İSTANBUL TECHNICAL UNIVERSITY ★ INSTITUTE OF SCIENCE AND TECHNOLOGY

MODELLING AND ANALYSIS OF A HYDROPNEUMATIC SUSPENSION SYSTEM

M.Sc. Thesis by
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Department : Mechanical Engineering

Programme : Automotive

JANUARY 2011

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JANUARY 2011

İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

BİR HİDROPNÖMATİK SÜSPANSİYON SİSTEMİNİN MODELLENMESİ VE ANALİZİ

YÜKSEK LİSANS TEZİ

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Annem ve ailem için,

FOREWORD

This study has been intitated as a basis for understanding the hydropneumatic suspension concept, its basic characteristics and why it has importance for the automotive industry. Back to back comparions with the conventional suspension systems are presented and thus differences and advantages are discussed.

I would like to express my deep appreciation and thanks for my advisor Prof. Dr. Ahmet GÜNEY for his great support and guidance throughout not only preparation of this study, but also my student career. I also would like to thank my colleagues and executives at Ford OTOSAN A.Ş. for their understanding and support throughout my Master's study.

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Mechanical Engineer

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ABBREVIATIONS

D	**:** Damping ratio
c	**:** Damping coefficient
k	**:** Stiffness coefficient
P	**:** Pressure
V	**:** Volume
γ	**:** Index of hydraulic fluid

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MODELLING AND ANALYSIS OF A HYDROPNEUMATIC SUSPENSION SYSTEM

SUMMARY

Automotive industry has always been in cope with harshness. Customers complain both for comfort and safety, yet many of them are not aware that these two parameters can be a trade off for a vehicle. The harder the suspension is, the harsher the feel becomes bu the road stability may increase. The preferences of a vehicle's definition in terms of ride performance and comfort characteristics have always been a compromise due to the nature of relevant affected system components. Both the comfort and ride performance are trade-off for each other for mechanical suspensions. The will to overcome this compromise and build vehicles outperforming in both these tasks has started with the implementation of accumulators instead of conventional helical or leaf springs. This has enabled the engineers to tune the system without compromising for ride comfort; implementing additional components to overcome the ride performance degredation. Hydropneumatic suspensions are the checkpoint for this task, and they are currently being used for passenger vehicles, heavy duty vehicles and military vehicles. Citroen is the most notable user of hydropneumatic suspension systems on its end products. This thesis studies hydropneumatic suspension systems in terms of basic suspension and damping characteristics, and compares it to conventional mechanical suspension systems. Transfer function responses are generated and analysed when the system is exposed to either step inputs, or undulating perturbations. Each case is simulated for mechanical suspension, and the hydropneumatic suspension with different sub-components such as the orifice, the checkvalve and leveling servo valve. Main differences between the hydropneumatic suspension and its mechanical predecessor are tried to be understood, and commented. The first section is an insight to the suspension basics and tries to compare a mechanical susspension to the hydropneumatic suspension in terms of its basic damping and stifness characteristics. Also non functional parameters of a hydropneumatic suspension are compared to conventional suspension such as the cost and weight. Section two covers the springing medium, while the section three gives a bit more detail about the design of the hydraulic damping elements. Section four is the start of building an analogical model via one dimensional simulation software AMESim. Each element used in the set up is analyzed under this section. The complete models and their corresponding analysis results are presented under section five. Graphs outline the behaviour of the systems. The last section summarizes the achieved results, and proposes further study subjects about hydropneumatic suspensions to the researcher.

BİR HİDROPNÖMATİK SÜSPANSİYON SİSTEMİNİN MODELLENMESİ VE ANALİZİ

ÖZET

Süspansiyon konforsuzluğu, otomotiv endüstrisinin her zaman başa çıkmak zorunda olduğu bir konu olagelmiştir. Çoğu kullanıcı, süspansiyon konforu ile sürüş güvenliğinin birbirine ters düşebilecek konular olduğundan haberdar değildir. Süspansiyon sertleştikçe seyir konforu bozulurken taşıtın seyir emniyeti artabilmektedir. Bir taşıtın yol tutuşu, sürüş performansı ve konfor özellikleri, bu özellikleri etkileyen parametrelerin karakteri nedeniyle birbirine karşı taviz verilmesi gereken nitelikleri olmuştur. Özellikle mekanik süspansiyonlar için sürüş performansı ve konfor özellikleri birbirleri için ters yönlü çalışmaktadırlar. Her iki özelliği birbirinden taviz vermeden iyileştirmek adına yapılan çalışmalar, akümülatörlerin mekanik helezon veya makas yaylar yerine kullanılmasıyla başlamıştır. Mühendisler böylece sürüş performansından ve yol tutuşundan taviz vermeden konfor özelliklerini de iyileştirebilidikleri sistemler tasarlamaya başlayabilmişlerdir. Hidropnömatik süspansiyonlar, bu durum için bir dönüm noktasıdır ve günümüzde de birçok yolcu taşıtında, ağır taşıtlarda ve askeri taşıtlarda kullanılmaktadırlar. Citroen firması, hidropnömatik süspansiyonları satışa sunduğu araçlarda kullanmasıyla bilinmektedir. Bu tez çalışmasında hidropnömatik süspansiyonlar temel yay ve sönüm özellikleri bakımından incelenmiştir ve klasik mekanik süspansiyonlarla karşılaştırılmıştır. Sistemin basamak ve salınımlı uyarılara verdiği tepkiler transfer fonksiyonları oluşturularak incelenmiştir. Modellenen her farklı sönüm ve seviye ayarı sağlayıcı bileşen için mekanik süspansiyonla karşılaştırma yoluna gidilmiştir. Mekanik süspansiyon ve hidropnömatik süspansiyon arasındaki temel farklar anlaşılmaya ve yorumlanmaya çalışılmıştır.

Birinci kısımda süspansiyonların temel özellikleri açıklanmış, hidropnömatik süspansiyonun konvansiyonel süspansiyonlar ile temel nitelik kıyaslamaları açıklamalı olarak yapılmıştır. Süspansiyon özelliklerinin yüke bağlı değişimleri irdelenmiştir. Seviye fonksiyonun temel nitelikleri belirtilmiş, hidropnömatik süspansiyonun konvansiyonel süspansiyonlara göre maliyet, ağırlık gibi niteliklerinin karşılaştırması sunulmuştur. Hidropnömatik süspansiyon sistemlerinin temel uygulama alanları tartışılmıştır.

İkinci kısımda hidropnömatik süspansiyon sistemlerinin genel yapısı ve fonsiyonelliği incelenmiştir. Sistemi oluşturan temel elemanlar sıralanmış, yay ve sönüm mekanizmaları irdelenmiş, yaylanma sönüm etkisini sağlayan parametreler göz önüne alınmıştır.

Üçüncü kısımda süspansiyona ait sönüm elemanlarının temel tasarımları sırasında göz önüne alınan hususlar nitelenmiş, farklı tipte sönüm elemanları hakkında bilgi verilmeye çalışılmıştır. Bu çalışmada sıcaklık etkisi göz önüne alınmış olmasa da bu bölümde sönümün sıcaklığa bağlı değişimi hakkında da açıklama yapılmıştır.

Dördüncü kısım, tek boyutlu simülasyon yazılımı AMESim ile oluşturulan analojik modellerin incelenmesine ayrılmıştır. Bu bölümde kullanılan yazılımın sağladığı elemanların teorik incelemeleri yapılmış, parametrelerin saptasında kriterler sunulmuştur.

Beşinci kısımda yapılan analizler ile ilgili sonuçlara yer verilmiş, farklı senaryolarda sistemin davranışı karşılaştırmalı olarak anlaşılmaya çalışılmıştır.

Altıncı ve son kısımda ise çalışmada elde edilen sonuçların bir özetine yer verilmiş, araştırmacıya hidropnömatik süspansiyonlar ile ilgili çalışılabilecek ileri seviyedeki konular hakkında öneriler getirilmiştir.

1. INTRODUCTION

Suspension systems are vital components of mechanical integration. They have functions assisting human comfort, safety, component fatigue and environmental isolation. Suspensions have a vast area of implementation, and thus have complex functional interactions while performing their function. Figure 1.1 (Bauer, 2008) describes the task interaction scheme for a vehicle suspension unit:

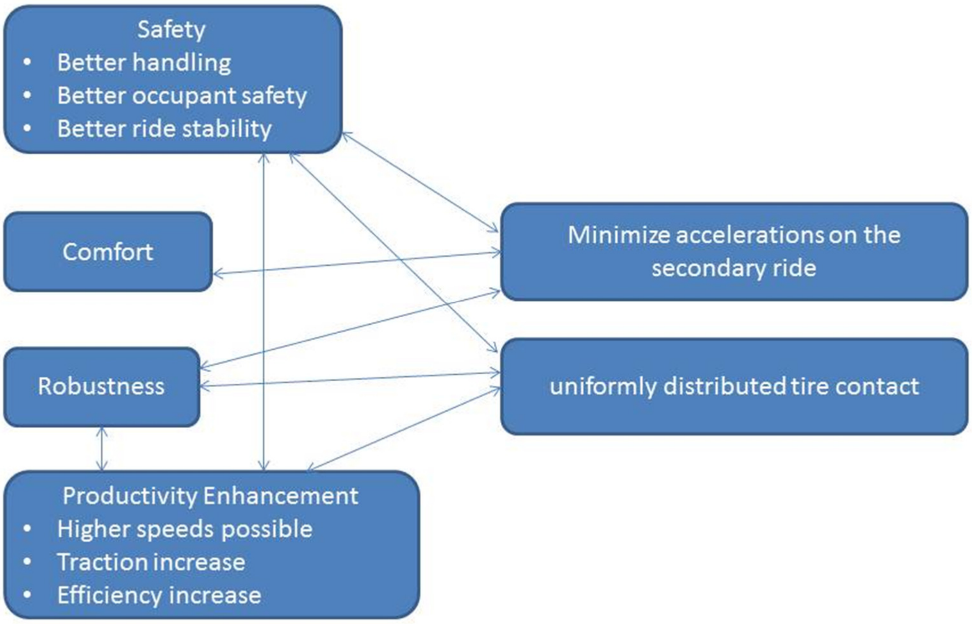

Figure 1.1 Task interaction scheme for a vehicle suspension.

A suspension system usually consists of a spring and a damper. The spring alone would already allow the decoupling of input side and isolated side just by its elastic properties and would compensate accelerations/displacements from the input side. Yet, due to the displacement, the spring would store energy and therefore the system would keep on oscillating permanently. Not only this, in case of further excitations with suitable frequency and phase, it would pick up further energy and the amplitude on the isolated side would increase even further (resonance). This is why a spring is almost always used in combination with a damper. The energy that has been temporarily stored in the spring is converted into heat by the damper and the amplitude of the oscillation therefore decays (Bauer, 2008).

There are different suspension concepts for different areas of usage. This thesis will be dealing with vehicle suspensions; specifically the "hydropneumatic suspension" concept. Hydropneumatic suspension concept was first implemented by the French engineer George Messier. In 1921, he developed a keen interest in pneumatic and oleopneumatic shock-absorbers. (SAFRAN Group, 2010). French car manufacturer Citroen is notably known for developing and commercially implementing hydropneumatic suspensions on its vehicles since 1950s.

1.1 The Hydro-Pneumatic Suspension Compared to Other Suspension Concepts

In a hydropneumatic suspension system, helical or leaf springs of a mechanical suspension are replaced by an accumulator; a metal sphere that consists of two chambers seperated by a thermoplastic polyurethane (TPU) membrane. One of the two chambers is filled with nitrogen gas; this gas has no transfer to the other side of the membrane and serves as the springing medium. The other chamber is connected to the hydraulic system of the suspension, and filled with a special hydraulic fluid (LHM - Liquide Hydraulique Minéral) which prevents water absorption, thus air bubbles dissolving within.

The main competing systems for hydropneumatic vehicle suspension systems are the pure-mechanical and pneumatic suspension. There are other concepts such as the suspension effect generated by a magnetic field but they are currently out of the scope for the passenger vehicle industry. In the following; the conventional mechanical, pneumatic and hydropneumatic suspensions with regard to the basic characteristics are compared with each other.

1.1.1 Suspension Characteristics

First, it would be assumed that the three suspension systems have design load in the middle of the same suspension spring rate and thus should have similar suspension characteristics. On closer inspection of these systems, there is an obvious major difference: While the mechanical suspension has uniform stiffness through the travel range (if not a progressive-rate spring is used), other design types are more or less progressive, due to the laws of the polytropic change of state of the gas as on Figure 1.2 (Bauer, 2008).

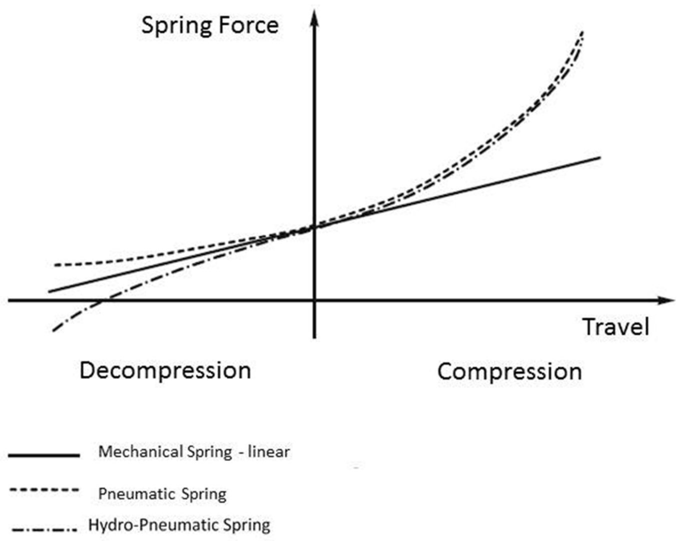

Figure 1.2 Force-distance curves for mechanical and gas spring systems (Bauer,2008)

From the standpoint of the constant natural frequency of the vibration system in principle, a linearly increasing spring rate is preferable. A constant spring rate for each load, as with the mechanical spring is in almost all cases only a compromise solution and is only recommended for suspension systems in which the relative load change is small. When the load has been increased without on the other hand increasing the spring rate and the suspension therefore becomes softer (lower natural frequency).For a good insulation, low natural frequency stands out, and thus a low spring rate, but without reaching the limits of travel. So to make sure that the suspension is able to cope with these extreme conditions it must be tuned more stiffly and with higher damping overall. The problem is that this worsens the tuning for all other load cases (for example with only the driver inside). Hence it can be easily deduced that a linearly wound coil spring can only allow a compromise for most driving situations. It is possible to address this problem by using progressively wound coil springs but this only partially solves the root cause. Therefore in most cases a level control is the far better and the far more effective solution – after a load change it brings the suspension back to its design position and ensures constant residual suspension travel in both compression and rebound direction (Bauer, 2008).

A mechanical spring will have a high natural frequency at low loads and a low natural frequency at high loads. For pneumatic suspension, low natural frequency is kept constant over all loading conditions, while the natural frequency of the hydropneumatic system with increasing axle loads will increase depending on the design. Figure 1.3 and 1.4 illustrates this case (Bauer, 2008).

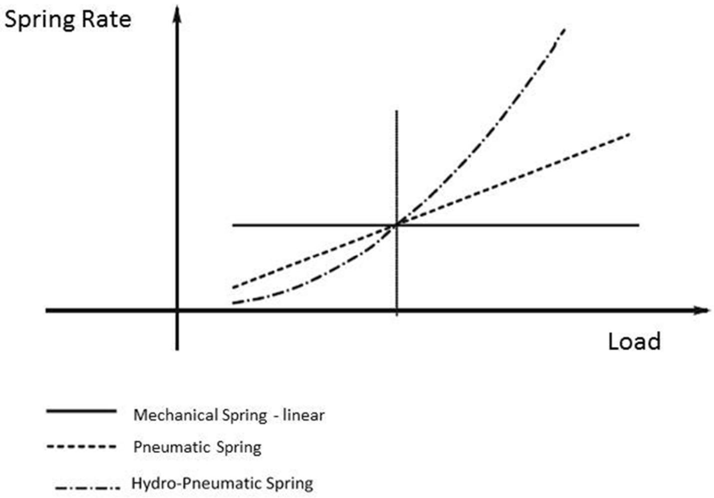

Figure 1.3 Spring rate as a function of load for for mechanical and hydropneumatic and gas spring systems (Bauer, 2008)

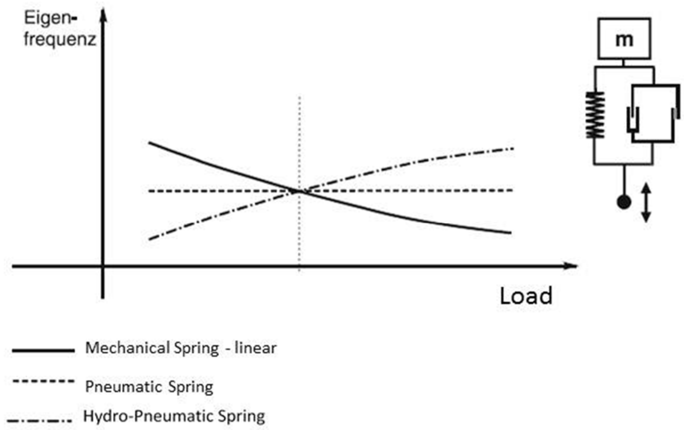

Figure 1.4 Force-natural frequency curves for mechanical and hydropneumatic and gas spring systems (Bauer, 2008)

1.1.2 Damping Characteristics

Damping comes through two mechanisms: The damping caused by solid friction, in particular within seals and guides, and the damping caused by viscous friction within a damping medium, usually oil. While the latter can be introduced as a pre-defined design element into the system in all three systems at the same degree, the solid friction is always an unpleasant side effect that must be avoided.

The mechanical spring with oil hydraulic damper cuts at this point from the best: This system introduces solid friction only on the oil-hydraulic dampers in the context of its seal friction. Although the pneumatic suspension system uses the same basic technology shock has, in the pneumatic system there is still the friction of the bellows, which originates from the deformation required during the spring movement, which is regarded as hysteresis. This friction causes a deterioration in ride comfort (also called "harshness"). To summarize, the solid friction on a pneumatic suspension system is thus always somewhat higher than the mechanical suspension.

For hydromechanical suspension system, the picture is again different. There are in fact only the suspension cylinders as components that can create friction, they must be sealed at very high pressures, thus the amount of friction without any special measures is expected to be higher than in the above two systems. Overall, the solid friction on hydropneumatic suspension cylinder is a particularly noteworthy and challenging topic, but out of the scope of this thesis.

1.1.3 Leveling

Leveling is the task of keeping the vehicle close to its nominal suspension height regardless of the carriage. For the hydro-pneumatic suspension system, a level control by injection of air or hydraulic fluid can be performed with accuracy through a control system. The same control rate that would be necessary in the pneumatic suspension would require very large volume flows and higher energy due to the compressible nature of the gas in system. This aspect plays a role particularly in implementing the leveling function; The hydropneumatic suspension is preferably used due to the compressibility resulting in quick responses and less volume flows for achieving the target level. When the vehicle is loaded, additional pressure is supplied by a hydraulic pump to lift the vehicle back to its nominal height, and when it is unloaded, it has to discharge and lower the vehicle to the nominal height again. Also this system can be manually controlled for different terrains. Vehicle height can be increased to meet off-road expectations for example.

Since the leveling function does not respond quickly, it does not come into action during ride when the vehicle meets with bumps, potholes and other road irregularities.

1.1.4 Compliance With The Non-Functional Requirements

This chapter will discuss the non functional requirements of the hydropneumatic suspension system.

1.1.4.1 Costs

At this point, the traditional mechanical suspension system obviously comes up on the pneumatic and hydropneumatic spring systems. This is because; first, that its components are optimized on an already long development period regarding the costs; and second, that here the technique for a complex level of regulation is not always required (Bauer, 2008). The pneumatic and hydropneumatic suspension have particularly higher manufacturing and replacement costs.

1.1.4.2 Space requirements of the shock absorber

Here, the hydro-pneumatic suspension has distinct advantages. Its suspension element is an integrated component. The space for the level control system can be provided in any position on the vehicle, thus it is not a matter of space. Also it can be said that a spring element of the pneumatic suspension under the same conditions can usually be accommodated in the space of a mechanical coil spring. The latter, however, usually has no level control system.

1.1.4.3 Reliability, safety, durability and maintenance

Generally speaking, reliability and security required for all systems are equally critical, if constructed accordingly, and the systems are maintained regularly. The mechanical springs can be described as maintenance free because the maintenance is limited here mainly to overtake and replace the oil-hydraulic dampers. Also the corrosion and damaging of metallic spring material must be kept in mind. The pneumatic system has basically the same maintenance as the mechanical suspension, but there must be a structurally higher operating expenses in order to protect the relatively fragile air spring elements since the rubber bellows are more vulnerable to climatic conditions. In particular, on the off-road vehicles the air bags are specially shielded against dirt, stones, sharp objects, etc. The bellows are subject to certain aging primarily by environmental influences such as UV radiation, dirt, chemicals, ozone, etc., and must therefore be replaced in many cases, after prolonged use. The hydropneumatic system requests maintenance depending on the interpretation. The commonly used membrane must prevent gas diffusion through itself, or there is a requirement for refilling the chamber. Special, gas-impermeable membranes or specific gases reduce these maintenance costs, but have higher initial costs. Despite all the maintenance required, the element has to be replaced after a long time due to aging of the membrane. Even an exchange of the oil may be necessary after some time because the oil ages and might absorb water. This affects the viscosity and the damping. A survey on compliance with various requirements under consideration by the three suspension systems is shown in Figure 1.5 (Bauer, 2008).

	Mechanical	**Pneumatic**	**Hydro-Pneumatic**
Suspension properties	o	++	++
Damping properties	++	++	+
Leveling	-	+	++
Cost	++	o	-
Installation space	o	-	+
Reliability / Robustness	+	o	+
Maintenance	+	o	o

Figure 1.5 Overview: meeting the requirements by the suspension systems (Bauer, 2008)

1.2 Applications for Hydro-Pneumatic systems

After the overview in Figure 1.5, it may be derived in general terms that hydro pneumatic suspension systems are primarily used among others because, when...

- a level control to compensate especially excessive load changes are required
- must respond quickly and often adjustable
- a manual adjustment of the level is requested
- little space for spring elements is available
- demand due to the harsh operating conditions
- an adjustment of the spring rate is desired
- hydraulic power is already available

These items are combined particularly in off-road vehicles of any kind, such as construction vehicles, cranes, tanks, agricultural mining and heavy-duty trucks. Other applications have become known as primarily by Citroen, and the passenger car sector, which in one particular load independent constant level position and the manual adjustment of the level position. For a passenger car, the hydraulic supply system was used not only to supply braking and steering, but also the suspension. Due to the high pressures required for the suspension, Citroen has brought in the Hydractive III system with a separate hydraulic circuit for the suspension. Also in rail transport, the hydro pneumatic suspension is used. For example, on mass transit rail vehicles, the level regulation function is used in particular to keep the car at any load at the desired height (platform edge). This simplifies the walk-in, exclusively for wheelchair users. A more exotic example of application is also towbarless tractor used at airports; they raise the nose wheel of an airplane to transport the plane to the starting position. Lifting loads of around fifty tons makes this height regulation a must (Bauer, 2008)

2. FUNDAMENTALS OF SUSPENSION AND DAMPING PROPERTIES OF HYDROPNEUMATIC SYSTEMS

2.1 General Structure and Function

Basic elements of the hydro pneumatic suspension are a hydraulic cylinder and a hydropneumatic pressure reservoir, which is mounted directly on the cylinder. In Figure 2.1 (Bauer, 2008), when the piston rod moves, the fluid volume changes, and thus the pressure in the accumulator (p1 → p2). This results in a change in force on the piston rod and thus the spring constant. This is the simple explanation of the variable spring rate.

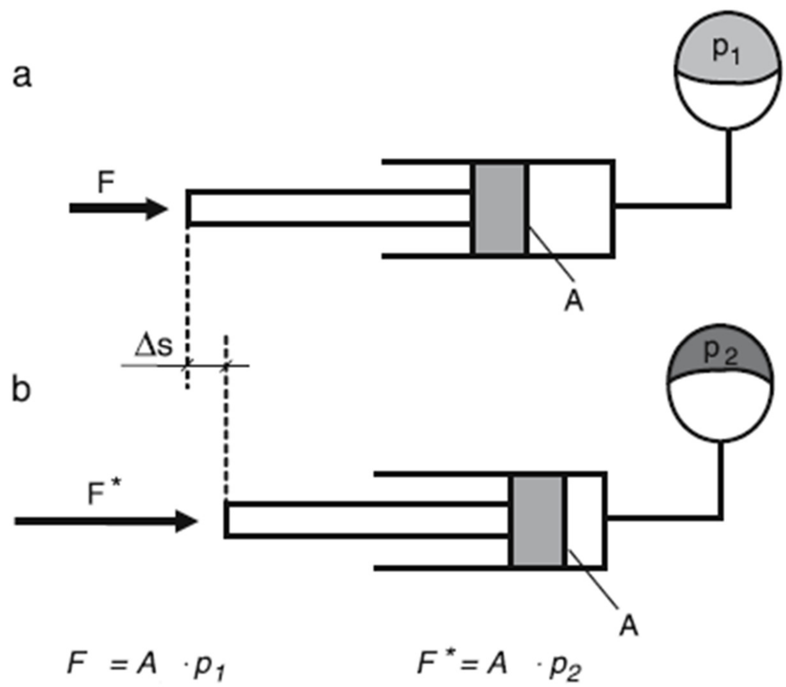

Figure 2.1 Balance of forces on the piston of a single acting cylinder (Bauer, 2008)

In the figure, there is an increase on the force, therefore the position of the piston changes (Δs) and a corresponding volume of hydraulic fluid is moved to the accumulator. This process continues until the pressure in the accumulator has therefore adopted a value on the piston surface, which keeps the system back in balance. This balance of forces is the basis for the function of the suspension system.

In order to achieve an additional damping, flow resistance is arranged, which converts a portion of the flow of hydraulic fluid into heat energy and thus damps the spring motion (viscous friction).

The level control unit consists of a position sensor and a check valve, which implements position adjustment of the suspension. There is also a tank to drain excess oil. This produces the following basic scheme of a hydropneumatic suspension system as seen in Figure 2.2 (Bauer, 2008).

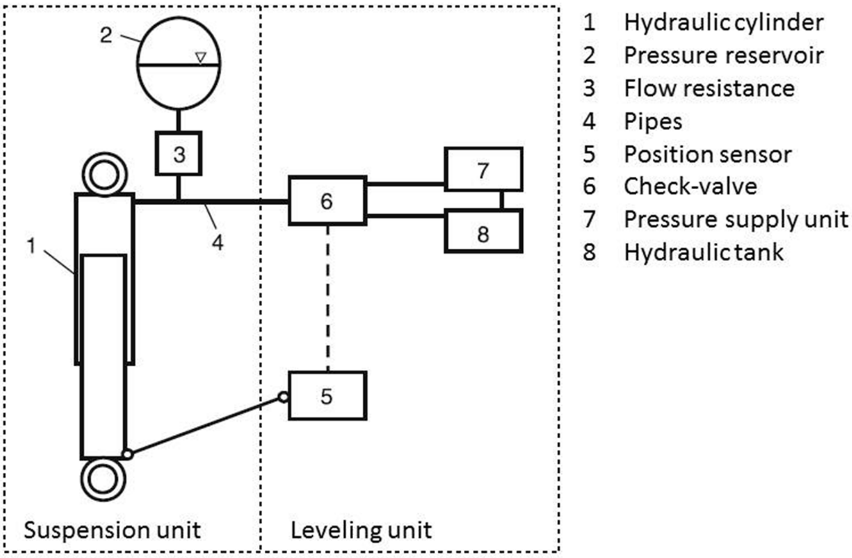

Figure 2.2 Basic design of a hydro-pneumatic suspension system (Bauer, 2008)

2.2 Suspension properties

An increase in pressure causes a variety of spring rates on the system:

- Compression of the gas in the accumulators
- Widening of the (elastic) line elements
- Compression of the hydraulic fluid

Each of these three effects specificy a different spring rate. To find the total spring rate, a serial connection of these three spring elements is required:

$$k = \frac{k_1.k_2.k_3}{k_1.k_2+k_1.k_3+k_2.k_3} \quad \textbf{(2.1)}$$

The stiffness of the line elements and the hydraulic fluid on suspension systems are generally very high; thus, their influence on the overall spring rate is so low. The significant characteristic property will be springing of the gas.

2.2.1 Physics of gases

The gas in the accumulators is the real elastic medium of the entire arrangement. The behavior of the entire suspension system is dependent on this medium. In the initial state with unpressurized hydraulic pressure system, certain mass of gas is present. This is indicated by the pressure reservoir volume V_0 (gas volume when no hydraulic pressure is applied) and define the accumulator pressure P_0. The accumulator pressure is always referred to the room temperature of 20 ° C (or 293.15 K). For this state, given the equation of state of ideal gas (ambient pressure is neglected):

$$P_0.V_0 = m_G.R.T_0 \quad \textbf{(2.2)}$$

$$P_1.V_1 = P_2.V_2 \quad \textbf{(2.3)}$$

$$P.V^{\gamma} = constant \quad \textbf{(2.4)}$$

2.3 Damping Properties

Damping mainly serves by converting the kinetic energy into heat during the movement by a decelerating force. This damping force is normally based on the principle of friction. There are two essential principles to be considered

- The solid friction, also called dry friction. the adhesion of a movement opposing frictional force.
- The fluid friction, also called the hydrodynamic friction. Here a liquid flow, a flow resistance is the opposite, making the liquid flow slow down. This deceleration creates a pressure buildup in the flow direction before the flow resistance and the active surfaces of the cylinder affects the pressure as a braking force as a damping force

2.3.1 Damping through solid friction

The solid friction acts as a resistance force between two solids pressed together with the normal force F_N. The resistance of friction force acts respectively opposite to the direction of movement (Figure 2.3). In the case of static friction, the friction force $F\mu$ is always as large as the tensile force F_Z. The static friction force $F\mu$ is the one depending on the nature of the two bodies, in particular materials and surface texture, which determines the friction coefficient μ, for others it is also dependent on the normal force F_N:

$$F_\mu = \mu . F_N \tag{2.5}$$

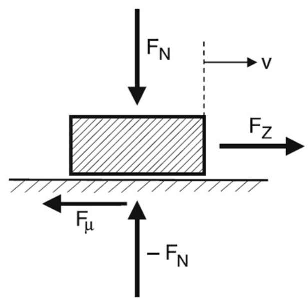

Figure 2.3 Forces in case of solid friction (Bauer, 2008)

Since the main damping mechanism for hydropneumatic suspensions is damping through fluid friction, solid friction section is cut here.

2.3.2 Damping through fluid friction

In a hydro-pneumatic suspension system oil is used as a medium for transmitting the pressure to the accumulator. The flow resistance produces a pressure drop Δ , which takes effect on the working surfaces as damping force (Figure 2.4).

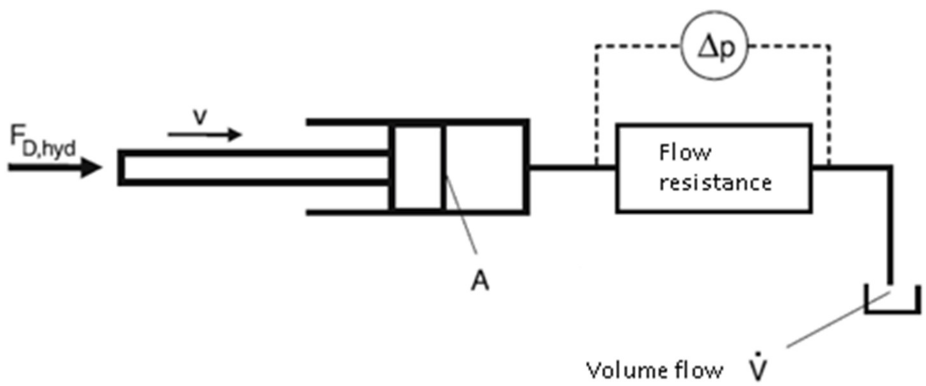

Figure 2.4 Forces in case of solid friction (Bauer, 2008)

$$F_{D,hyd} = \Delta p . A \quad \textbf{(2.6)}$$

$$P_{D,hyd} = F_{D,hyd} . v = \Delta p . \dot{V} \quad \textbf{(2.7)}$$

Most hydraulic fluids have a strong temperature dependent viscosity (Figure 2.5) and the damping is so temperature sensitive. A quite remarkable fact is the change in oil viscosity with the pressure. As the chart can be inferred from the kinematic viscosity ν in the ISO reference temperature 40 ° C at 200 bar bar increases by about 50% compared to the value at 0.

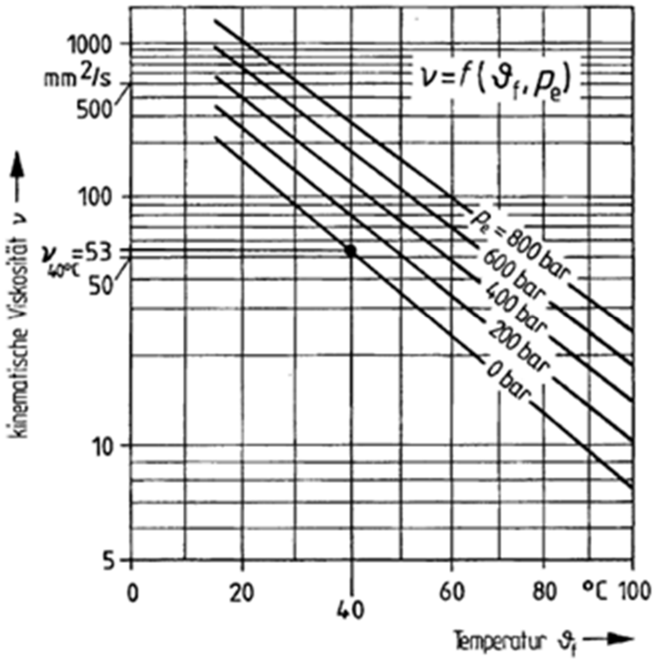

Figure 2.5 Viscosity-temperature-pressure behavior of typical hydraulic oil (Bauer, 2008)

Temperature dependence will not be considered to simplfy the calculations in the following chapters.

3. DESIGN OF THE HYDRAULIC DAMPING ELEMENTS

3.1 Overview

As mentioned, the hydraulic damping is achieved by flow resistance, which causes a loss of pressure. This then acts on the respective active surfaces within the cylinder and builds the appropriate damping forces. The cavitation limits the potential damping forces. Depending on the nature of the hydropneumatic suspension a different interpretation of the damping elements are implemented, thus no cavitation can occur (Bauer, 2008).

The flow resistance is interpreted:

$$F_{D,hyd} = \Delta p . A \tag{3.1}$$

$$P_{D,hyd} = F_{D,hyd} . v = \Delta p . \dot{V} \tag{3.2}$$

The cavitation is not a problem because the pressure that moves the fluid through the flow resistance is generated directly from the cylinder (Bauer, 2008).

Damping valve has to work with a back pressure of 7Mpa or more on heavy duty vehicles, which is quite different from common damping valves widely used in the shock absorbers. (Design of damping valve for vehicle hydro pneumatic suspension, 2006)

3.2 Flow Resistance

The pressure drop in a typical orifice is:

$$\Delta p = \dot{V} . v . \rho . K_D \tag{3.3}$$

wherein ρ is the density of the hydraulic fluid and the K_D Throttle constant from the geometric dimensions of the choke in Figure 3.1:

$$K_D = \frac{128.l_D}{\pi}.d_D^4 \tag{3.4}$$

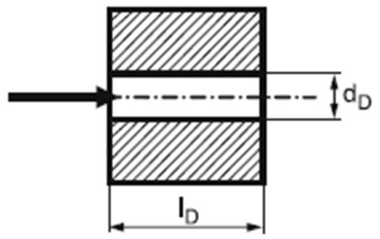

Figure 3.1 Schematic of a typical orifice (Bauer, 2008)

The flow experiences a sudden transition. This leads to the formation of strong turbulence in the hydraulic fluid, internal friction, generation of heat and thus causes a withdrawal of energy from the suspension system. Figure 3.2 displays the orifice mounted on the accumulator.

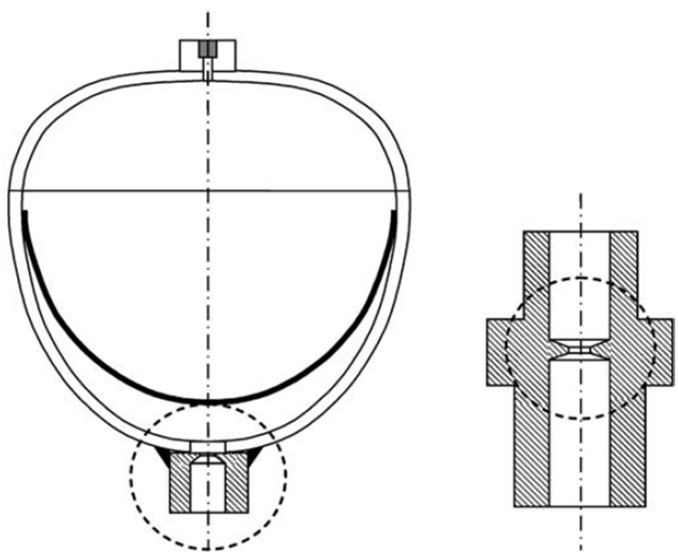

Figure 3.2 Integration of flow resistance (Bauer, 2008)

4. BUILDING THE ANALOGICAL MODEL

In this chapter, an analogical model of the hydropneumatic system will be constructed using one dimensional simulation software AMEsim. Each component and its theory and equations will be detailed. The main idea here is to equally construct the damping and suspension properties for the hydropneumatic suspension to its corresponding mechanical predecessor.

4.1 Design explanation of the basic quarter car model for a mechanical suspension

The quarter car model consists of the following;

- Quarter car body mass
- Suspension – spring
- Suspension – damper
- Wheel mass
- Tire stiffness and damping
- Road input

Figure 4.1 indicates these components on the sketch.

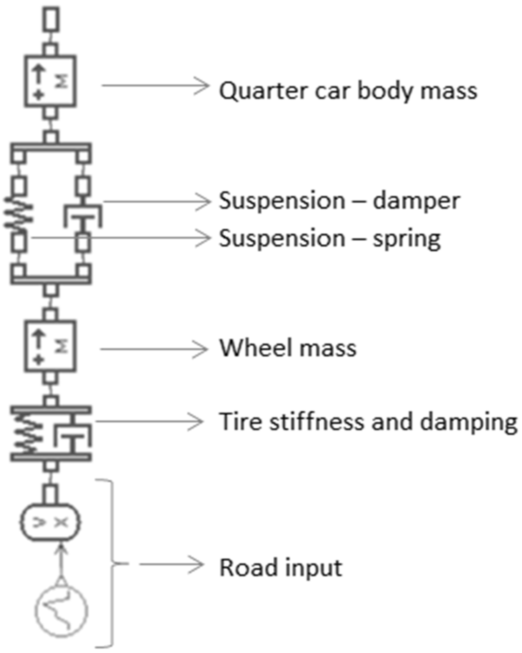

Figure 4.1 Schematic of a quarter car model with mechanical suspension

Seating system is neglected to simplify the situation. One fourth of the total mass for the vehicle will apply to the quarter car model.

The mechanical suspension coefficients are also selected in order to compromise D=0.25 damping ratio which is considered as construction standard for vehicle industry. As the damping increases, vibration amplitude is reduced at the body natural frequency range (around 1.7 Hz) and axle natural frequency range (around 11 Hz). In between these frequency ranges, the increased damping has an incremental effect on the vibration amplitude. This frequency range is related to the seating system. Seating comfort is seen to have a minimum around damping ratio D=0.2, and dinamic axle load ratio has a minimum at D=0.3, then constant. Considering both the comfort and ride safety, D=0.25 is considered as an optimum for vehicle construction (Taşıt Titreşimleri ve İrdenmesi, 1989), (Göktan, 1982).

The tire mass, stifness and damping are selected in order to simulate a tire as a spring-damper model also. No special tire model is employed. Each component of AMESim library used for this model is explained in detail below;

Road profile is constructed as a step input simulating a bump.

4.1.1 Two port mass capable of one-dimensional motion

Figure 4.2 represents one-dimensional motion of a two ports mass under the action of two external forces in N. The submodel returns the velocity in m/s, the displacement in m and the acceleration of the mass in m/s/s.

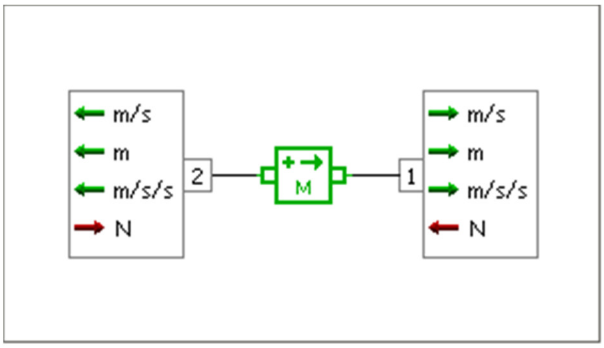

Figure 4.2 Two port mass capable of one-dimensional motion (LMS Imagine SA, 2009)

The acceleration and the derivatives of the two state variables are set as follows:

$$\dot{v} = (F_2 - F_1 - 9{,}81.m.\sin\theta)/m \qquad \textbf{(4.1)}$$

$$\dot{x} = v \qquad \textbf{(4.2)}$$

$$a = \dot{v} \qquad \textbf{(4.3)}$$

4.1.2 Ideal linear spring

The submodel in Figure 4.3 has two ports and gives force in N as outputs at both these ports. The spring compression in m is calculated from the two displacement inputs and the real parameter spring force with both displacements zero. From this and the spring stiffness the force at each port can be calculated. No limitation on the spring length is employed.

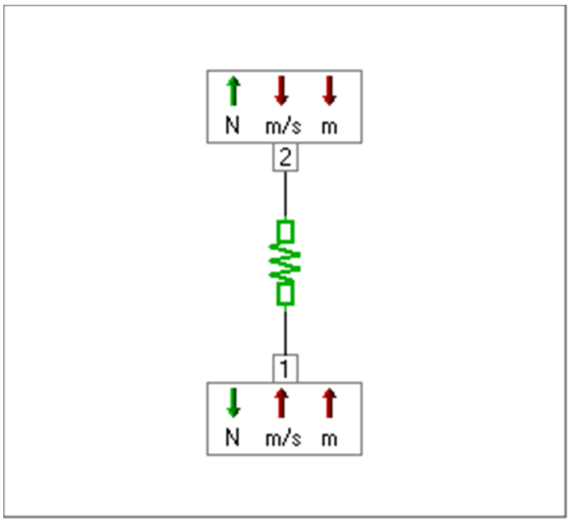

Figure 4.3 Ideal linear spring (LMS Imagine SA, 2009)

When geometrical parameters are used, the stiffness is computed using the expression for an helicoidal spring :

$$k = \frac{G.d^4}{8.D^3.n_a} \tag{4.4}$$

Where:

G is the material shear modulus [N/m²]

n_a is the number of active coils

D is the spring diameter [mm]

d is the wire diameter [mm]

D and d are represented on Figure 4.4:

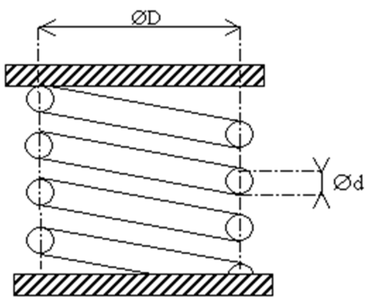

Figure 4.4 Geometrical indication of linear spring (LMS Imagine SA, 2009)

4.1.3 Damper with two ports

Figure 4.5 is a damper with a constant damper rating. The submodel has two ports and gives a force [N] as outputs at both ports.

The damper rating information is required by the submode in[N/(m/s)] for constant damping. (LMS Imagine SA, 2009)

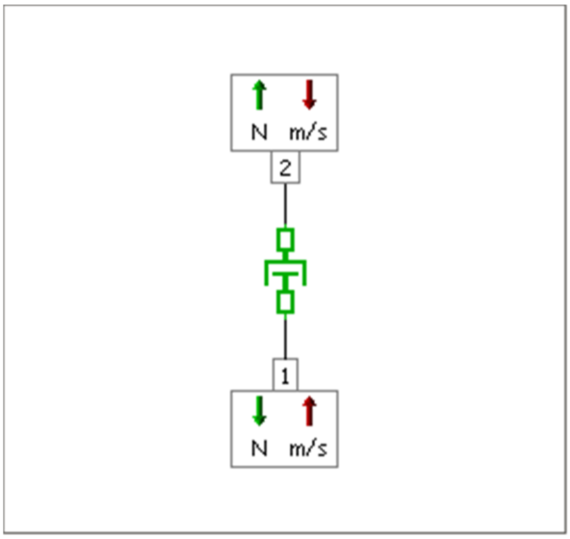

Figure 4.5 Damper with two ports (LMS Imagine SA, 2009)

The computations here will be based on constant damper rating thus the output forces *force1* and *force2* are computed as follows:

$$F_1 = F_2 = c.(v_1 + v_2) \tag{4.5}$$

4.1.4 Mechanical spring and damper (Tire suspension modeling)

Figure 4.6 is an ideal spring-damper system. The submodel has two ports and gives force in N as outputs at both these ports. Although this submodel is suitable for general use in linear mechanical systems, it will be used for modeling the tire stiffness and damping as linear for simplifying the model.

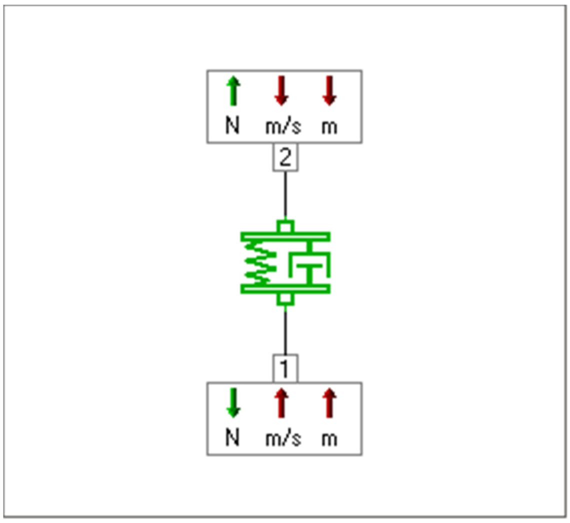

Figure 4.6 Mechanical spring and damper (Tire suspension modeling) (LMS Imagine SA, 2009)

4.1.5 Signal converter

Figure 4.7 converts a dimensionless signal input at port 1 to a linear displacement with the same value in m which is output at port 2 with a linear velocity in m/s. This velocity is obtained by an approximate differentiation of the angle. The initial velocity is set to zero.

Figure 4.7 Signal converter. (Tire suspension modeling) (LMS Imagine SA, 2009)

4.1.6 Piecewise linear signal source

Figure 4.8 is a submodel with a dimensionless signal output. Linear interpolation is used to determine the output. Thus constant sections, ramps and steps may be constructed.

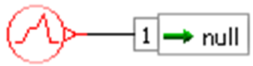

Figure 4.8 Piecewise linear signal source (LMS Imagine SA, 2009)

4.2 Design of a hydro-pneumatic suspension unit having the same damping and stiffness with the mechanical one

The purpose is to model a hydro-pneumatic suspension and design it compared to a damping and a stiffness we choose.

4.2.1 Determining the elements to replace the mechanical spring and damper

The vehicle construction requires that the vehicle body is connected directly to the spring and damper unit, and this mass is conveyed to the tires and road through this unit. It has already been described in section 2 that the damper unit consist of a piston blocked by an orifice and an accumulator as the springing medium. The first step is constructing the piston to transfer road forces to the hydraulic fluid causing pressure differences, then putting an orifice in between to create appropriate damping through pressure drop, and framing the system with an accumulator as the energy storage and release medium as the spring unit.

As discussed earlier, the hydro pneumatic suspension unit employs a hydraulic damper, and a gas chamber (actuator) as the spring. Hydraulic damper unit may consist of a hydraulic piston plus an orifice to simulate the damping effect. After the orifice, the actuator will be positioned in order to simulate the springing effect. Following is the discussion of each element in detail.

4.2.2 The Hydraulic Piston

The main element of the hydraulic damper unit is the piston. Before employing the orifice, the piston is examined as follows;

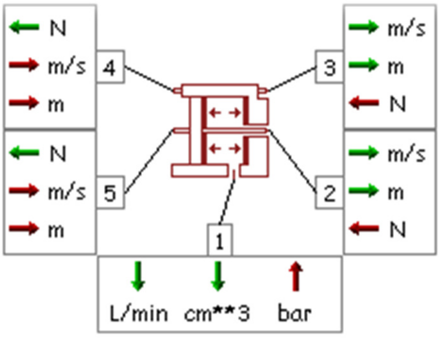

Figure 4.9 The hydraulic piston (LMS Imagine SA, 2009)

As seen on figure 4.9, Port 4 will be connected to the body of the vehicle (input), port 5 will be connected to the tire (output). Ports 2 and 3 will be dummy. Port 1 is the opening for the fluid flow through the orifice and the accumulator.

The chamber length and volume are defined as;

$$length = x_4 + x_0 - x_5 \quad \textbf{(4.6)}$$

$$vol = length.\frac{\pi}{4}(d_p^2 - d_r^2) \quad \textbf{(4.7)}$$

The flow rate at port 1 is:

$$Q_1 = (v_5 - v_4).\frac{\pi}{4}(d_p^2 - d_r^2).\frac{\rho(p_1)}{\rho(p_0)} \quad \textbf{(4.8)}$$

The force at port 5 is calculated from the force at port 2 plus and the inner pressure and inner area:

$$F_5 = F_2 + p_1..\frac{\pi}{4}(d_p^2 - d_r^2) \quad \textbf{(4.9)}$$

Calculation of the force at port 4 is similar:

$$F_4 = F_3 + p_1..\frac{\pi}{4}(d_p^2 - d_r^2) \quad \textbf{(4.10)}$$

Default piston area is 10 cm². It is left unchanged. The same damping ratio with the mechanical suspension has to be achieved.

4.2.3 The Orifice (Fixed)

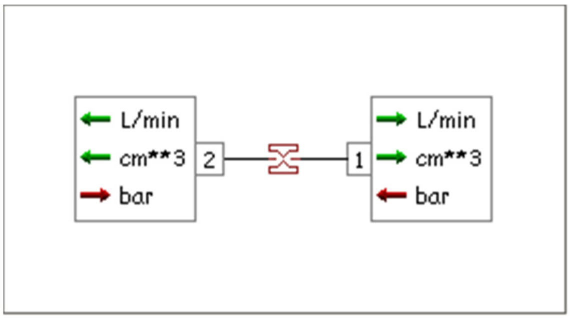

Figure 4.10 The Orifice (Fixed) (LMS Imagine SA, 2009)

Background information about the orifice is presented in Chapter 3. Considering the Bernoulli equation, it is possible to find the value of the equivalent orifice area. Figure 4.10 shows the fixed orifice model

4.2.4 The Orifice (Alternative)

This submodel in Figure 4.11 is used to calculate the flow rate through a laminar resistance orifice.

The pressure at each port is input in bar and the flow rate in L/min are computed at both these ports.

Figure 4.11 The Orifice (Alternative) (LMS Imagine SA, 2009)

The flow rate is directly proportional to the differential pressure and inversely to the absolute viscosity and the contact length presented in Figure 4.12.

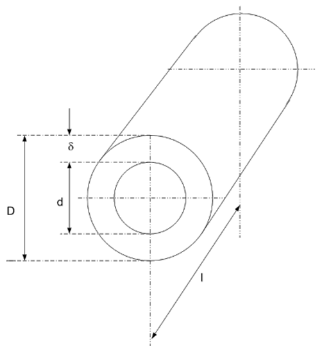

Figure 4.12 Geometrical scheme for the orifice (Fixed) (LMS Imagine SA, 2009)

4.2.5 The Check Valve

This component in Figure 4.13 is a hydraulic shuttle valve. A pressure is input at each port and a flow rate is computed and output at those ports.

No dynamics are incorporated. When one of the flow paths is open, the shuttle valve flow rate - pressure characteristics are assumed to be linear.

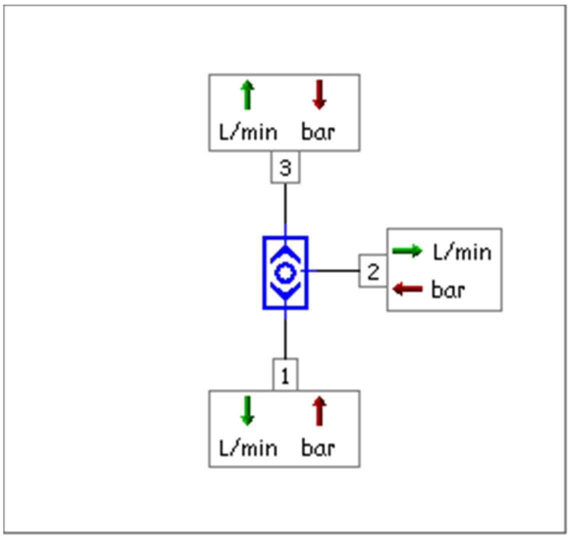

Figure 4.13 The check valve (LMS Imagine SA, 2009)

Operating modes of the check valve are shown in Figure 4.14 and 4.15:

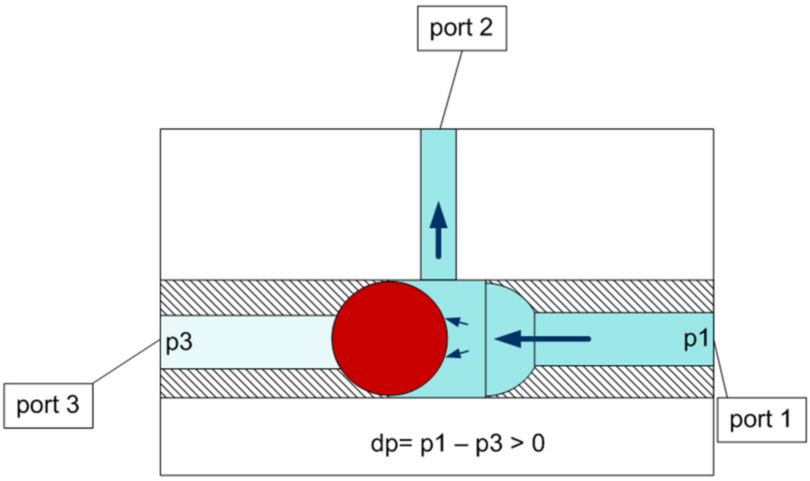

Figure 4.14 Check valve operating mode 1 (LMS Imagine SA, 2009)

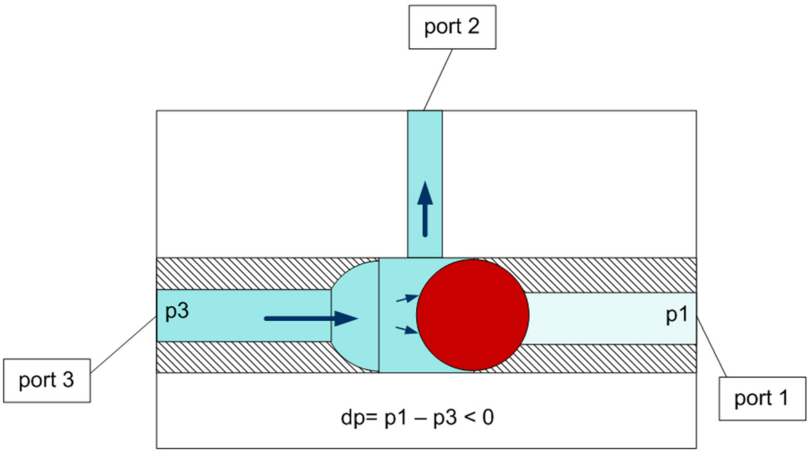

Figure 4.15 Check valve operating mode 2 (LMS Imagine SA, 2009)

4.2.6 The Hydraulic Chamber

This chamber in Figure 4.16 is employed in order to supply the required type of input for both the piston and the orifice, which is pressure. Also when the leveling function is required, this chamber is required to combine inputs from different sources (accumulator, pump) into the hydraulic piston.

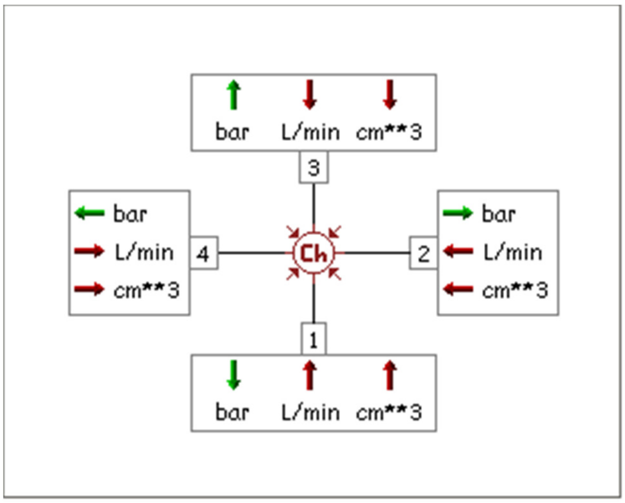

Figure 4.16 The hydraulic chamber (LMS Imagine SA, 2009)

Represents a hydraulic chamber with variable volume and pressure dynamics. Each port receives a flow rate in L/min and a volume in cm^3 as inputs and gives a pressure as output. The total volume is calculated by summing the four volume inputs and a dead volume which is a parameter of the submodel. This is used, together with the bulk modulus of fluid at the current pressure, to compute the hydraulic stiffness and hence the time derivative of pressure. Then the pressure in bar can be output at each port.

The basic formula used for computing the derivative of pressure in terms of the net flow rate and total volume is

$$\frac{dp1}{dt} = \frac{B(p1)\sum q(p1)}{\left(\sum v + vol0\right)} \tag{4.11}$$

4.2.7 The Accumulator

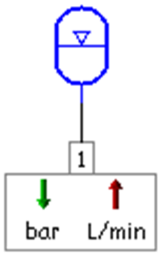

Figure 4.17 The accumulator (LMS Imagine SA, 2009)

The hydraulic accumulator in Figure 4.17 is an energy storage device. It is a pressure storage reservoir in which a non-compressible hydraulic fluid is held under pressure by an external source, compressed gas in this case. The accumulator gas is assumed to obey a polytropic gas law of the form

$$P.V^{\gamma} = constant \tag{4.12}$$

where the constant is defined by the precharge pressure and the accumulator volume. The hydraulic fluid within the accumulator is assumed to have the same pressure as the gas.

In our case, the accumulator will simulate the spheres in hydro pneumatic system, thus the springing medium. It is composed of two chambers seperated by a desmopan rubber membrane.

This component of the model requires three parameters: P_0, V_0 and P_i. It is required to determine the gas precharge pressure P_0, the accumulator volume V_0 and the initial pressure P_i to have an equivalent mechanical stiffness.

The spring stiffness of the mechanical spring was chosen so that the damping coefficient would be around 0,25 which is the considered value for general ride comfort.

The state is assumed to be isothermal; the accumulator is initially at equilibrium, thus its initial operating is considered isothermal:

$$P_0.V_0 = P_i.V_i \qquad \textbf{(4.13)}$$

and adiabatic;

$$P.V^{\gamma} = constant \qquad \textbf{(4.14)}$$

Gamma here is the polytropic index and is 1,4. Mechanical stiffness is defined by

$$K_{mec} = \frac{dF}{dx} \qquad \textbf{(4.15)}$$

And hydraulic stiffness is defined by

$$K_{hyd} = \frac{dP}{dV} = \frac{d(F/S)}{d(S.x)} = \frac{K_{mec}}{S^2} \qquad \textbf{(4.16)}$$

Derivation of the adiabatic equation yields;

$$(dP.V^{\gamma}) + (P.\gamma.V^{\gamma-1}dV) = 0 \qquad \textbf{(4.17)}$$

$$dP.V^{\gamma} = -P.\gamma.V^{\gamma-1}dV \qquad \textbf{(4.18)}$$

$$K_{hyd} = \frac{dP}{dV} = \frac{-P.\gamma.V^{\gamma-1}}{V^{\gamma}} = -P.\gamma.V^{(\gamma-1)-\gamma} = \gamma.\frac{P_i}{V_i} \qquad \textbf{(4.19)}$$

Pi initially equilibrates car weight:

$$P_i = \frac{F}{S} = \frac{m.a}{S} \qquad \textbf{(4.20)}$$

P_i is calculated using equation 4.20.

$$V_i = \gamma.\frac{P_i}{K_{hyd}} = \gamma.\frac{P_i}{\frac{K_{mec}}{S^2}} \qquad \textbf{(4.21)}$$

From the isothermal equation;

$$P_0 = \frac{P_i . V_i}{V_0} \qquad \textbf{(4.22)}$$

V_0 can be calculated using the maximum clearance of the suspension X_{max} and surface area of the piston S;

$$V_{0min} = 2.X_{max}.S \qquad \textbf{(4.23)}$$

Finally, the last parameter P_0 is calculated using;

$$P_0 = \frac{P_i . \left(\gamma . \frac{P_i}{\frac{K_{mec}}{S^2}} \right)}{V_0} \qquad \textbf{(4.24)}$$

4.2.8 3 position 3 port hydraulic control valve

This component in Figure 4.18 is used for leveling and is a simple submodel of a 3 port 3 position closed center proportional valve with pressure input at each hydraulic port and flow rate in computed and output at those three ports.

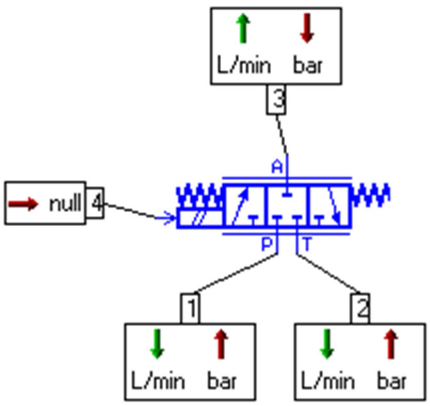

Figure 4.18 Servo valve (LMS Imagine SA, 2009)

The spool dynamics is modeled as a 2nd order system with user supplied natural frequency and damping ratio.

For each position, the flow paths are :

- from P to A in the left block

- no ports are linked in the central block
- from A to T in the right block

A flow rate and a corresponding pressure drop have to be supplied to the hydraulic ports 1, 2 and 3. The input signal at port 4 is limited to a user supplied value.

4.3 The Resulting Analytical Hydro-Pneumatic Suspension Model

Figure 4.19 to 4.25 display the models constructed using mentioned submodels. Analysis results will be presented in Chapter 5.

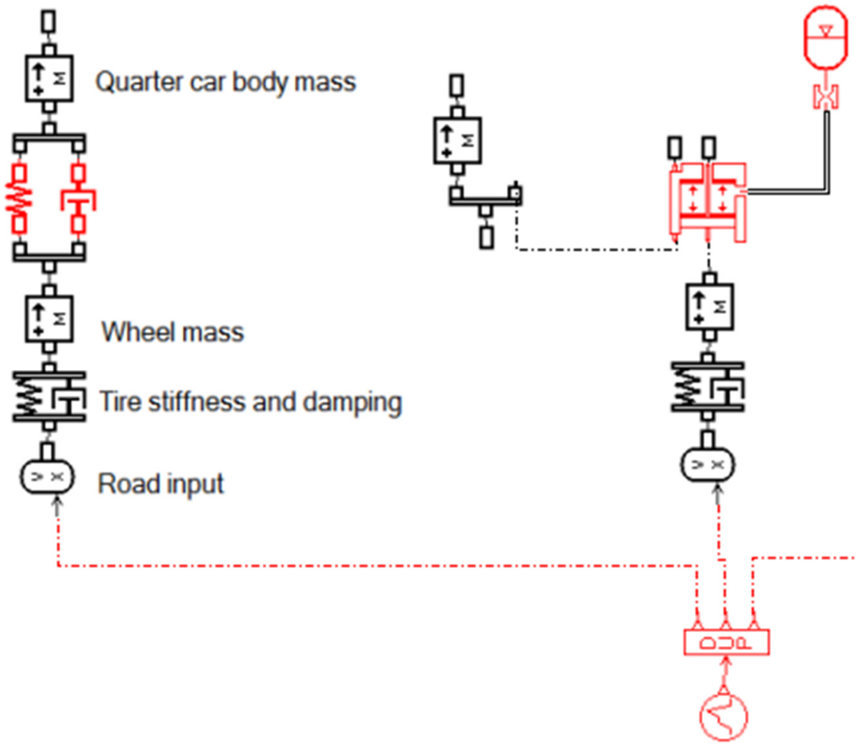

Figure 4.19 Mechanical and hydropneumatic model used for quarter car simulations

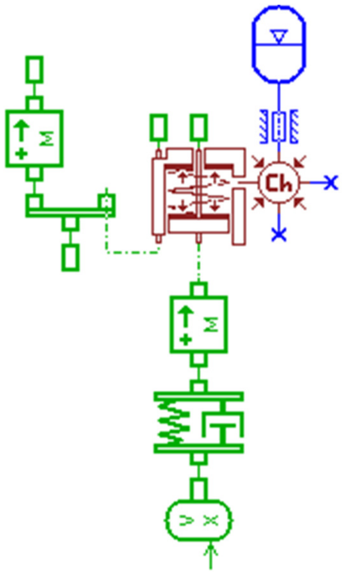

Figure 4.20 Hydropneumatic model with alternative orifice and additional mechanical spring for quarter car simulations

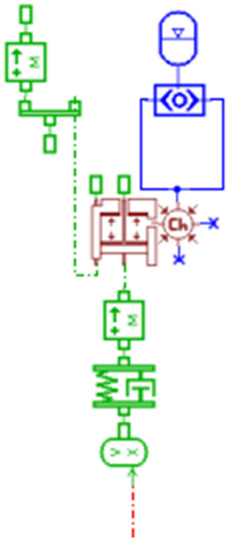

Figure 4.21 Hydropneumatic model with check valve for quarter car simulations

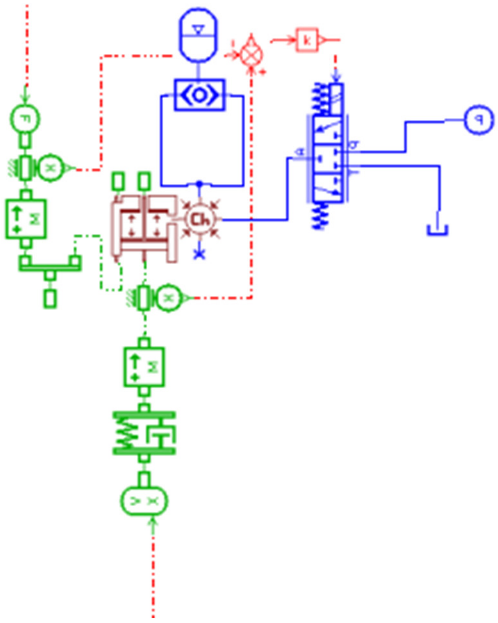

Figure 4.22 Hydropneumatic model with check valve and leveling system for quarter car simulations

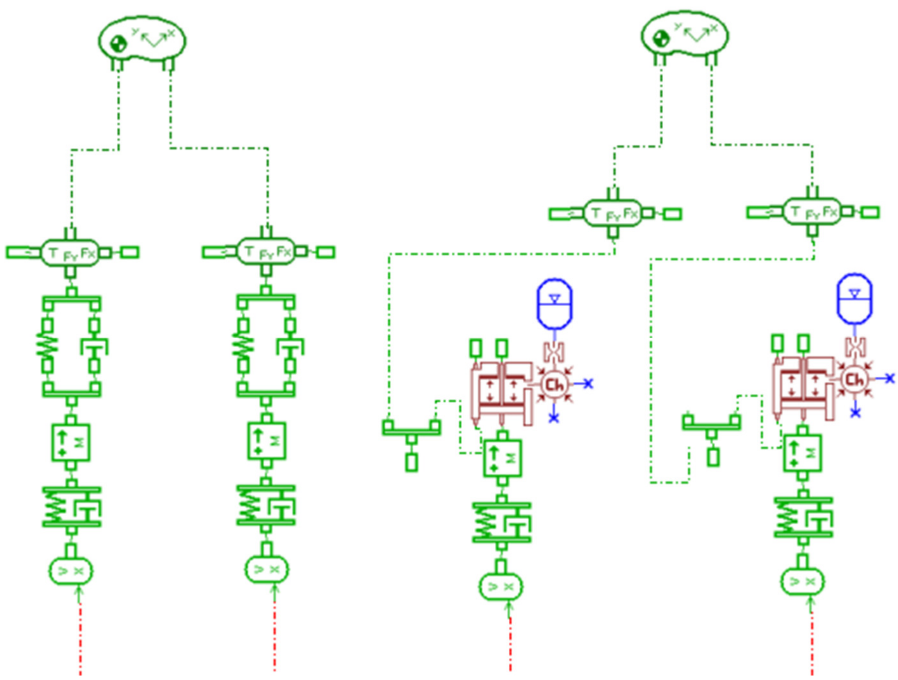

Figure 4.23 Mechanical and hydropneumatic model with check valve and leveling system for bicycle model simulations

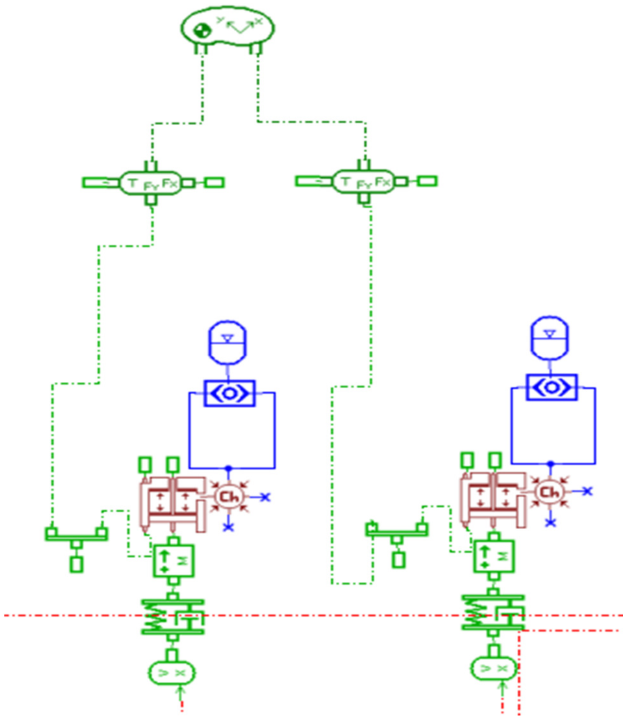

Figure 4.24 Hydropneumatic model with check valve for bicycle model simulations

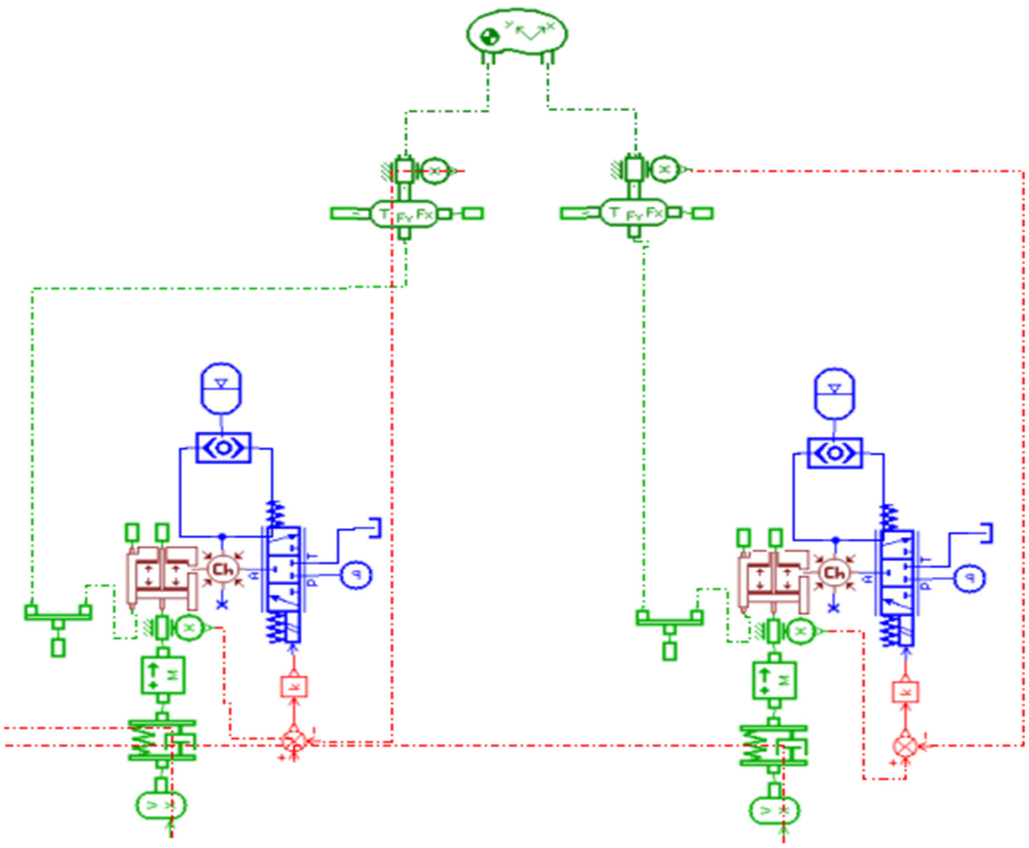

Figure 4.25 Hydropneumatic model with check valve and leveling for bicycle model simulations

5. SIMULATIONS WITH THE ANALYTICAL MODEL

5.1 Quarter car model – Mechanical versus Hydro Pneumatic

The response to a step excitation in Figure 5.1 will confirm the calculations:

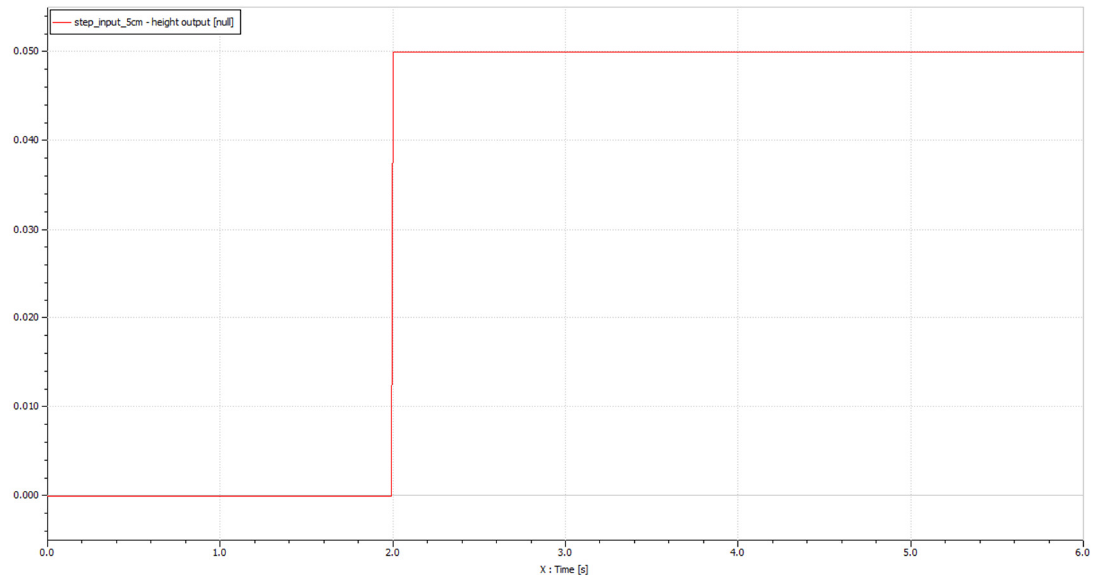

Figure 5.1 Step input used for simulations

Figure 5.2 displays the time-body displacement graph excited by the step response.

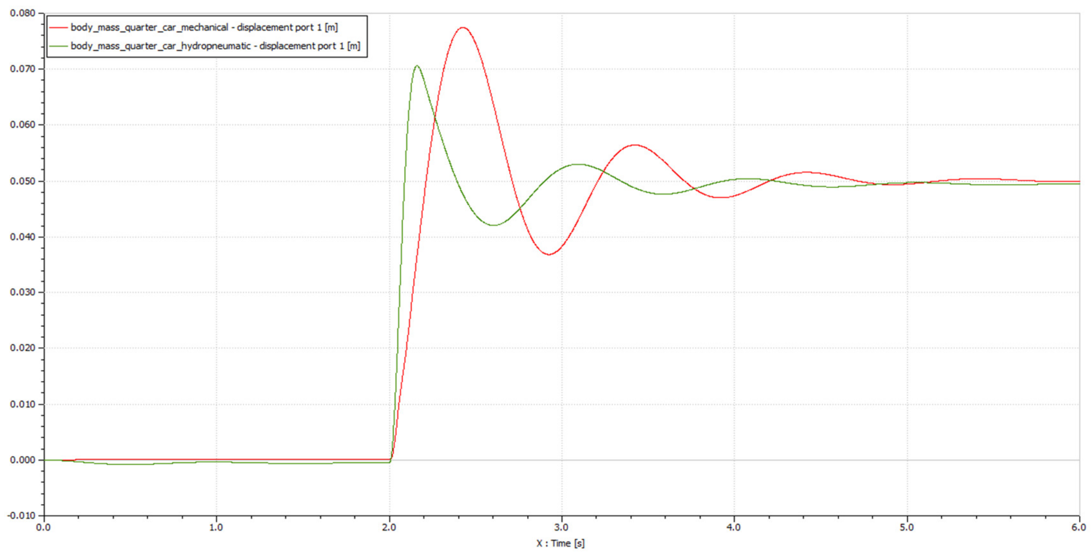

Figure 5.2 Displacement esponse of the systems to a step input

Chart analysis confirms that the stiffness and damping values are equivalent. Stiffness is equivalent because the mass and the period of the motion (natural frequency) are the same. The damping coefficient is the same because the ratio of the amplitude peaks is the same. Also the damping coefficient can be directly calculated from the chart through the following:

$$c = 2.m.\frac{\ln\left(\frac{f1}{f2}\right)}{t2 - t1} \quad \textbf{(5.1)}$$

Here, f_1 is the amplitude of the first peak in time-displacement graph, f_2 is the value of second amplitute peak; t_1 and t_2 are corresponding time stapms of these two amplitudes (Göktan, 1982).

Up to this point, the stiffness and damping concept are successfully implemented into the hydropneumatic suspension model. It was just a piston, an orifice, and an accumulator in hand; and according to the time-displacement graph, it is understood that equivalent hydraulic damping and pneumatic springing effects with the mechanical predecessor are achieved.

It should be noted that the hydropneumatic suspension has a shorter settling time than the mechanical suspension.

In order to judge the efficiency of a car damper, two criteria are commonly used. The first one is the criterion of handling, to see if the wheel follows the road. Hence, we can look at a transfer function Zwheel/Zroad to put forward the frequency interval where the value is closed to one .

The second criterion concerns comfort, this is to say the way the body car will insulate passengers from road perturbation. To apply this criterion, we usually look at vertical acceleration with a transfer function Z..body/Zroad. But this criterion can be differently considered, knowing that the car body must follow low frequencies undulations of the road and filter middle and high frequencies. It is equivalent to analyse a transfer function Zbody/Zroad. Which must be closed to one at low frequencies and as weak as possible for middle and high frequencies. (LMS Imagine SA, 2009)

The model with the orifice presented an overdamped result in terms of body displacement transfer function in Figure 5.3.

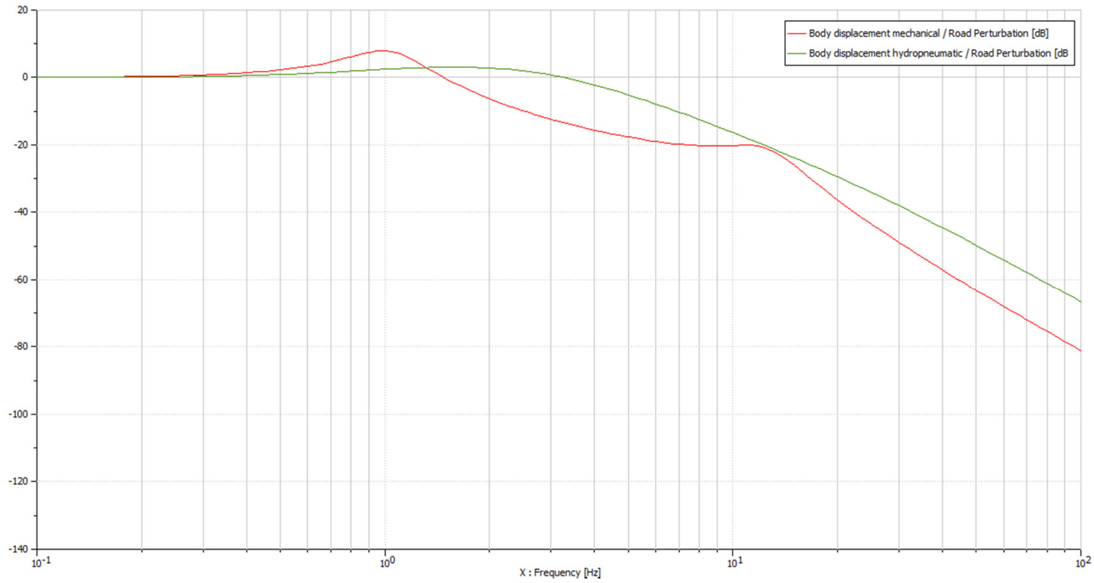

Figure 5.3 Body displacement transfer function

When the orifice is replaced by the check valve, the overdamped nature has changed to another version in Figure 5.4, still requiring to be tuned around 1,3 Hz.

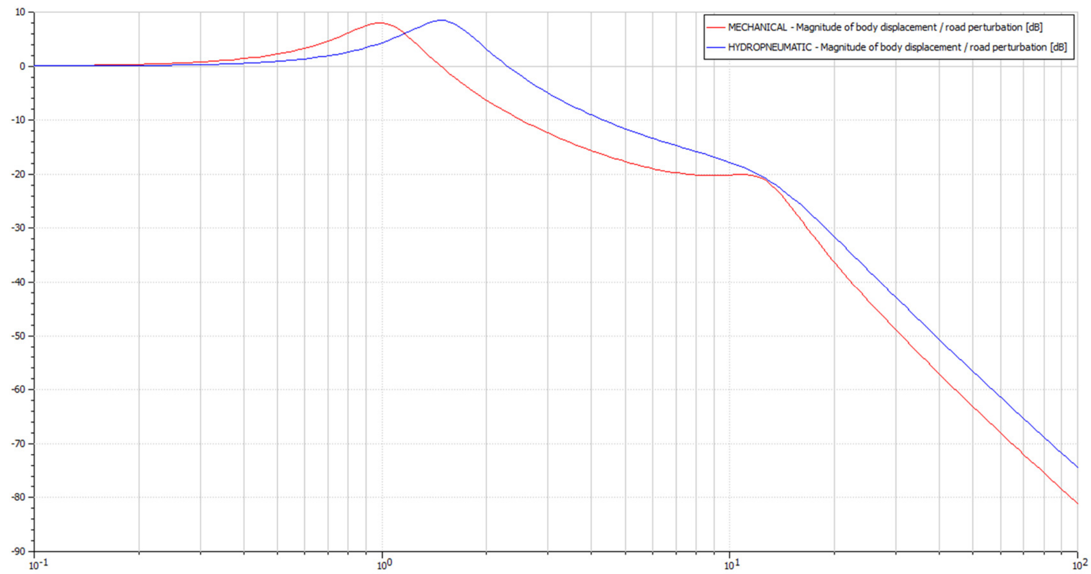

Figure 5.4 Body displacement transfer function (with check valve)

Body acceleration transfer functions of the first two models are seen on Figure 5.5. The result seems overdamped although the damping calculated was the same.

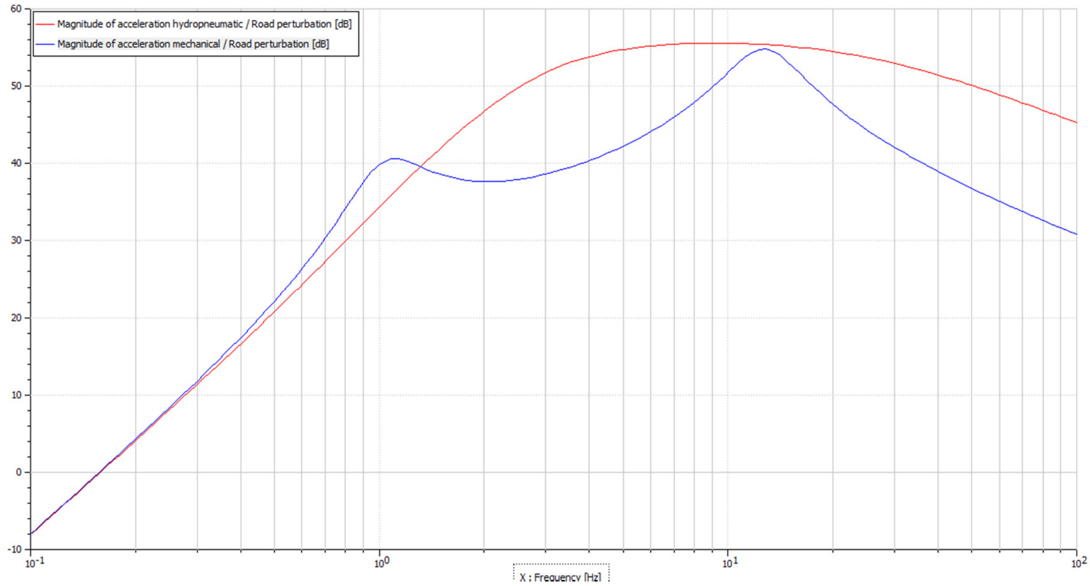

Figure 5.5 Body acceleration transfer function

Using the check valve returns a less damped transfer function result as in Figure 5.6, but requiring to be tuned around 1,3 Hz. On the span of frequency interval, the result with the check valve is rewarding.

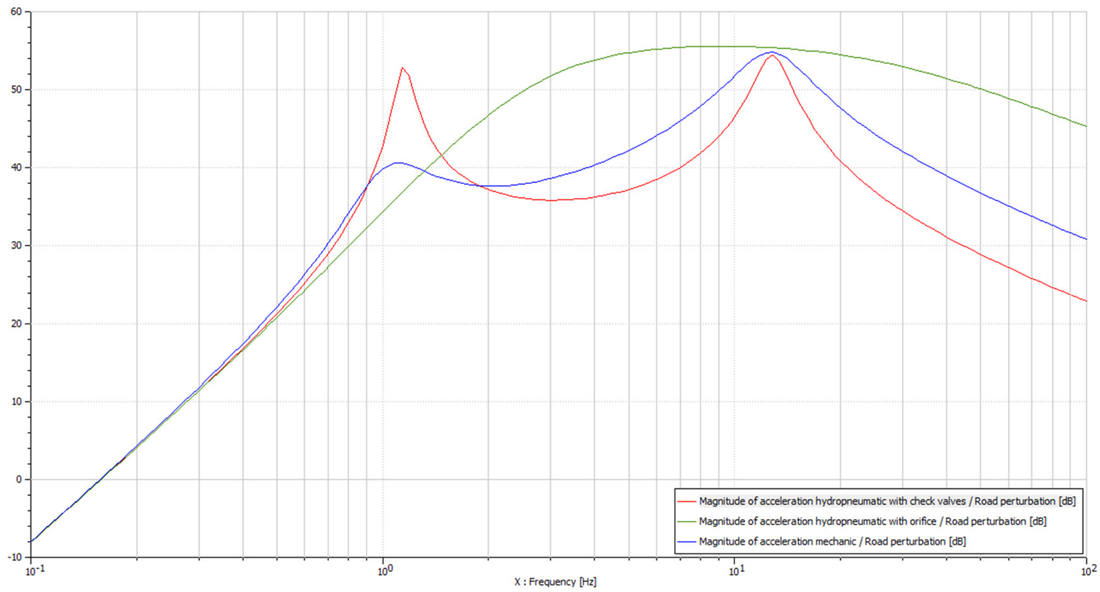

Figure 5.6 Body acceleration transfer function (added the model with check valve)

To judge if the wheel follows the road, wheel displacement transfer function is extracted as in Figure 5.7. Hydropneumatic seeems to perform better than mechanical alternative in terms of ride performance.

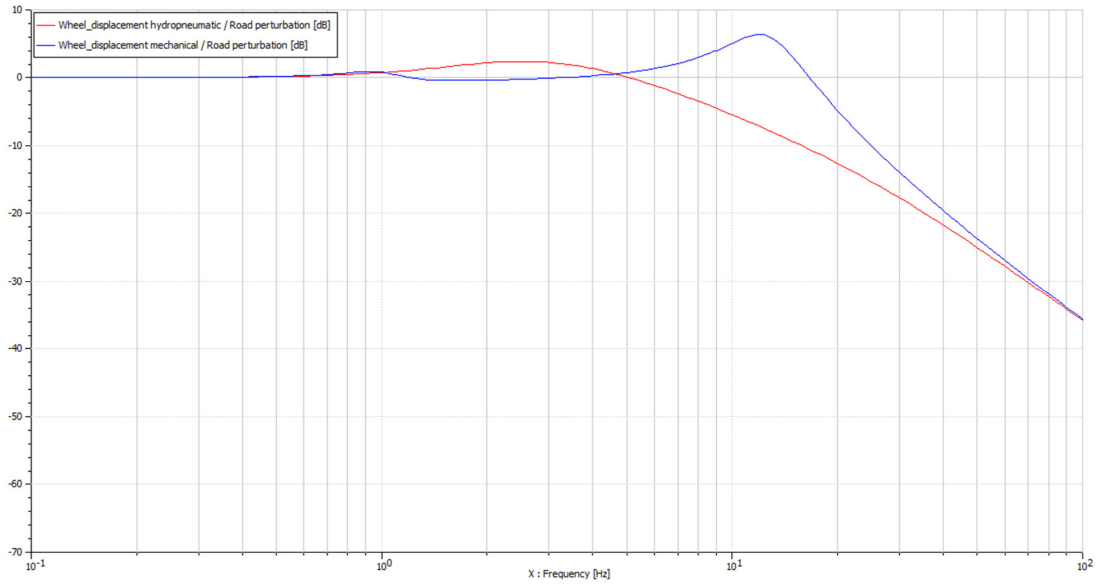

Figure 5.7 Wheel displacement transfer function

Figure 5.8 shows the FFT of body acceleration. Although the model could not be verified experimentally, the correlation between two models is satisfactory.

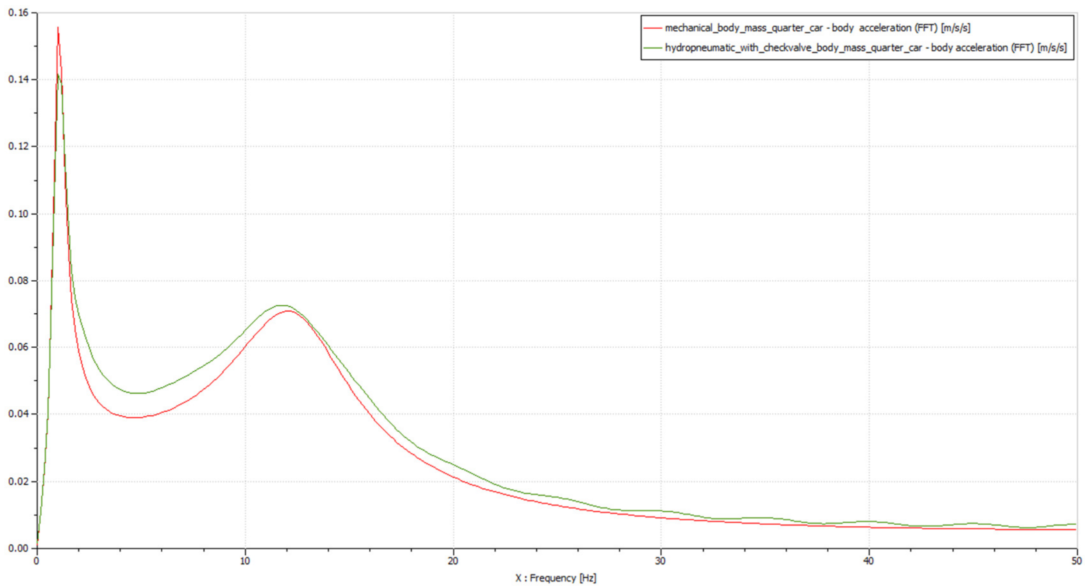

Figure 5.8 Body acceleration transfer function

5.2 Leveling function

Leveling is one of the superior functions for a hydropneumatic suspension. Four different simulation results are combined in Figure 5.9. The case simulates a quarter car exposed to 500 N of additional load at t=0. Between t=0 and t=2s, only the hydropneumatic with leveling starts to compensate the height decrease created by additional loading and ends compensation around four seconds. Mechanical performs the worst in terms of height compansation. It is seen on the graph that the complete compensation of height will be around t=4s; which indicates that this type of height compensation will not be activated for sudden inputs during ride, which may preserve tha car from invariably chaning height without the driver's will.

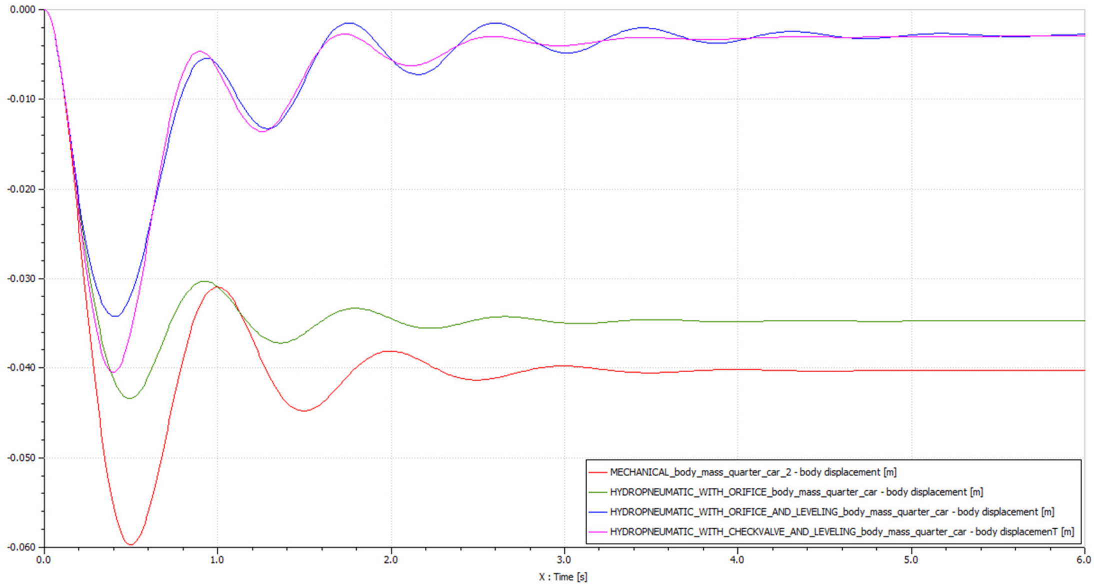

Figure 5.9 Body level change after initial loading

5.3 Bicycle model

The bicycle model is constructed with a length of 3m, front and rear wheels on one side of the vehicle. The excitation signal in Figure 5.10 is seleceted so that when the front wheel hits a bump, the rear wheel will be hitting a hole, this will maximize the effect of excitation on the suspension.

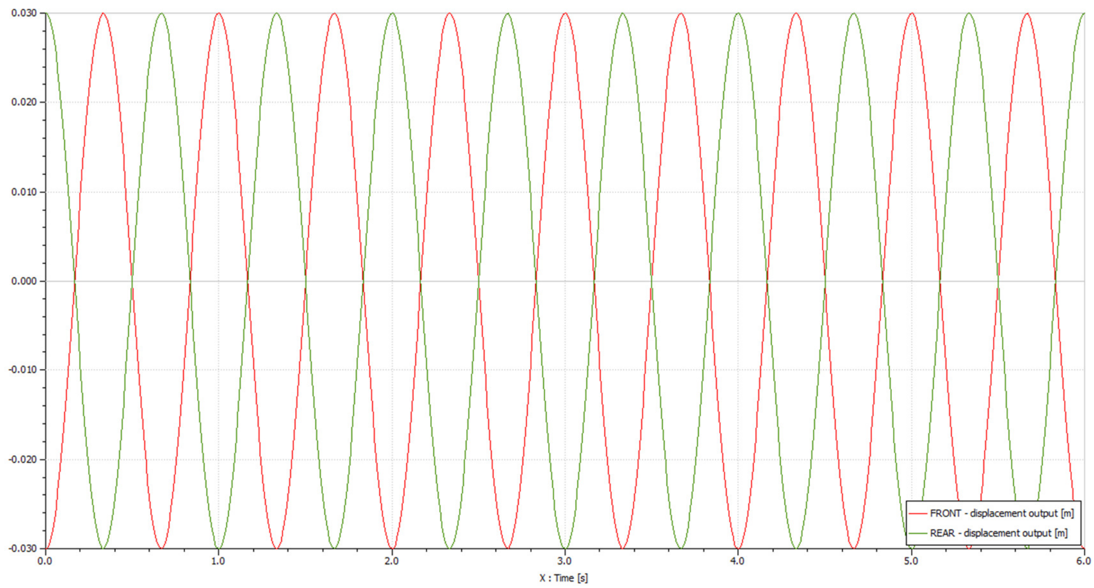

Figure 5.10 Sine excitation signals for front and rear wheels

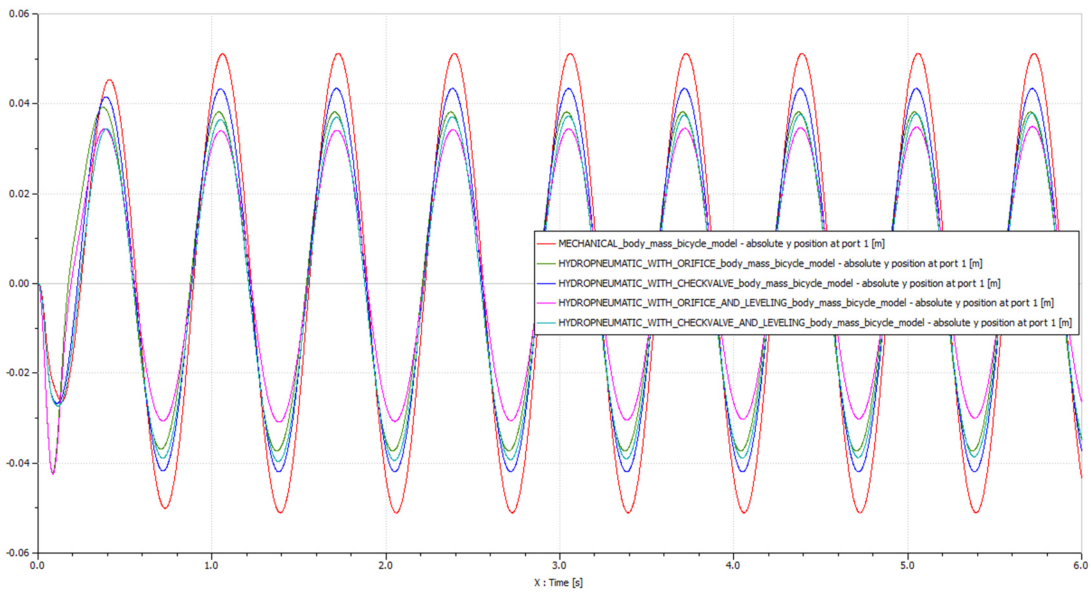

Figure 5.11 Body displacement response to sine excitation signals

Figure 5.11 shows the response of the vehicle's body to the sine input. Although the damping is calculated to be the same on each suspension model, hydropneumatics give less response, especially when used with orifice and leveling.

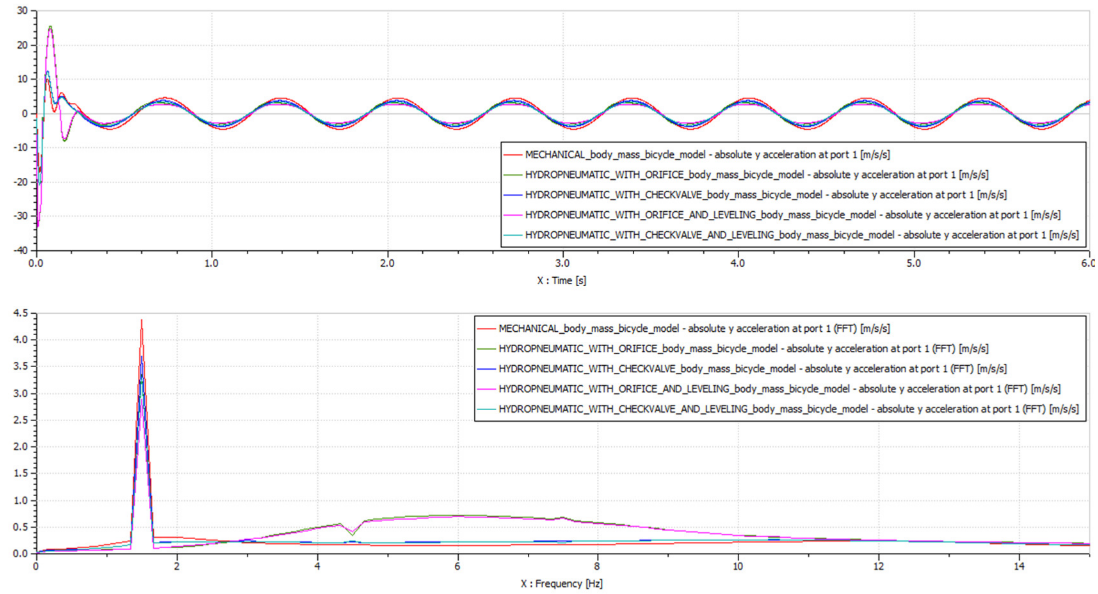

Figure 5.12 Body acceleration response and FFT to sine excitation signals

Figure 5.12 shows that the models with orifice present an overdamped nature. When they are taken out, the result is as in Figure 5.13

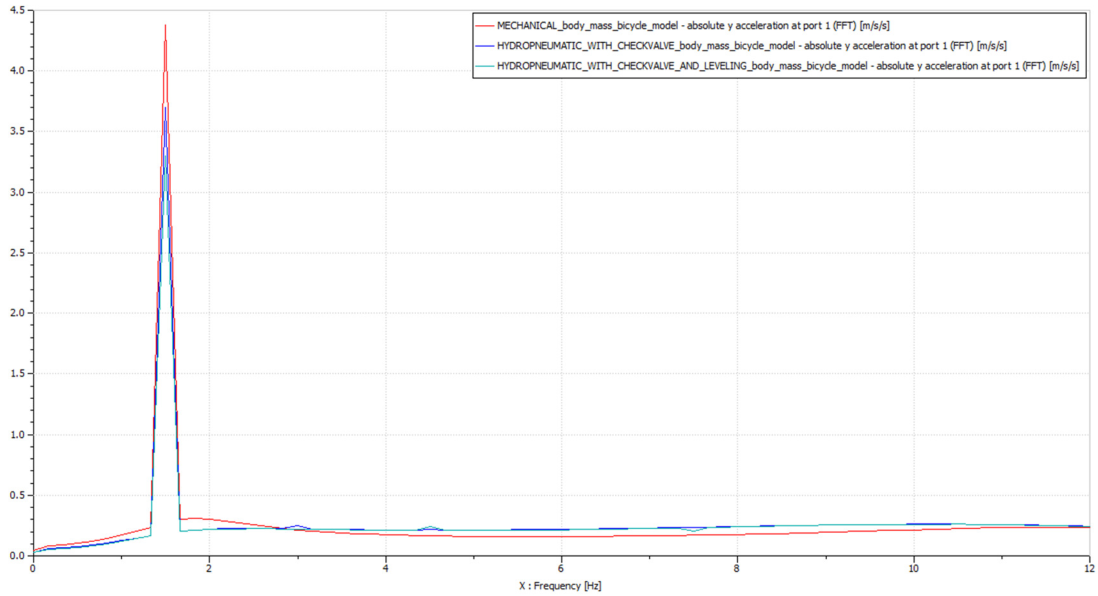

Figure 5.13 Body acceleration response and FFT to sine excitation signals

5.4 Unladen and laden cases for tuning the suspensions

Mechanical suspension is required to be tuned for the laden case. This is because it will have lower natural frequencies when laden and will have lower damping ratio. Contrary to mechanical suspensions, hydropneumatic suspensions can be tuned for the unladen case and still perform around the tuned range when laden thanks to the leveling system yet providing a more comfortable ride due to the softer springs. Parameters are detected for this scenario and are presented.

5.4.1 Main aspects of the tuning case:

- The damping coefficients are set to 1000 N/m/s for both systems
- Body masses are 250 kg for unladen and 400 kg for the laden case
- Tuned frequency is 1.3 Hz and this can be achieved at the laden case for the mechanical and at the unladen case for the hydropneumatic.
- Corresponding spring rates are 26688 N/m for the mechanical spring and 16683 N/m for the hydropneumatic spring, thus the hydropneumatic spring is softer at the same tuning frequency

5.4.2 Tuning parameters for the mechanical suspension

Since the damping ratio has to be around D=0.25 for comfort and the system natural frequency has to be around 1.3 Hz, the following parameters are set for the mechanical suspension.

- The unladen mass is 250 kg while the laden mass will be 400 kg.
- Damping coefficient is constant regardless of the load and is 1000 N/m/s.
- Spring rate is constant regardless of the load and is 26688 N/m.

These parameters enable the natural frequency of 1.6 Hz for the unladen case and 1.3 Hz for the laden case.

5.4.3 Tuning parameters for the hydropneumatic suspension

Since the damping ratio has to be around D=0.25 for comfort and the system natural frequency has to be around 1.3 Hz, the following parameters are set for the hydropneumatic suspension.

- The unladen mass is 250 kg while the laden mass will be 400 kg.
- Damping coefficient is set for the unladen case as 1000 N/m/s.
- Spring rate is set for the unladen case as 26688 N/m.

These parameters enable the natural frequency of 1.6 Hz for the unladen case and 1.3 Hz for the laden case.

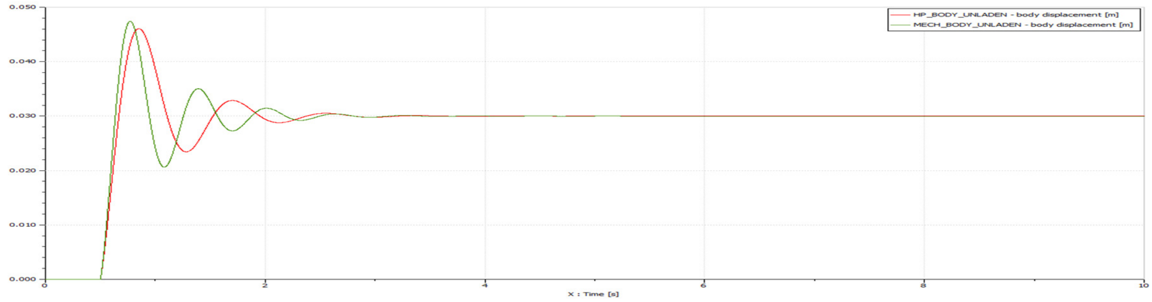

Figure 5.14 Unladen body displacement response to a step input

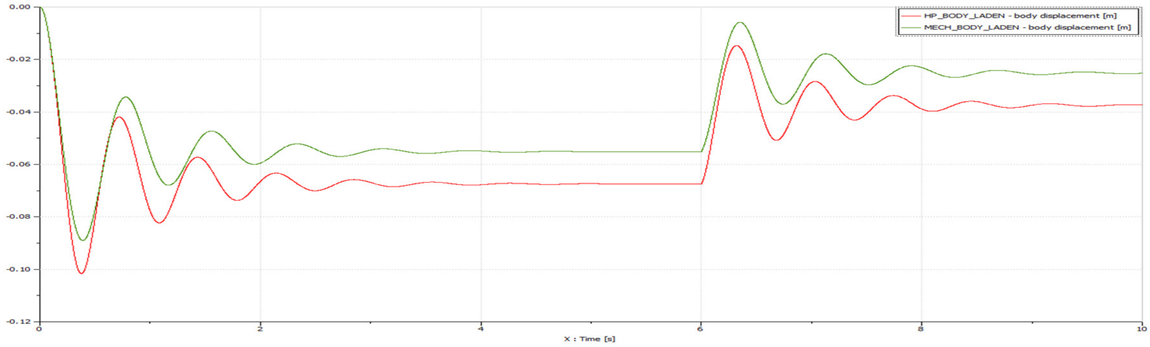

Figure 5.15 Laden body displacement response to a step input

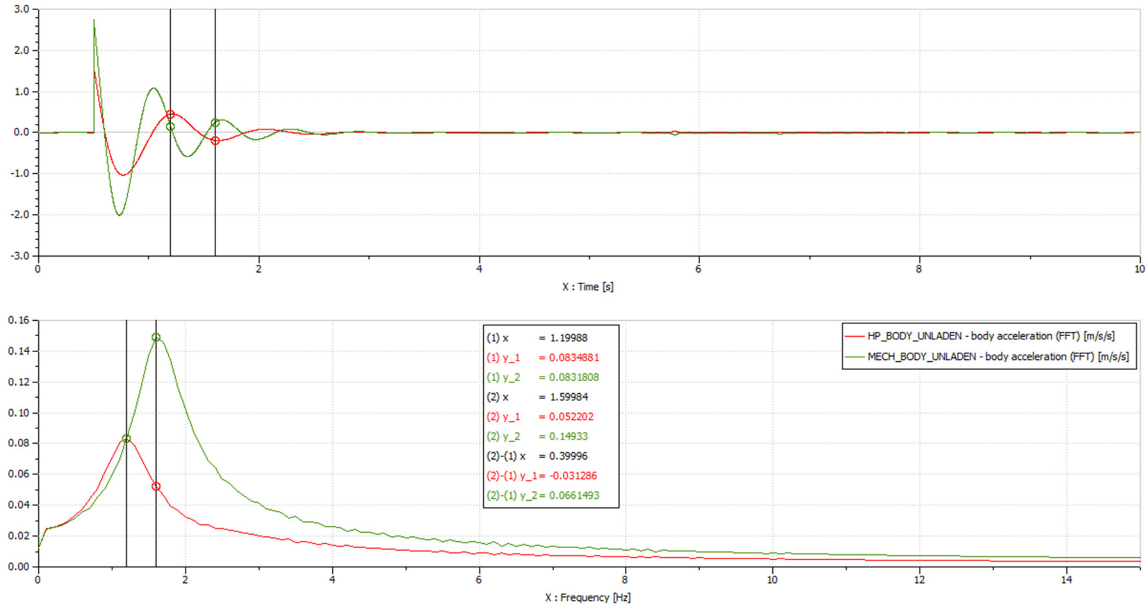

Figure 5.16 Unladen body acceleration response and FFT to sine excitation signals

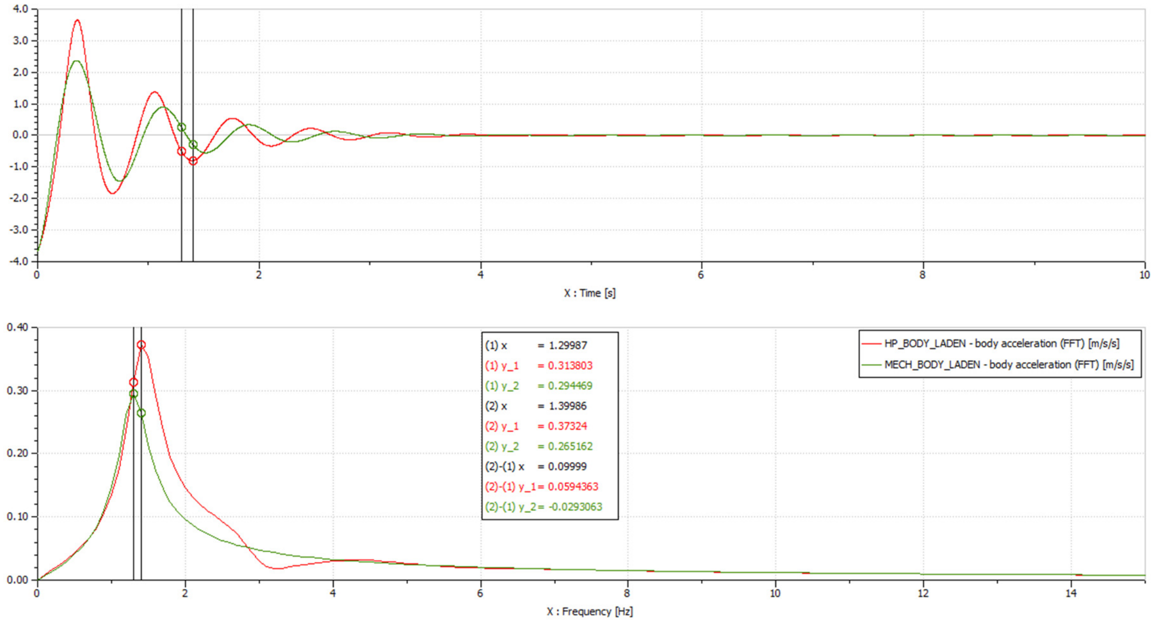

Figure 5.17 Laden body acceleration response and FFT to sine excitation signals

6. RESULTS AND FUTURE WORK

In this thesis, basic shock and vibration characteristics of the hydropneumatic suspension system are analyzed via mathematical modeling and computer simulation. The developed models are simulating a mechanical suspension and the hydropneumatic suspension with a quarter car model and a bicycle model. Leveling function is employed for the hydropneumatic system, as well as the communicating model. Transient response characteristics of the hydropneumatic suspension are evaulated for step input and sine input.

The outlines of the achieved results can be listed as follows;

Main elements of the hydropneumatic suspension are analyzed considering the basic theoretical background. Suspension parameters that have conventional acceptance have been applied for both unladen and laden cases, simulations are run for different scenarios and the upsides of the hyropneumatic suspension are thus pointed out.

Hydropneumatic and mechanical suspensions having the same damping and stiffness coefficients are compared and it is shown how the correlation changes by changing the damping elements and level control mechanisms.

Hydrodynamical damping model can be detailed and adaptive damping elements can be added to the system in order to employ the best damping for a specific case.

Hydraulic pump design can be studied in more detail and a central hydraulic circuit can be constructed for the full vehicle model so that the axles can communicate and braking, maneuvering, pitch and roll motions can be analyzed.

Hydraulic design alternatives that will enable the cancellation of mechanical stabilizer bars can be studied.

Compression and expansion cycles can be analyzed seperately in order to determine and improve the weakness of the hydropneumatic suspension for the expansion cycle; thus the addition of a mechanical spring can be discussed.

Vibration isolation performance is evaluated in terms of transmissibility characteristics. The shock and vibration response characteristics of the hydropneumatic suspension system are compared to those of conventional suspension, employing linear spring and damping characteristics.

The writer was unable to verify this analytic model with an experimental model. This may result in severe differences than physical expectations, but is a standpoint of view to deeper analysis of hydropneumatic suspension models because the base model is successfully constructed for further tweaking.

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