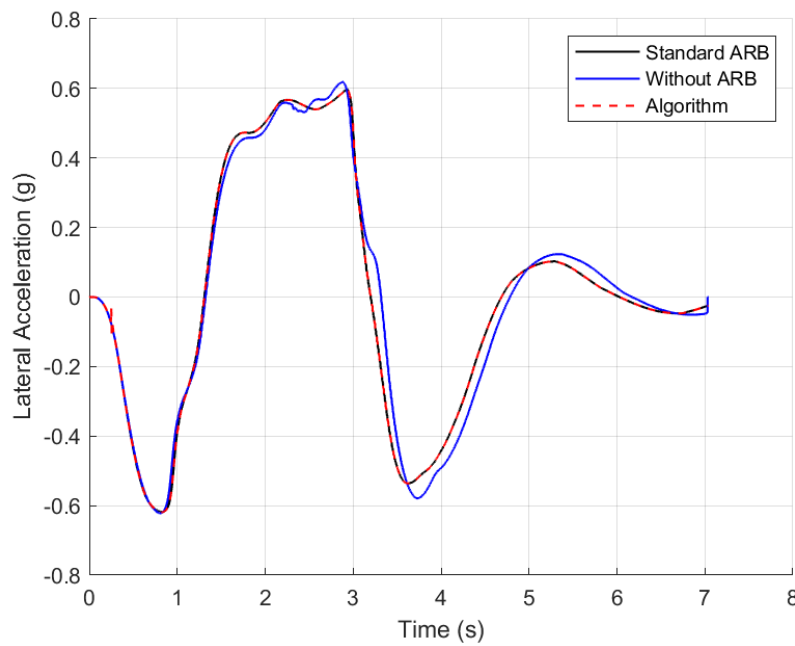


Figure 4.46 : Lateral acceleration results in ISO 3888-2 double-lane change maneuver for 54 km/h speed.

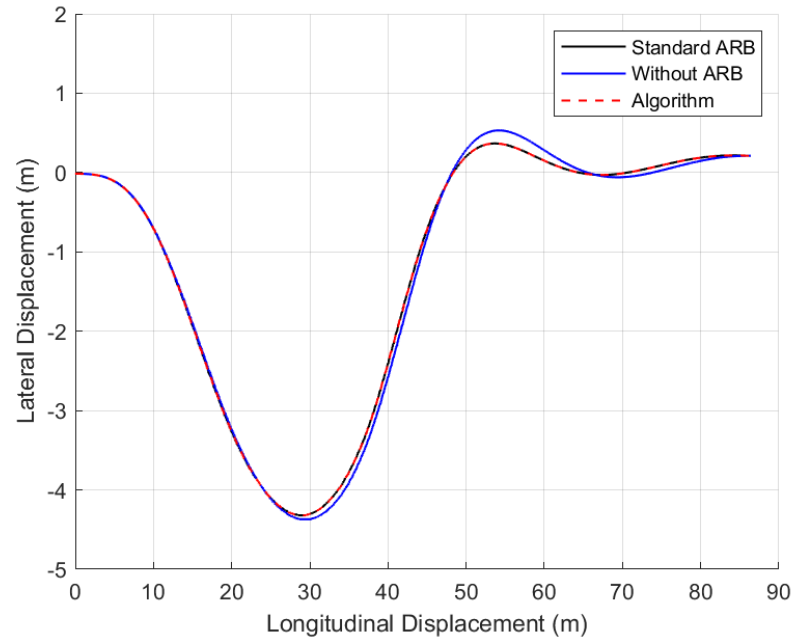


Figure 4.47 : The trajectory of vehicles in ISO 3888-2 double-lane change maneuver for 54 km/h speed.

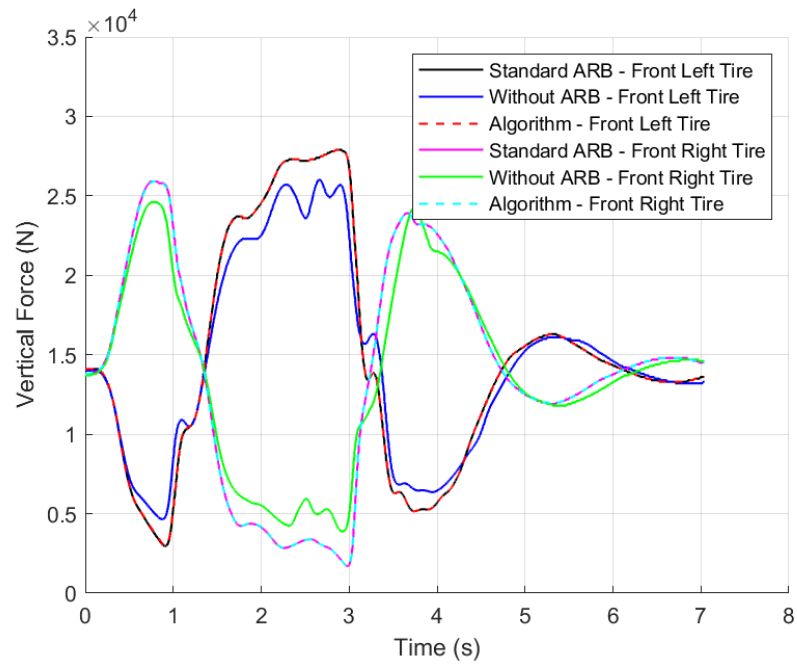


Figure 4.48 : Front tires vertical load in ISO 3888-2 double-lane change maneuver for 54 km/h speed.

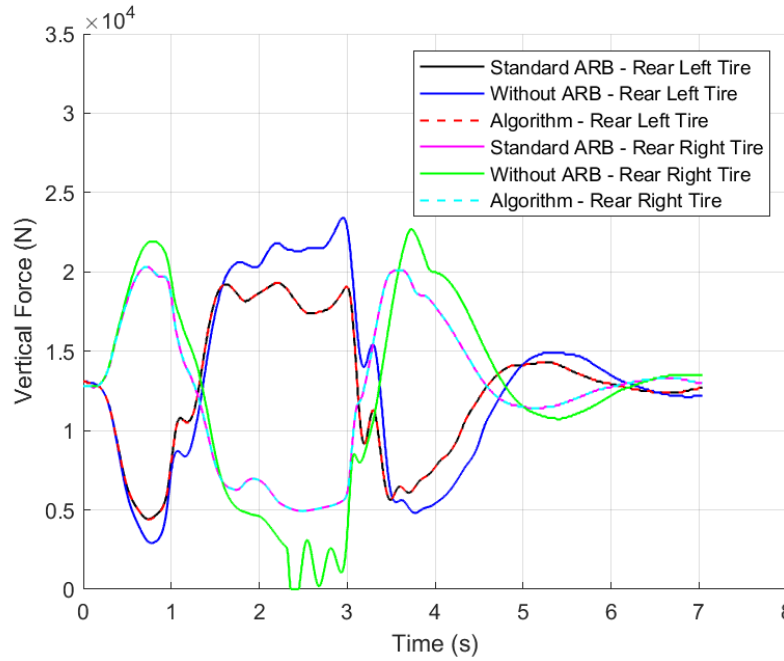


Figure 4.49 : Rear tires vertical load in ISO 3888-2 double-lane change maneuver for 54 km/h speed.

According to the results obtained at 54 km/h, the model without an anti-roll bar has higher roll angle and lateral acceleration values than the models with active and standard anti-roll bars. In addition to the driver negatively receiving this situation, it causes very unsafe driving. The model with the active anti-roll bar shows the same values as the vehicle model with a standard anti-roll bar.

When we look at the trajectory of the vehicles, it is seen that the model without an anti-roll bar goes out more than the other two models and therefore touches the cone.

When the front wheel forces are examined, as expected, the load transfer is higher in the models with the anti-roll bar at the front. This is because the anti-roll bar provides extra load transfer at the front. The situation is the opposite for the rear wheel vertical forces compared to the front. Using an anti-roll bar at the front caused the load transfer to shift from the rear to the front wheels.

One of the essential points is that the right rear wheel is out of contact with the ground during the analysis. This causes the test to be invalid.

As a result, with the active anti-roll bar model, the vehicle could stay at the required level in the corner, and the result is positive.

4.2.2 Vehicle speed of 60 km/h

The roll angle, lateral acceleration, trajectory, and front and rear vertical wheel forces for the vehicles at 60 km/h are shown in Figures 4.50, 4.51, 4.52, 4.53, and 4.54.

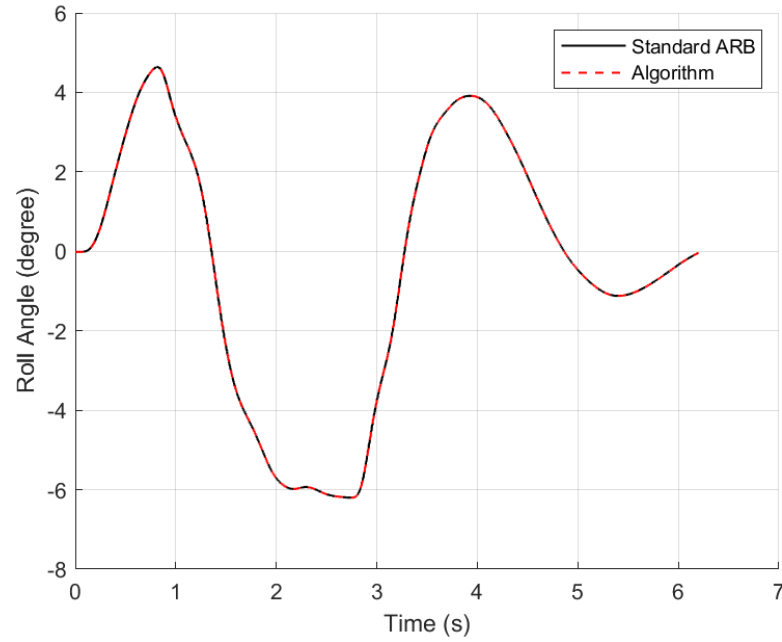


Figure 4.50 : Roll angle results in ISO 3888-2 double-lane change maneuver for 60 km/h speed.

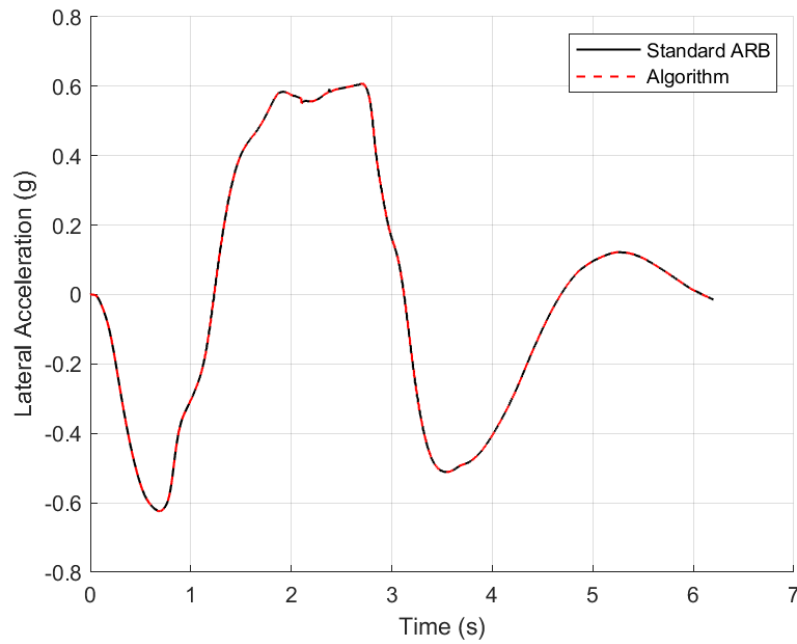


Figure 4.51 : Lateral acceleration results in ISO 3888-2 double-lane change maneuver for 60 km/h speed.

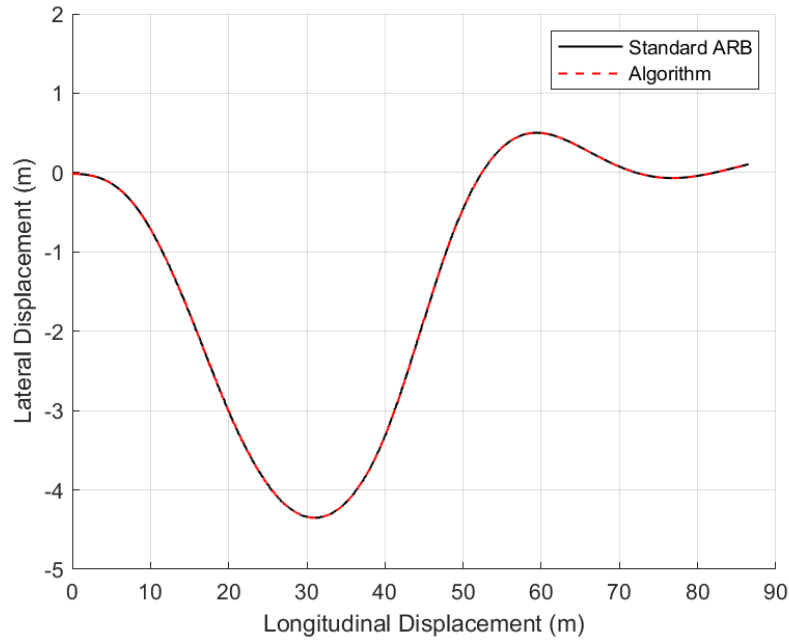


Figure 4.52 : The trajectory of vehicles in ISO 3888-2 double-lane change maneuver for 60 km/h speed.

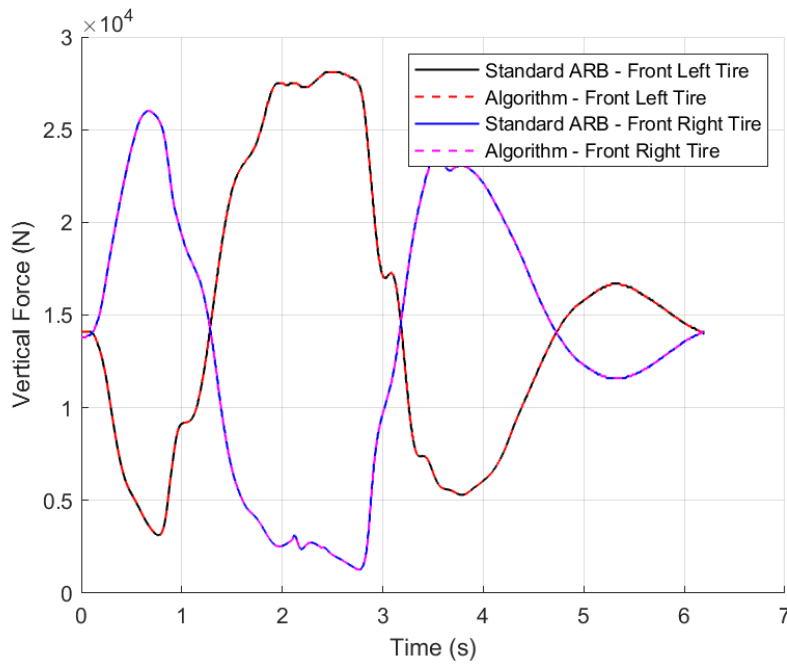


Figure 4.53 : Front tires vertical load in ISO 3888-2 double-lane change maneuver for 60 km/h speed.

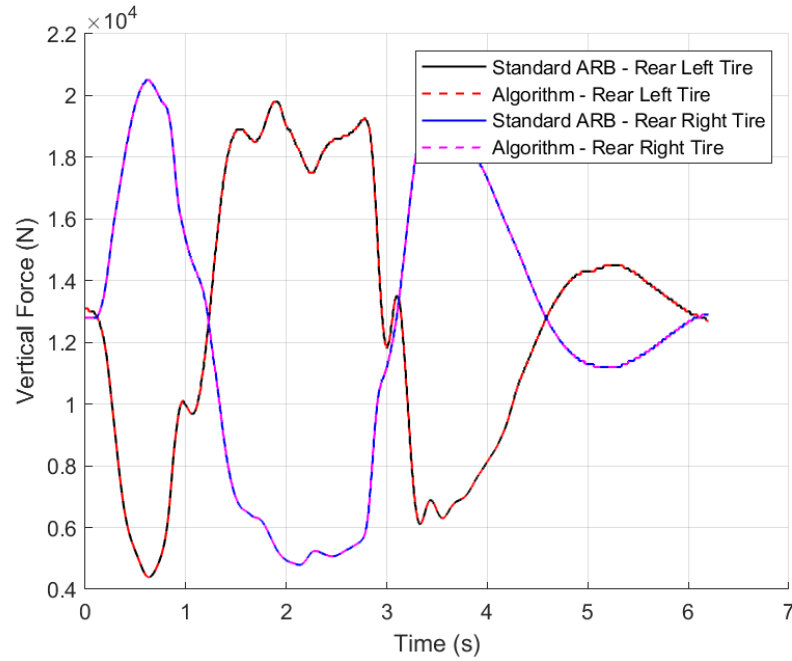


Figure 4.54 : Rear tires vertical load in ISO 3888-2 double-lane change maneuver for 60 km/h speed.

The last speed value vehicle models with active and standard anti-roll bars can pass without touching the pyloon is 60 km/h. When examining this speed value, it is seen that the roll angle exceeds 6 degrees. While this value is observed in the roll angle, the lateral acceleration value is around 0.6 g. Considering the two situations together, it can be determined that the safety status of the vehicle is relatively low.

Looking at the wheel forces of the vehicle, it is seen that the lateral load transfer at the front is higher than at the rear, as in the 54 km/h scenario.

It has better safety even when compared to the vehicle without an anti-roll bar at the lower speed. Also, although the roll angle is lower at a higher speed, the wheel does not lift off the ground.

It is seen that the values of the active anti-roll bar are the same as the vehicle with the standard anti-roll bar.

Analyzing the results, it can be observed that the active anti-roll bar performs the required performance in this case and is in perfect condition in terms of performance.

4.3 Fishhook Maneuver Analysis Results

The roll angle, lateral acceleration, trajectory, and front and rear vertical wheel forces for each vehicle included in the analysis performed are shown in Figures 4.55, 4.56, 4.57, 4.58, and 4.59. Scenario information is given in the previous chapter.

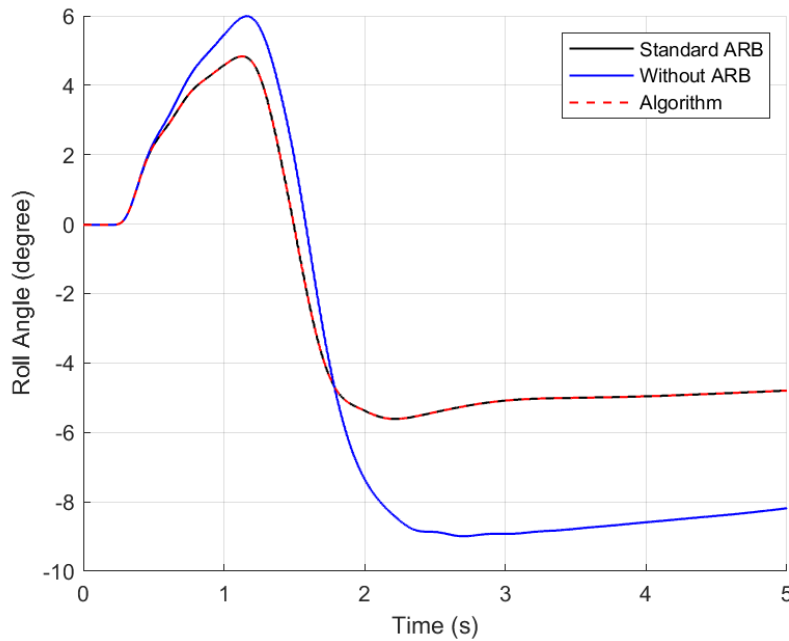


Figure 4.55 : Roll angle results in fishhook maneuver.

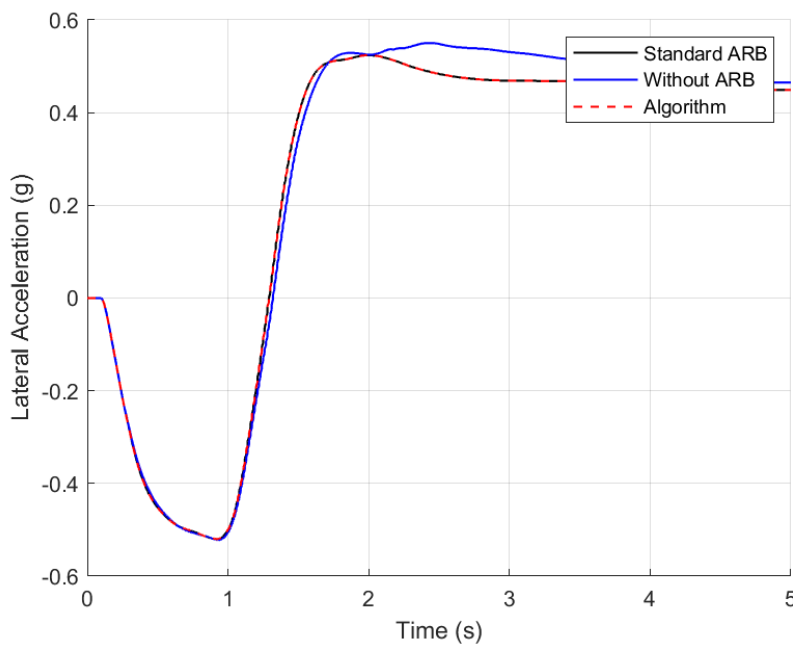


Figure 4.56 : Lateral acceleration results in fishhook maneuver.

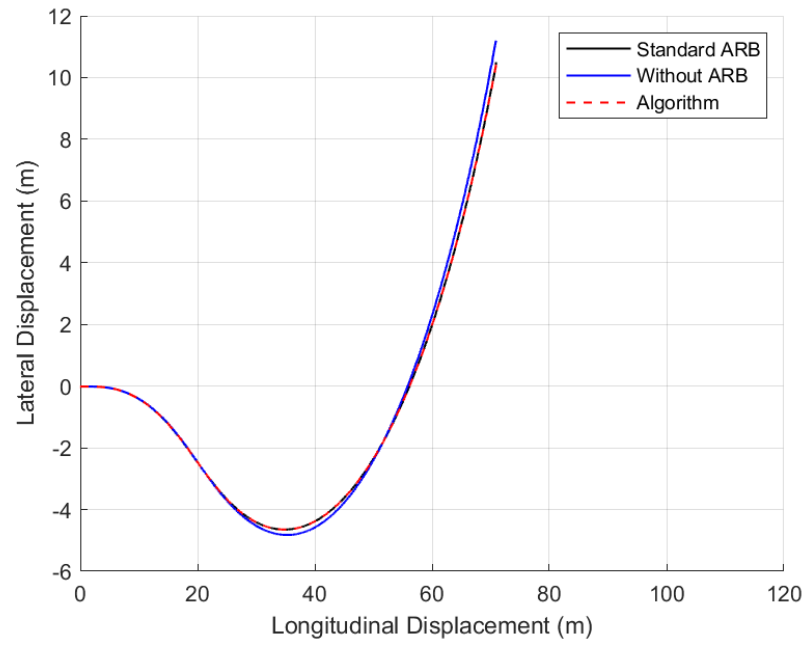


Figure 4.57 : The trajectory of vehicles in fishhook maneuver.

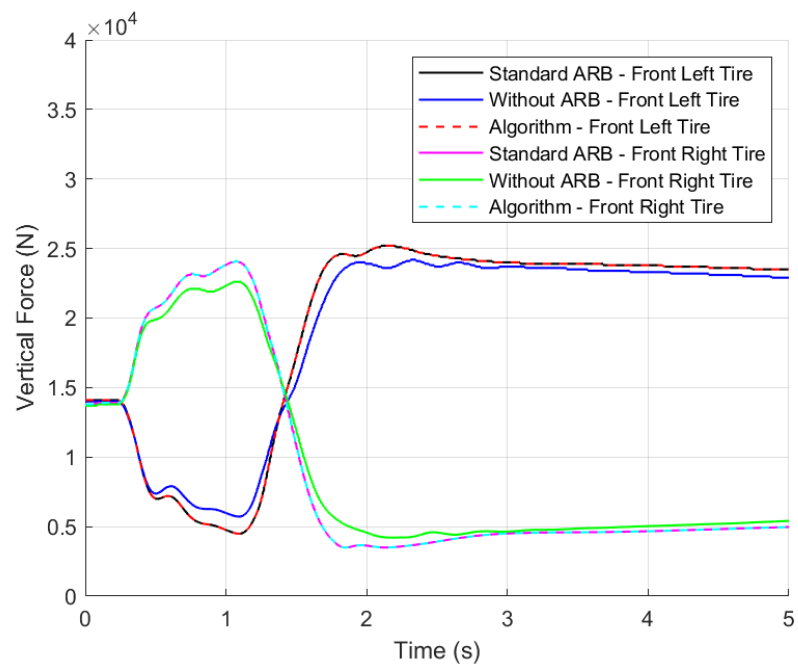


Figure 4.58 : Front tires vertical load in fishhook maneuver.

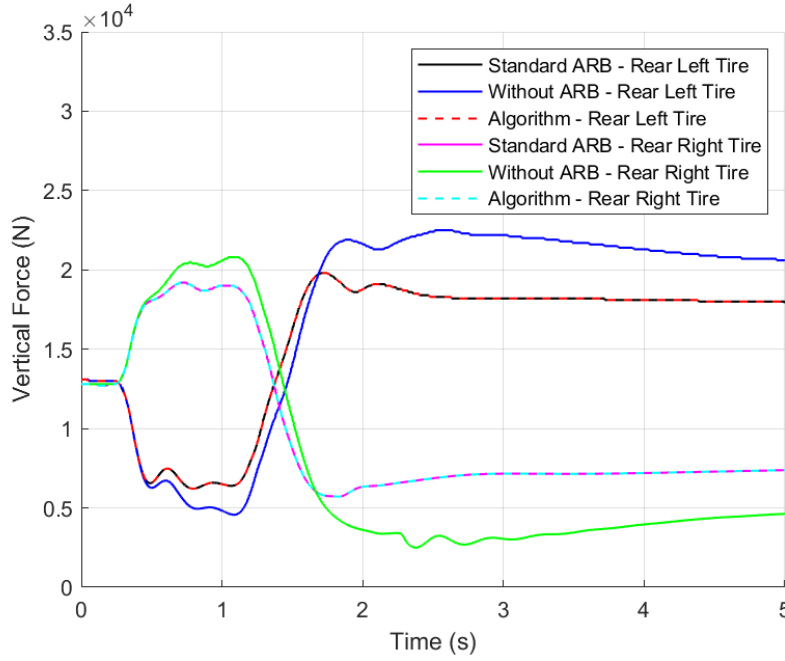


Figure 4.59 : Rear tires vertical load in fishhook maneuver.

In the Fishhook maneuver, the highest roll angle obtained in vehicles with standard and active anti-roll bars is 5.5 degrees, while this value is 9 degrees in vehicles without anti-roll bars. In addition, when the highest values are compared regarding lateral acceleration, the vehicle without an anti-roll bar witnessed approximately 5% higher value than the others. Considering two situations, systems with standard and active anti-roll bars are better in terms of safety and stability.

On the other hand, when looking at the vehicle's trajectory in the longitudinal and lateral axis, it is seen that the vehicle without an anti-roll bar exhibits an oversteer feature compared to the others. Considering that the speeds are the same and the steering inputs are fixed, it can be interpreted that the vehicle without an anti-roll bar exhibits a more oversteer behavior.

When the lateral load transfers are compared, it is observed that the load transfer shifts towards the front axle with an anti-roll bar at the front.

As a result, systems with active anti-roll bars showed the same performance as systems with standard anti-roll bars, which led to positive results for the vehicle.

4.4 J-turn Maneuver Analysis Results

The roll angle, lateral acceleration, trajectory, and front and rear vertical wheel forces for each vehicle included in the analysis performed are shown in Figures 4.60, 4.61, 4.62, 4.63, and 4.64. Scenario information is given in the previous chapter.

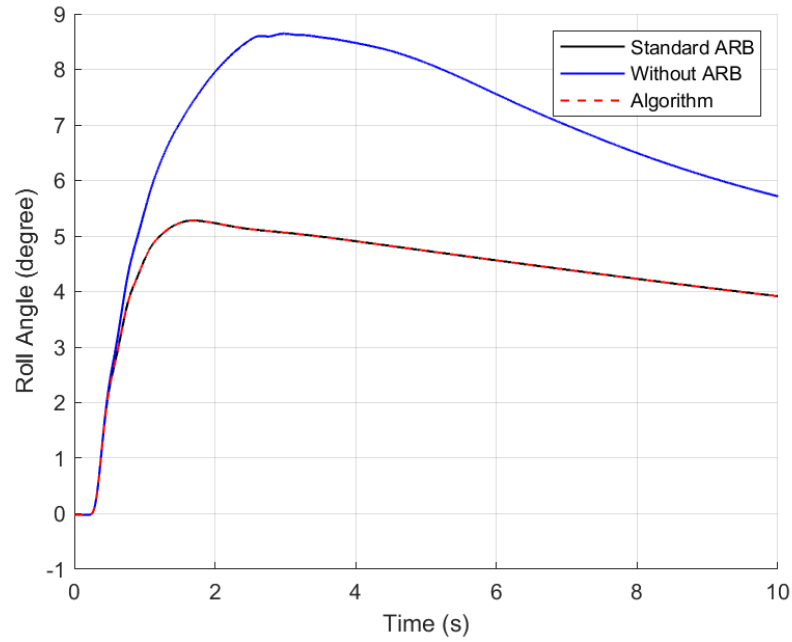


Figure 4.60 : Roll angle results in J-turn maneuver.

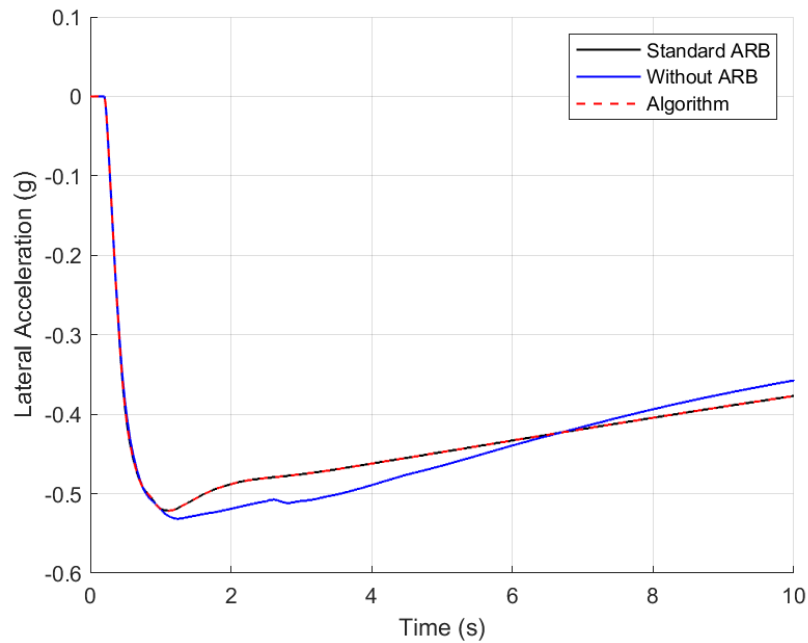


Figure 4.61 : Lateral acceleration results in J-turn maneuver.

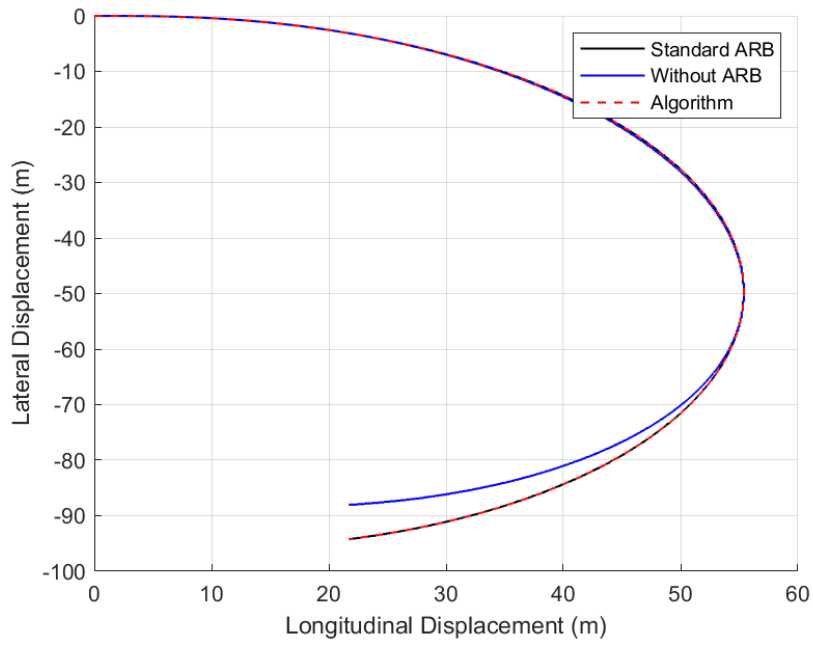


Figure 4.62 : The trajectory of vehicles in J-turn maneuver.

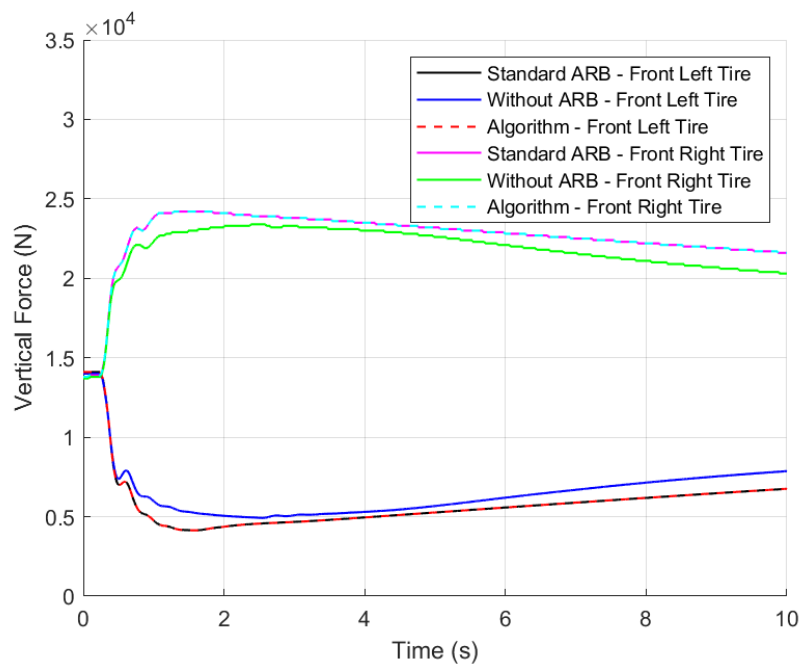


Figure 4.63 : Front tires vertical load in J-turn maneuver.

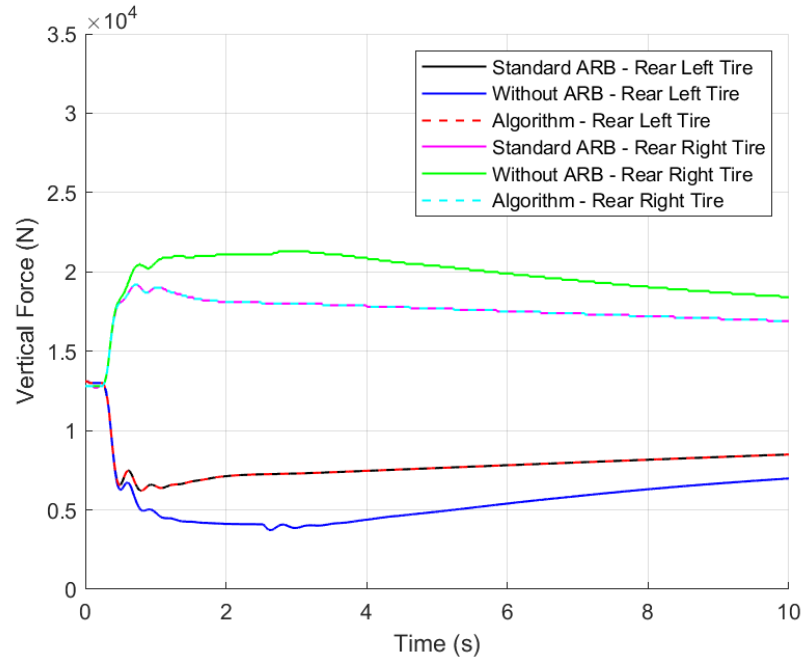


Figure 4.64 : Rear tires vertical load in J-turn maneuver.

In the J-turn maneuver, the highest roll angle obtained in vehicles with standard and active anti-roll bars is around 5 degrees, while this value is around 8.5 degrees in vehicles without anti-roll bars. In terms of lateral acceleration, when the highest values are compared, it is seen that the vehicle without an anti-roll bar has a higher value. Considering the two situations, systems with standard and active anti-roll bars are more reliable in terms of safety and stability.

On the other hand, looking at the vehicle's trajectory, it was determined that the vehicle without an anti-roll bar, as in the fishhook maneuver, showed a more oversteer behavior.

When the lateral load transfers are compared, the load transfer has shifted towards the front axle with an anti-roll bar at the front.

When the J-turn maneuver and the fishhook maneuver are compared, it is seen that the determined fishhook maneuver produces more significant results.

As a result, systems with active anti-roll bars showed the same performance as systems with standard anti-roll bars, which led to positive results for the vehicle.

4.5 Single-lane Change Maneuver Analysis Results

The roll angle, lateral acceleration, trajectory, and front and rear vertical wheel forces for each vehicle included in the analysis performed are shown in Figures 4.65, 4.66, 4.67, 4.68, and 4.69. Scenario information is given in the previous chapter.

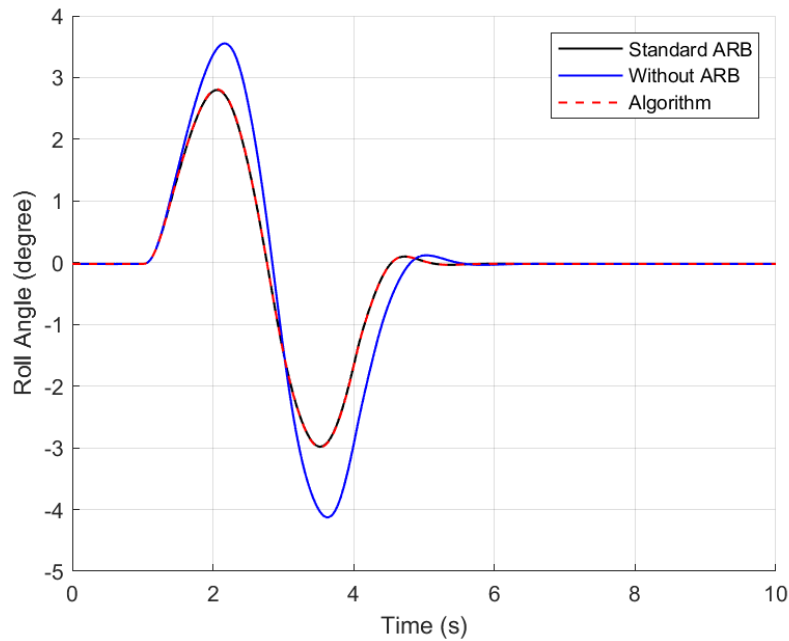


Figure 4.65 : Roll angle results in single-lane change maneuver.

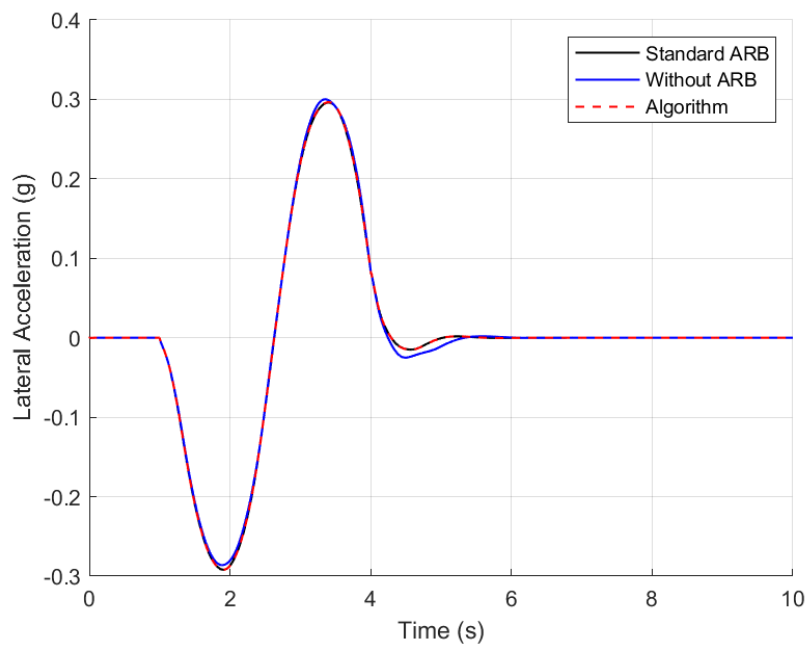


Figure 4.66 : Lateral acceleration results in single-lane change maneuver.

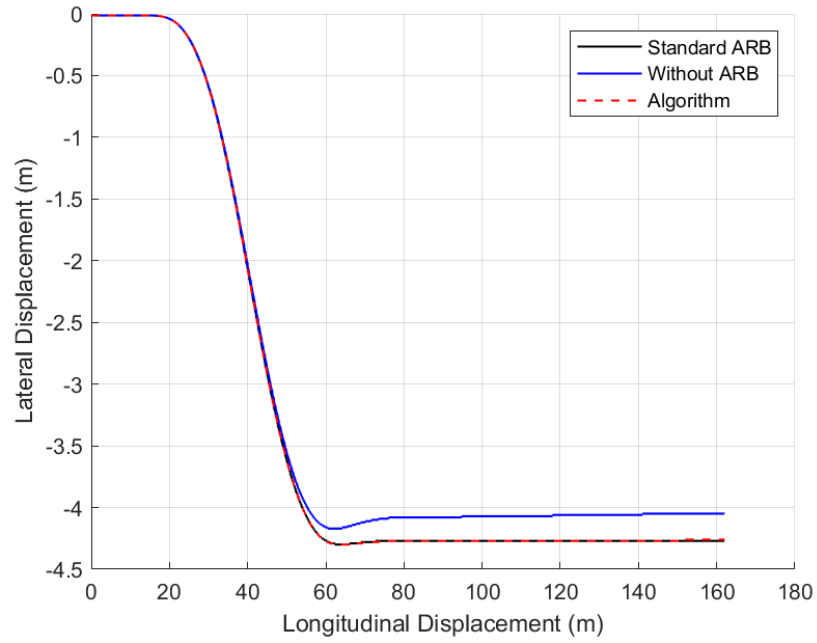


Figure 4.67 : The trajectory of vehicles in single-lane change maneuver.

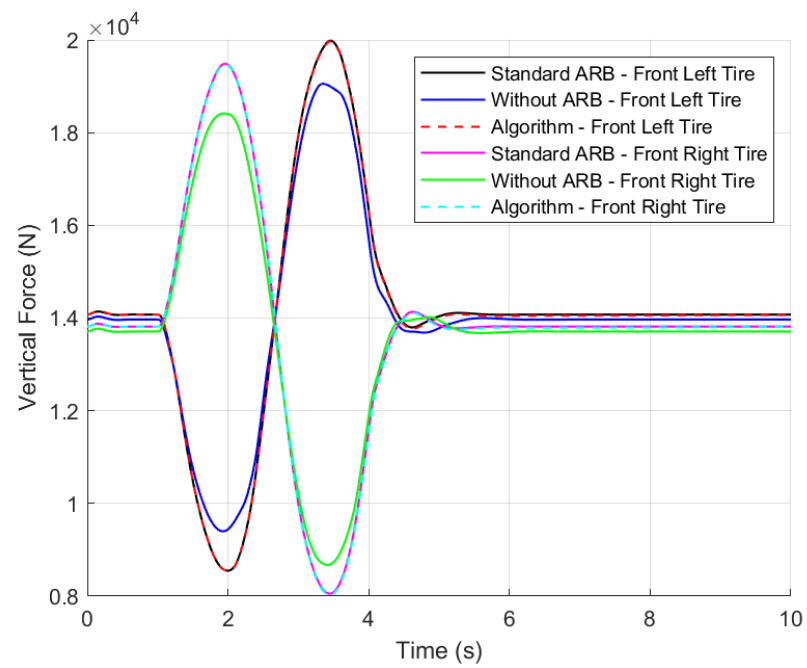


Figure 4.68 : Front tires vertical load in single-lane change maneuver.

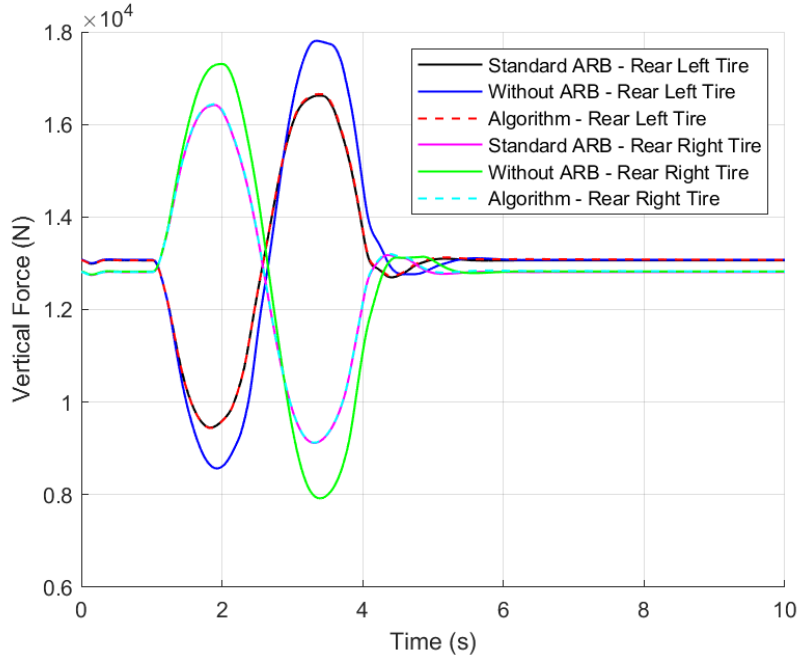


Figure 4.69 : Rear tires vertical load in single-lane change maneuver.

In the single-lane change maneuver, the roll angle values obtained in vehicles with standard and active anti-roll bars are approximately 35% lower than in the system without anti-roll bars. There was no noticeable difference in lateral acceleration values. Regarding roll angle, vehicles with standard and active anti-roll bars are much more reliable.

On the other hand, looking at the vehicle's trajectory, the vehicle without an anti-roll bar showed more oversteer behavior as in other analysis types. As a result, the position where it enters the lane at the same inputs is approximately 0.25 m closer to the beginning line on the lateral axis. However, it can be seen that the vehicle without an anti-roll bar is in a worse condition in terms of stability observed while entering the lane.

When the lateral load transfers are compared, it has been observed that the load transfer shifts from the rear axle to the front axle, with the vehicle having an anti-roll bar at the front.

As a result, systems with the active anti-roll bar showed the same performance as systems with standard anti-roll bars. The roll angle has been an essential parameter in the single-lane change analysis. It can be interpreted that vehicles with standard and active anti-roll bars are more reliable.

4.6 Constant Radius Cornering Analysis Results

The roll angle, lateral acceleration, and front and rear lateral load transfer forces for each vehicle included in the analysis performed are shown in Tables 4.6, 4.7, and 4.8. Scenario information is given in the previous chapter.

Table 4.6 : Constant radius cornering results of the vehicle with the standard anti-roll bar.

Lateral acceleration (g)	Roll angle (deg)	Lateral load transfer in front (N)	Lateral load transfer in rear (N)
0.2	2.01	7596	4400
0.3	3.05	11608	6737
0.4	4.11	15558	8998
0.5	5.30	19920	11438

Table 4.7 : Constant radius cornering results of the vehicle without an anti-roll bar.

Lateral acceleration (g)	Roll angle (deg)	Lateral load transfer in front (N)	Lateral load transfer in rear (N)
0.2	2.70	6397	6008
0.3	4.29	9935	9418
0.4	6.21	13958	13193
0.5	8.48	18287	17718

Table 4.8 : Constant radius cornering results of the vehicle with the active anti-roll bar.

Lateral acceleration (g)	Roll angle (deg)	Lateral load transfer in front (N)	Lateral load transfer in rear (N)
0.2	2.01	7601	4405
0.3	3.05	11616	6743
0.4	4.11	15567	9005
0.5	5.30	19920	11438

When the tables are examined, it is seen that the vehicle with the standard anti-roll bar has a lower roll angle than the vehicle without the anti-roll bar at the same acceleration values. With the increase in lateral acceleration, the percentage difference between them also increases. The conclusion to be drawn from this is that the vehicle without an anti-roll bar is unsafe at high lateral accelerations.

When the lateral load transfers are also examined, it can be seen that the front load transfer of the vehicle with the anti-roll bar is higher than the vehicle without the anti-roll bar.

As is well known as a basic calculation, if v^2/r exceeds the lateral acceleration (g), understeer will result from the vehicle's inability to produce enough lateral force to maintain its circular course. In contrast, the vehicle will be able to produce enough lateral force to maintain its circular course and will not experience understeer if v^2/r is smaller than the lateral acceleration. When 0.2 and 0.3 g, which have low lateral acceleration values, are examined, these values are 0.201 and 0.302 g, respectively, for vehicles with active and standard anti-roll bars. In comparison, these values are 0.193 and 0.296 g, respectively, for vehicles without anti-roll bars.

As a result, vehicles with active and standard anti-roll bars show understeer, while vehicles without anti-roll bars show oversteer. Considering that the understeer has more accessible solutions than the oversteer, it can be concluded that vehicles with active and standard anti-roll bars are also more reliable in this respect.

The values of vehicles with active and standard anti-roll bars are very much the same. The conclusion that can be drawn from this is that the active anti-roll bar system behaves like a standard vehicle during cornering and carries the good features of this system.

5. CONCLUSION AND RECOMMENDATIONS

5.1 Conclusion of Results

Regardless of the anti-roll bar setup, decreases were observed in the absorbed power values on the x-axis and z-axis with the increase of the phase angle in the same speed values. On the other hand, with the increase of the phase angle, the absorbed power values in the y-axis increased.

While the anti-roll bar has no significant effect on the ride quality at low phase angles, such as 0 and 45, it has been determined that the effect of the anti-roll bar on the ride quality is high at high phase angles, such as 90 and 180. At these high phase angles, the anti-roll bar negatively affects the ride quality, while the lateral axis is the axis where this effect is most evident. However, it can be observed that it also has a negative effect on the z-axis. In addition, when looking at the wheel loads, it can be observed that the lateral load transfer in the front suspension increases with the anti-roll bar. However, it is observed that the lateral load transfer shifts from the rear suspension to the front suspension compared to the models without an anti-roll bar.

On the other hand, in vehicles with the active anti-roll bar, it was observed that the vehicle's ride quality is close to that of the standard anti-roll bar at low phase angles. In contrast, the effect of the anti-roll bar is observed to be deactivated at high phase angles. In this way, it has been observed that the ride quality values are at the desired level at high phase angles. Especially at the 180-degree phase angle, all three ride quality assessment values support this result. These results also support that the driving quality worsens with increased speed.

Considering the frequency-weighted rms and vibration dose values, it supports the values obtained by absorbed power. According to these results, while the effect of an anti-roll bar in low phases is low, its effect on ride quality increases with phase increase. While it has a very negative effect on the lateral axis at high phase angles, it can be deduced that it has a negative effect on each axis according to the 180-degree phase angle results.

On the other hand, in handling analyzes, it has been observed that vehicles without anti-roll bars in the front suspension perform worse. Vehicles without anti-roll bar exhibit oversteer behavior compared to vehicles with an anti-roll bar, with higher roll angle and lateral acceleration values in each analysis type. In addition, the double lane change analysis determined that the vehicle model without the anti-roll bar had ground clearance at the right rear wheel at much lower speed values than the vehicle with the anti-roll bar. According to these results, an anti-roll bar in the front of this vehicle is essential for stability and safety. The results of vehicle models with the active anti-roll bar are the same as those with the standard anti-roll bar.

As a result, it has been observed that the vehicle model with the active anti-roll bar offers better ride quality conditions compared to the standard vehicle while not making any concessions on handling. While this positively affects the ride quality of the vehicle, it ensures that it does not experience any adverse effects and shows that the stated purpose has been achieved.

5.2 Practical Application of This Study

Practically, this system can be operated with this algorithm. However, due to this section's low required time values, the supply of microprocessors and sensors suitable for this system can be challenging. While choosing higher values in terms of the values obtained as a result of the analyzes and the input values given to the engine may cause the system to be triggered later and at a lower resolution, it is still possible to obtain results similar to the results obtained in the active system.

Establishing different systems as a mechanism may help to simplify the active system. Thus, while the requirements can be met with a more straightforward system, providing energy efficiency with lower energy consumption could be possible.

5.3 Future Recommendations

The analyzes and the algorithm are mostly valid for off-road vehicles. When energy consumption is essential in vehicles, such systems can be developed in this period, especially for passenger vehicles. This will have positive aspects for drivers to stay fit during the day. In addition, when people who have to travel long distances with their vehicles are considered, there will be positive returns in terms of health.

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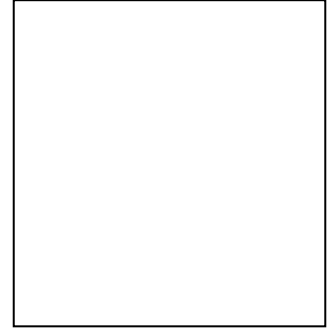
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