Energy Harvesting Mathematical Model on Vehicle Shock Absorber by the Introduction of Compress Air Integration

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ABSTRACT

The objectives and the purpose of this paper are to acquire a mathematical model of energy gain, from car suspensions by incorporating a pneumatic cylinder, in order to produce compress air energy. With the current configuration of a passive suspension on most of a vehicle in the market today, the potential energy produces from the oscillating shock absorber can be retrieved and converted into useful energy. By the introduction of the pneumatic cylinder, a compressed air can be converted from the mechanical movement and transferred into DC electrical energy via pneumatic motor generator. The passive suspension poses an ability to generate a potential energy without compromising the handling and comfort ride, as the compensating dampener introduce by the pneumatic cylinder will be balanced against the existing dampening effect. The balancing of the suspension can be achieved by using a performance suspension which isusually used on a racing car. The performance suspension has the flexibility to balance or adjust the dampening and spring effect in order to compensate the dampening from the pneumatic cylinder. Hence, a different variable from the design can be equalized on the system in order to achieve the right tune of energy harvested, as well as sustaining the function of the shock absorber itself. In this paper, a mathematical model using Simulink and Matlab will be presented.

Keywords—*Shock Absorber Suspension, Energy Harvesting, Mathematical Model, Dynamic Air Compression.*

I. INTRODUCTION

AS the vehicle growth has grown to a substantial numbers over the years, the oil consumption for the transportation usage has also increased in a very significant amount. It is noted that the transportation energy using a fossil fuel is accounted at 36% of the total energy consumption with the trend stillconsistently moving upwards [21]. Based on Velinsky[25], the decapitated energy in the shock absorber system is related to the road roughness, vehicle speed, suspension stiffness and coefficient of damping itself.The challenges that facing the world with regards to this matter have been studied by many researchers in order to make a contribution, by minimizing the actual energy being used and eventual reducing the emission to the atmosphere. Typically, only around 10%-16% of the fuel energy consumption is being used to drive the vehicle, which is to overcome the resistance from road friction and air drags. With the vast increase of EV (Electric vehicle) status development, the hybrid energy transport studies are quite encouraging. However, due to the high price of the vehicle batteries, limited distance of traveling and long charging time, the limitation does play an important role oncommunity acceptance. Hence, research effort have somehow redirecting the focus towards to the developments of energy harvesting generated from the kinetic energy as part of the source to reduce the cost.In one of the research, Hammad[22] have modeled and analyze a dissipated energy lost in a shock absorbing suspension. He has claims that 1.5 kW of energy were lost at the suspension alone. However, Hsu [24] claims only 400 Watts of power are dissipated from all four shock absorbers on the same condition. Hence a very potential energy available, which can be retrieve for a good used.

The basic theoretical concept for this research involved the design construction and mathematical model of the compress air shock absorber behavior, based upon linear dynamic pressure characteristic. The idea is to study the internal physics of the compress air excited by the oscillation cycle of the dampener, which later converted the produce thermodynamic energy into a useable harvested electrical energy.In another term, it is to determine the design requirements by establishing mathematical model of the system from the understanding of vibration control systemperspective. To apprehend the understanding into a bigger picture of the design and control, a simulation by software will be incorporated as part of the establishments. A knows Laws for such model using a dynamic principle are to be derived. This will lead to a mathematical simulation as well as the optimization to be fine tune for dampening values. The design parameter will be produced as part of the deliverable.

II. DESIGN CONFIGURATION

The design of suspension is based on two separate systems which integrated together. On one system, a normal performance suspension being used and calibrated to accommodate the dampening factor attached by another system. The other system is to integrate a performance suspension with a pneumatic cylinder which reacts as a dampener, as well as to generate a compress air to support the power generating system. The energy output to be calculated is based upon a benchmark of a typical pneumatic motor datasheet which will be elaborated. The calculated volume and pressure produce by the air compression can then be established by the power output from the datasheet of the motor.

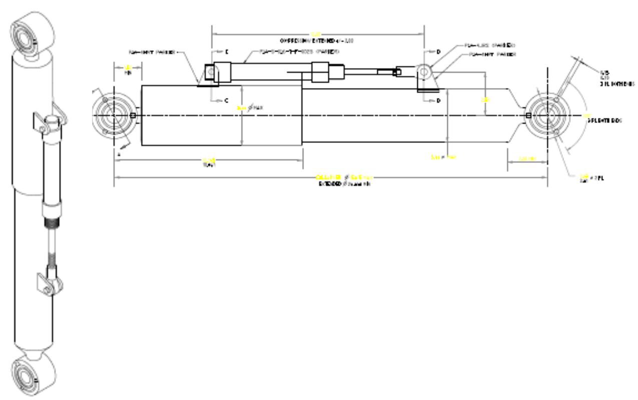


Figure 1: Conceptual design of pneumatic cylinder havesting energy suspension

The model tendto investigate the effective compressibility behavior, in order to prove that the waste heat energy from the conventional shock absorber can be utilized and harvested into a prominent and reliable usage. This model also will demonstrate the effectiveness of the steady flow air compression equation to stabilize the effect and balance between the dampening mode and excessive air release pressure, for generatingthe power. The understanding and mathematical concept produce will showsthe efficiency of energy utilization based upon the compression level and standard equation of the thermodynamic flow according to the pressure drop across the dampener chamber. The flow dynamic discharged coefficients will be assumes in this mathematical model.

Another assumption by the mathematical model is the correction factor of the flow restriction forces which contributes from the leak efficiency, check-valve/line flow restriction, and spring stiffness.The model for the suspension dampener will also discuss the frequency effect on hysteresis.The unstable compression of air can increase the hysteresis due to the added phase loss happening at the same input frequency. A consistence air pressure in the dampener chamber is important, plus the air frictions which also increase the hysteresis near the zero velocity regions. In this model, the total flow comprised of actuating piston, check valve, pressure regulator including the piston leakage flow that give the resistance will be assume upon each stage of flow based on the pressure drop across the compression dampener chamber and compress air receiver. The compression air chamber pressure, $P_{chamber}$, and pressure at gas receiver storage, $P_{storage}$, are balance by the pressure regulator.

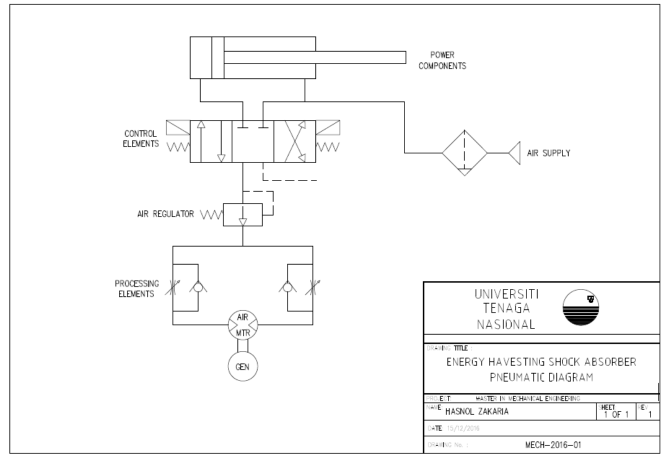


Figure 2: Pneumatic circuit diagram of pneumatic cylinder

Even though the concept of the suspension proposed is not so complicated, the design itself represents quite a complex with non-linear characteristics that produce a challenge, whichthe mathematical model can be difficult to measure accurately. However, due to the limitation on budget and time frame given, the actual prototype is not being able to be constructed for testing. The mathematical model is used to evaluate the predictive dynamic response.

The calculation will be divided into 4 different categories;

a. Performance indicator of shock absorber based on the integrated pneumatic impulse concept.

b. Mathematical model of shock absorber incorporated with the pneumatic generation cylinder.

c. Simulation of design based cyclic energy performance using Simulink and Mathlab.

d. Projecting the power produce based on the volumetric compress air vs. pressure in accordance to the pneumatic motor datasheet by OEM.

The suitable suspension which can be used to further developing this study in the future is by using a performance suspension that can be adjusted and often used by racing drivers. The adjustable on both compression and rebound damping can benefit the suspension tolerance. It has the flexibility to nip precisely the dampening effect.

The calculation is based on the power of the compress air produced. The calculations in term of generating electricity need to be calculated, based on the pneumatic generator output, which not yet to be determined.

Based on the compression dampening force as mentioned above, the integration of the newly design absorber with pneumatic cylinder need to be elaborated. The basic, round cylinder type of pneumatic actuator consist of tube sealed by an end caps. As the load acting or enforce, the piston rod will react. In this design, a double-acting cylinder will be compressed on bothsideof the cylinders. The other advantage is it compact cylinder which can be fitted at a constraint space. It has a light and ability to absorb high load.

III. System Power Output Modelling

In this section, it will focus on the compression air at the pneumatic cylinder oscillation. At this particular mechanism, the component will relate between the force and velocities. We can look at the Boyle's Law which reflects the relationship between pressure and volume of a sealed off gas, assuming that the temperature will remains constant. The type of cylinder used in this project is the single-acting cylinder spring without any spring return. Hence, the effect of opposing force of pulling/pushing as the stoke react can be neglected. However, the principle of the suspension for this project is not to hold the pressure as applied on industrial application as its stroke. The compressed air in the system will be relief at a certain pressure calculated and later to be converted into kinetic energy. Hence, the Boyle's Law will not be considered to determine the volumetric flow of the compress air for this mathematical model.

Basically, the vibration system signifies of a components configuration which corresponds and reacts towards each other's. In order to establish the equation of motion, it is important to understand the characteristics of excitation-response. The component can generally divide into three classes, which are the components of forces that react due to "proportional to displacement" (example of spring), "proportional to acceleration"(example of rigid mass translation) and "proportional to velocities" (example of dashpot). The component itself will store, release potential and dissipate energy.

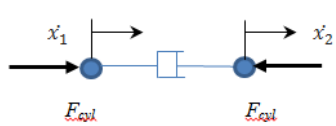


Figure 3: Damper under a Compression Force

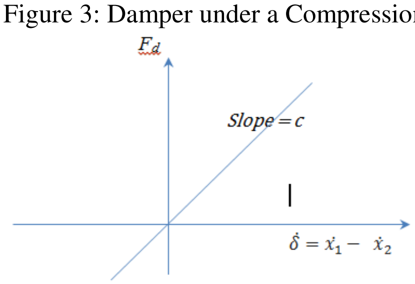


Figure 4: Force versus Separation Velocity

Based on simplified diagram above, we can describe that the configuration of a pneumatic cylinder dampener is more relates to forces and velocities. This is a type of dashpot or viscous damper which structures from a moving piston movement inside a cylinder that filled with compress air. By the configuration of the design proposed, the compress air is been permitted to flow or channel out of the system due to the force compression acting by the piston. The force F_{cyl} by the air compression on one side of the piston were balanced out by the force (F_{cyl}) from the oscillating force of the suspension. We can also implicit that forces (F_{cyl}) is linear to velocities ($\dot{\delta}$), whereby, $\dot{\delta} = \dot{x}_1 - \dot{x}_2$. The simplifying term for relationship between for (F_{cyl}) and velocity difference ($\dot{\delta}$) can be descript as

$$F_{cyl} = c\dot{\delta} = c(\dot{x}_1 - \dot{x}_2)$$

The proportionate constant c as shownon above graph above, whereby, F_{cyl} versus $\dot{\delta}$ is known as *coefficient of viscous damping*. Hence, the viscous damper can be identified as coefficient c. The constant c unit will be known as newton · second per meter (N · s/m).

Here we also need to derive an expression for dampening constant of a piston cylinder dashpot. The dashpot damping constant can be identified by identifying the equation of the air viscous fluid flow and the degree of air flow equation. Looking at the diagram above, which reflects on the parameter such as piston of the diameter (D) and displacement (d_a), which move at the velocity of ($\dot{x}$) filled with an air at certain air viscous damping coefficient ($\dot{\delta}$).

The first equation required is to obtain the velocity ($\dot{x}$) of the suspension. The fundamental motion of the vehicle suspension system can be defined as a sinusoidal wave, based on the excitation of the road profile. Hence, the input of the velocity ($\dot{x}$) can be defined as;

$$\dot{x} = \dot{x}_{\text{amp}} \sin\ (2\pi f t)$$

Whereby the velocity $\dot{x}_{amp}$amplitude is;

$$\dot{x}_{amp} = 2\pi f d$$

Where,

f = Frequency

d = Maximum displacement of the piston

The effective displacement d_a of the piston can be demonstrated as;

$$d_a = \int \dot{x}_0\, dt = d \sin(2\pi f t)$$

Based on the research conducted by Pei Sheng Zheng [16], the average dissipative power for mid size cars with ISO class B (good), C (average) and D (poor) categories of shock absorber have a vertical velocity ($\dot{x}_{\text{amp}}$) of 0.15 – 0.4 m/s relative motion dampening at the average displacement of (d_a) 0.25 -0.30m. The typical dampening force is estimated around 1400 Ns/m. The comparison of the results will be produced and analyzed.

Using the information calculated above, the flow rate of air compression (Q) can be translated into equation as following:

$$Q = A_D\,\dot{x}, when\ v > 0$$

Where,

Q = Air flow rate in compression chamber

$\dot{x}$ = displacement velocity

r= Radius of Cylinder Bore = 31.5 mm

Hence,

$$A_D = \pi r^2 = (\pi)(0.0315)^2\, m = 0.00312\, m^2$$

$$Q = 0.00312\, m^2 \left(0.4\ \frac{m^2}{s}\right) = 0.00125\frac{m^3}{s} = 4.5\frac{m^3}{hr} = 1.25\,\frac{l}{s}$$

The area of the A_D is known based on the datasheet given for JOUCOMATIC's double acting pneumatic cylinder model CIS 63 NA 100 - DM. It is around cylinders type component which has bore size of 63.0 mm (2.5") in diameter.

Figure 5: Pneumatic Double Acting Cylinder

Depending on the road profiles, which can be defined in term of frequency (f) and vertical velocity $\dot{x}$(t), an optimize volumetric air compressor can be obtain through Simulink.

IV. DYNAMIC FLOW

One of the important criterions determines the force output of the pneumatic cylinder is the air pressure generated and the bore size. In an industrial application, whereby the pneumatic cylinder is used to actuate a load by an air pressure acting upon the cylinder, the amount of force is twice the load required. In addition, the force also needs to compensate the friction.

One of the advantages using the industrial pneumatic cylinder is due to its ability for fast-cycling action during suction and exhaust of the equipment with a higher compression ratio at a short cyclic displacement. When calculating the requirement of the air consumption of the single acting cylinder, it is crucial to take consideration of the required air flow based on the bore area, bore length and as well as the cycle rate isolation of the system.

If we look at the normal application of pneumatic cylinder whereby the air is required to move a load, the process engineer need to calculate the cylinder force that need to be multiplied by the effective piston area by the working pressure.

$$F_{cyl} = \frac{(\pi.P.D^2)}{4}$$

Where,

F_{cyl} = Force exerted (N)

D = cylinder bore area (m)

P = Pressure (N/m^2)

In this project, the required pneumatic pump to be used is requiring a pressure at minimum of 4.2 bar (420 KN/m^2) at certain amount volumetric flow. Hence the required F_{cyl1} (force) to be exerted by the spring cylinder is;

$$F_{cyl} = \frac{(\pi.P.D^2)}{4} = \frac{\left(\pi\,.420000\,\left({}^{KN}/_{m^2}\right)0.0315\,^2(m^2)\right)}{4} = 3{,}308\,N$$

The force absorb is react against the spring motion of the suspension. Hence, the force (F_{cyl}) calculated above function as an additional dampener to the system. The absorption of the vibration from the road profile being additional dampens aside from the suspension used.

It is also known that the damping coefficient (c) is a ratio of the force (F_{cyl}) over the velocity ($\dot{x}_{amp}$).

$$c = \frac{F_{cyl}}{\dot{x}_{amp}}$$

Hence,

$$c = \frac{F_{cyl}}{\dot{x}_{amp}} = \frac{3308\ (N)}{0.3\ (m/s)} = 11.025\ \frac{KNs}{m}$$

The force of the pneumatic dampening $F_{cyl}$is taken from equation calculated above and the velocity ($\dot{x}_{amp}$) is based on data taken from Pei Sheng Zheng [16].

Hence, we can then calculate *coefficient of viscous damping* ($\dot{\delta}$).

$$F_{cyl} = c\dot{\delta} = c(\dot{x}_1 - \dot{x}_2)$$

Whereby,

$$\dot{\delta} = \frac{F_{cyl}}{c} = \frac{11{,}025\,(Ns/m)}{19{,}946\,(Ns/m)} = 0.55$$

It's being noted that the power of the piston oscillation with inline to the velocity $|\dot{x}_{amp}\ (t)|$ with respect to time. The outcome of the vibration can lead and react as dampening force as well as releasing dynamic flow power output.

With the known amount of force (F_{cyl}) acting upon the compression of the pneumatic cylinder, the proficiency of the power generated (P) can be shown as per following:

$$P = F_{cyl}.A_D|\dot{x}_{amp}(t)|$$

$$P = 3{,}308\left(\frac{Ns}{m}\right)0.0351(m)0.3\left(\frac{m}{s}\right) = 35\,watt$$

In addition, the air consumption for the application needs to be calculated. In principle, the air intakes were consumed from atmosphere via air filter. It is importance that adequate capacity of air supplies under worst-case scenario being consumed, in order to avoid any air starvations that reflect the performance of the whole system.

V. Calculation of Dampening System

In this section, we need to compare the normal passive suspension system reaction with the design incorporating the pneumatic compression air. The objective is to understand the suspension reaction under normal condition in comparison to the project design system. The analyst is to apprehend the ride performance of the suspension with two degree of freedom.

By a basic design of a **normal passive suspension** system is shown on the below diagram:

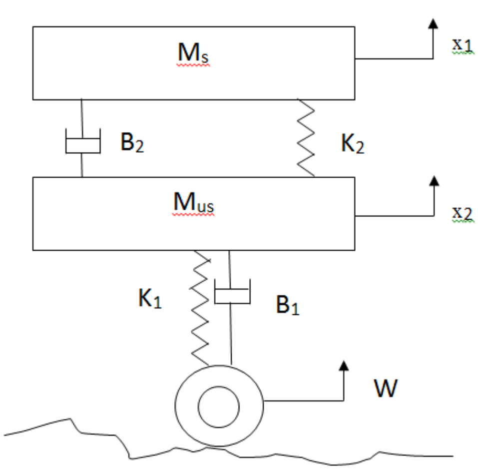


Figure 6 : Passive Suspension of two degree of freedom

excitation system

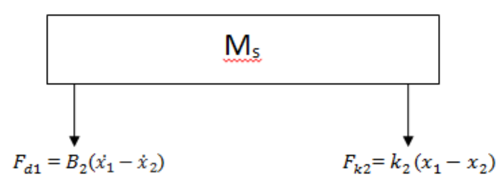
Whereby,

- B_1 = Dampening Coefficient of Tire (N/m)
- K_1 = Radial Tire Stiffness (N/m)
- K_2 = Suspension Stiffness (N/m)
- M_s= Sprung Mass (Kg)
- M_{us} = Un-Sprung Mass (Kg)
- Z_s= Maximum Displacement of Sprung Mass (m)
- Z_{us}= Maximum Displacement of Un-Sprung Mass (m)
- Z_r= Maximum Displacement of Tire on the Road (m)
- W= Road Excitation (m)

In general, the shock absorber system varies between one manufacturer to another, due to variety vehicle model available in the market. Depending on the concept of the model based on functionality, such as off-road vehicles, heavy duties vehicles, normal sedan passenger vehicles, etc., it is a known design that are required to compensate the road unevenness oscillations transmitted to the axles.

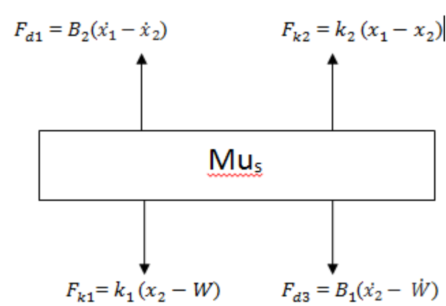
Based on typical normal suspension system, a mathematical model will be analyzed. The diagram has shown that the sprung mass of M_s is representing the body of the vehicle and the M_{us} represent the un-sprung mass of the axle and wheel. At this point, we can only consider the mass kinetic movement on vertical axis without considering the rotational movement reaction.

Based on the two degree of freedom, we can conclude that the equations of motion can be achieved using a Newton's second law for both of the masses (M_{san} andM_{us}).



$$\sum F_s = -F_{d1} - F_{k2} = M_s\ddot{x}_1$$

$$M_s\ddot{x}_1 + B_2(\dot{x}_1 - \dot{x}_2) + k_2(x_1 - x_2) = 0$$



$$\sum F_{ms} = F_{d1} + F_{k2} - F_{k1} - F_{d3} = M_s\ddot{x}_2$$

$$M_{us}\ddot{x}_2 - B_2(\dot{x}_1 - \dot{x}_2) - k_2(x_1 - x_2) + k_1(x_2 - W) + B_1(\dot{x}_2 - \dot{W}) = 0$$

The W reacts as a vertical movement based on the road profile. The x_2 and x_1 also respectively actuate at a vertical displacement in respect to W. However, since the irregularities of the road is difficult to be measured $(x_2 - W)$ and the deformation (dampening effect) of the tire $(\dot{x_2} - \dot{W})$ is quite subjective, hence, it will be negligible. The cyclic oscillation of the dampener can be translated based on Newton's second law 2-DOF base motion of equation:

$$M_s\ddot{x}_1 + B_2(\dot{x_1} - \dot{x_2}) + k_2(x_1 - x_2) = U$$

$$M_{us}\ddot{x}_2 - B_2(\dot{x_1} - \dot{x_2}) - k_2(x_1 - x_2) + k_1(x_2 - W) + B_1(\dot{x_2} - \dot{W}) = U$$

With regards to the integrated compression air shock absorber propose, it is installed directly at the body of the car frame and connected to the support frame of the vehicle itself. By a separate system of conventional shock absorber and the generating tube of the compress air suspension integrated together, the adjustment of the shock absorber spring has been re-design to be more elastic (a lower k, constant value). This is to compensate the dampening effect, generated by the cyclic shock vibration, which will be absorbed by both shock absorber and compress air tube.

The idea for this concept is for the compress air tube to be the primary part with a function to absorb the vibration and the conventional shock suspension to react as a secondary vibration absorbing mechanism. The configuration of the proposed design is using a two-tube mass system. The integration of a conventional hydraulic cylinder with a parallel connection to pneumatic actuator works as a pneumatic air generator as well as a compensator.

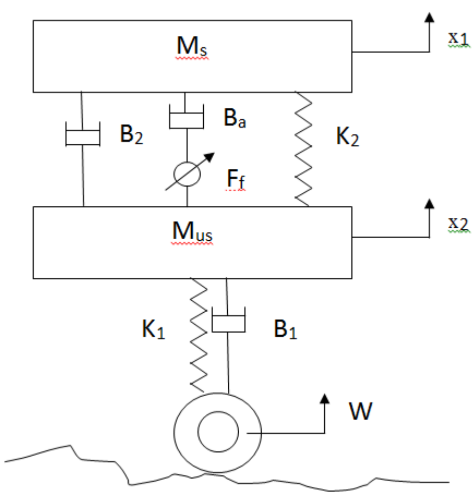


Figure 7 : Two degree of freedom excitation system

Whereby,

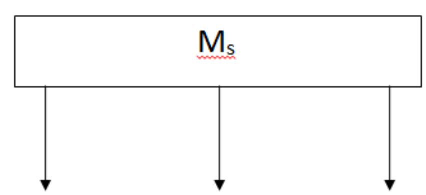
F_f = Pneumatic Friction Force (N/m)
B_a = Dampening Force by Compress Air (N/m)
B_1 = Dampening Coefficient of Tire (N/m)
K_1 = Radial Tire Stiffness (N/m)
K_2 = Suspension Stiffness (N/m)
M_s= Sprung Mass (Kg)
M_{us} = Un-Sprung Mass (Kg)
Z_s= Maximum Displacement of Sprung Mass (m)
Z_{us}= Maximum Displacement of Un-Sprung Mass (m)
Z_r= Maximum Displacement of Tire on the Road (m)
W= Road Excitation (m)

Using the load profile information from other researcher, the vibration disturbance profile reaction data can be used to configure the correct spring load constant and to regulate the optimum pneumatic flow-rate adjustment to this model of configuration design. A two degree-of-freedom (2-DOF) mass spring damper system has being modeled to simulate the dynamic function of the suspension. This will lead to the desire control reaction pace and lead to the estimated power generation produce by the system itself.

The dampener B_a has the function not only as a secondary dampener but also act as an air compression generator. It is employed not just to dampen the vehicle cyclic vibration but more importantly to adjust the air flow rate for the purpose of controlling the displacement, as well as providing supplied compress air to the receiver tank.

At the same time, a passive suspension of the system with a linear damping coefficient B_2, react at the proportion together with B_a to counter the equivalent stiffness of spring, K_2. The compression air which part of its function is to create an actuating force of dampening effect, that sprung and un-sprung the masses.The F_f represents the pneumatic decompression friction occurs every time it happens during the isolation cycles. However, since the friction is very low, the value will be negligible.

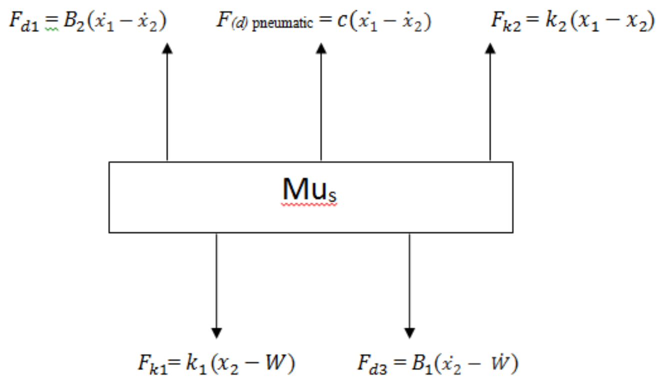
In this system, let x_1=0, x_2=0 as the system is in static equilibrium.



$F_{d1} = B_2(\dot{x_1} - \dot{x_2})$ $F_{(d)\ \mathrm{pneumatic}} = c(\dot{x_1} - \dot{x_2})$ $F_{k2} = k_2(x_1 - x_2)$

$$\sum F_s = -F_{d1} - F_{(d)\ \mathrm{pneumatic}} - F_{k2} = M_s\ddot{x}_1$$

$$M_s\ddot{x}_1 + k_2(x_1 - x_2) + B_2(\dot{x_1} - \dot{x_2}) + c(\dot{x_1} - \dot{x_2}) = 0$$



$$\sum F_{ms} = F_{d1} + F_{(d)\ \text{pneumatic}} + F_{k2} - F_{k1} - F_{d3} = M_s\ddot{x}_2$$

$$M_{us}\ddot{x}_2 - B_2(\dot{x}_1 - \dot{x}_2) - c(\dot{x}_1 - \dot{x}_2) - k_2(x_1 - x_2) + k_1(x_2 -) + B_1(\dot{x}_2 - \dot{W}) = 0$$

Again, since the irregularities of the road is difficult to be measured $(x_2 - W)$ and the deformation (dampening effect) of the tire $(\dot{x}_2 - \dot{W})$ is quite subjective, hence, it will be based on assumption. The cyclic oscillation of the dampener can be translated based on Newton's second law 2-DOF base motion of equation:

$$M_s\ddot{x}_1 + k_2(x_1 - x_2) + B_2(\dot{x}_1 - \dot{x}_2) + c(\dot{x}_1 - \dot{x}_2) = U$$

$$M_{us}\ddot{x}_2 - B_2(\dot{x}_1 - \dot{x}_2) - c(\dot{x}_1 - \dot{x}_2) - k_2(x_1 - x_2) + k_1(x_2 - W) + B_1(\dot{x}_2 - \dot{W}) = -U$$

VI. SYSTEM EQUILIBRIUM USING MATLABSIMULINK SIMULATION

By using the first part of the equations, Matlab Simulink software will be used to solve the system equilibrium. The technique is using MATLAB/SIMULINK, which model the mass (spring and un-sprung) of inertia and forces of gravity. It is to determine the damping physical characteristics of the system introduce by the cyclic vibration of the shock absorber at diverse damping coefficient and excitation frequencies. By understanding the ideal working data (e.g. spring coefficient), the understanding of the equipment concept can be elaborated into a better modeling design. Based on the formula above, most of the values can be retrieved based on typical passenger car datasheet. For example, the value of the damping ratio based by GadiaUtsav D. et al.,[14]the optimal damping ratio between 0.2 to 0.4 and at an average of 0.32. The gross weight of a car (M_s) are in the average of 1250 Kg. The pneumatic friction force (F_f) can be neglected as the design of the pneumatic cylinder does come with a very low resistance when stroking. The parameters in this project which can be optimized are the dampening factor by air compression (B_a), suspension stiffness that uses the performance shock absorber (k_2), and dampening factor of the suspension (B_a). These values can be adjusted in order to get the best/ optimal suspension frequencies and at the same time to harvest the energy from the compress air to drive the pneumatic motor.

In comparison to the passenger car application, the values are much different but have a high compression pressure value at the different rate of damping force. With equation of the of the free body diagram mentioned on Figure: 7, a block diagram can be shown as a mathematical equation of the mechanical dynamic flow motion based on compression ratio and parameter as shown below.

Motion ratio of quarter car suspension system	
Un-Sprung Mass (M_{us})	40 Kg
Sprung Mass (M_s)	300 Kg
Suspension Stiffness (K_2)	15,000 N/m
Vertical Stiffness of Tire (K_1)	150,000 N/m
Dampening Factor of Suspension (B_2)	1000 Ns/m
Dampening Factor of Tire (B_1)	150 N/m
Max Displacement of Tire on the Road (W)	25 mm
Max. Displacement of Sprung Mass (x_1)	25 mm
Tire Velocity ($\ddot{W}$)	30 mm/sec
Vertical Suspension Velocity ($\ddot{x}_2$)	30 mm/sec

Table 1: Parameters to be used for Simulink Simulation Data, Based on Conventional Passive Suspension System [36]

Assuming the acceleration at the ($\ddot{x}_2$) is already identified from practical data which was used as input to the model from Lei Zuo and Pei-Sheng Zhang[16]as shown figure 8 and 9.

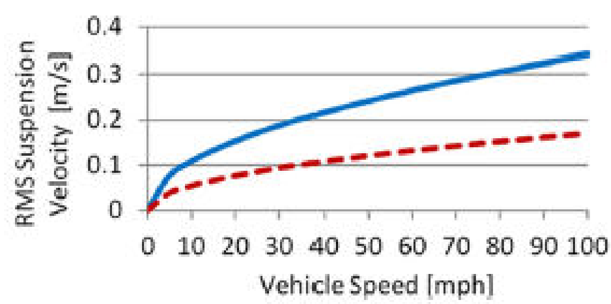


Figure8:Displacement analysis over vehicle speeds; on good road (dashed) and average road (solid) [10]

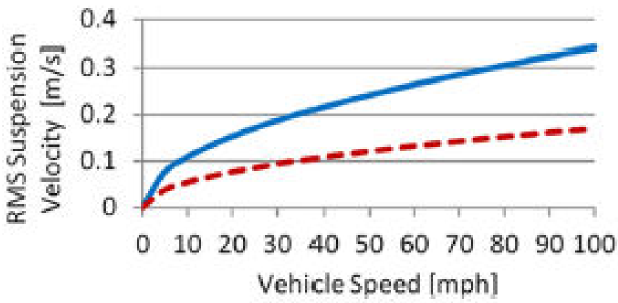


Figure 9 : Velocity analysis over vehicles speeds on good road and average road (solid) [10]

The dampening factor by compress air is based on the power out of the pneumatic compression pressure that is set to 6 barg. However, this value can be configured at different value in application as well as at the Simulink model in order to get the most optimum level of comfort ride.

The configuration of the suspension design parameter can differ from one application to another. As example of a normal passenger car compared to a race car, the spring and damper compression ratio can be very different in order to suit the application. In the racing car, the lightest damper was used, which can provide maximum stroke with a least amount of space consumption. Basically, the car spring need to be over dampers due to the proportional amount of damping versus the velocity. At high speed, the directional stability, steering characteristic and road handling need to suit the dampening value.

VII. SECOND ORDER EQUATION SYSTEM MODEL

From the previous table1shown earlier, the vertical suspension velocity was taken as an input to the model as part of the developing Simulink Matlab control system block diagram. The first Simulink block diagram to be elaborated is a normal passive suspension system. The simulation produce is to compare at later stage the equilibrium of the dampening normal suspension absorption comparatively to the incorporated pneumatic harvested suspension equilibrium.

Hence, the dampening reaction of the normal passive suspension result will be the benchmark in comparison to the proposed pneumatic harvested suspension. The integration of the dampening mode of the suspension with pneumatic compression on the new design should be more less the same as to reaction of the normal passive suspension. The new design will be based on this output. On a conventional passive suspension system, if the stiffness is too extreme, the system can be stable but the it will compromised the confort ride of the passanger. However, if it is too stiff, then the vehicle can become unstable, even if the comfort ride is achieved. The balance should be consider and the benchmark of the conventional suspension will be used to evaluate the new integrated pneumatic suspension configuration.

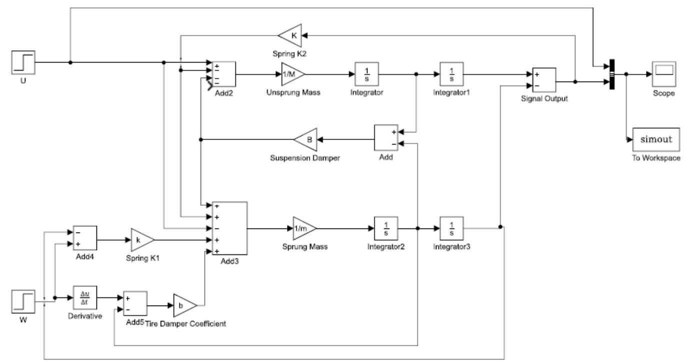


Figure 10: Discrete Block Functions of Matlab Simulink Block Diagram based on conventional passive suspension system

A simulation block diagram produce are based on the Matlab platform, in order to analyze the system configuration response specifically on displacement and acceleration. Knowing the reaction and response of the conventional passive suspension system produced the amplitude and dampening factor will then compared to the proposed system output of the pneumatic integration suspension at later stage.

The advantage of using the Simulink is the benefit of enhancing the data configuration for the simulation either by adjusting the damping coefficient or spring stiffness. The dynamic response of this mathematical modelling will be examined. Once the benchmark has been established, the second modelling will be configured in form of a pneumatic integration suspension unit. The proposed configuration will add the pneumatic dampening coefficient that will be balance out by decreasing the suspension coefficient as well as the un-sprung spring stiffness of the shock absorber. The output amplitude of the new configuration should react at the same manner as the conventional suspension which already established.

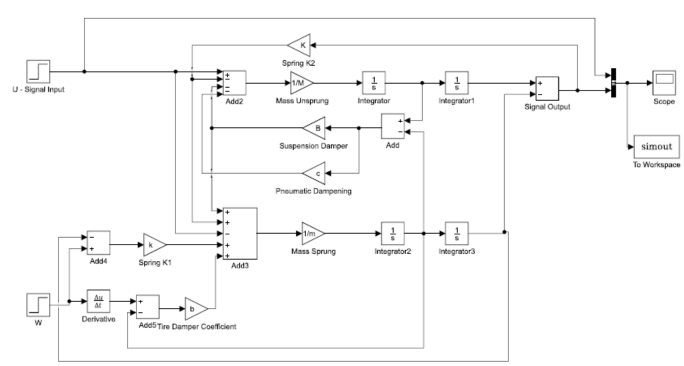


Figure 11: Discrete Block Functions of Matlab Simulink Block Diagram based on new configuration suspension integrated with pneumatic system

The value of the optimized spring stiffness and dampening coefficient have been evaluated based on the output performance tested using Simulink. The table of data for the new configuration is as shown below.

Motion ratio of quarter car suspension system	
Un-Sprung Mass (M_{us})	40 Kg
Sprung Mass (M_s)	300 Kg
Suspension Stiffness (K_2)	10,000 N/m
Vertical Stiffness of Tire (K_1)	150,000 N/m
Damp Coef of Pneumatic Cylinder (B_a)	0.55
Dampening Factor of Suspension (B_2)	800 Ns/m
Dampening Factor of Tire (B_1)	150 N/m
Damping Coef of Pneumatic Actuator (c)	400 Ns/m
Max Displacement of Tire on the Road (W)	25 mm
Max Displacement of Sprung Mass (x_1)	25 mm
Tire Velocity ($\ddot{W}$)	30 mm/sec
Vertical Suspension Velocity ($\ddot{x}_2$)	30 mm/sec

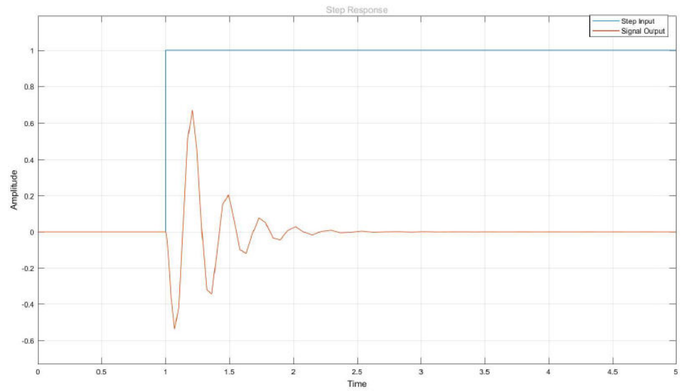
Table 2: Parameters to be used for Simulink Simulation Data, Based on Integrated Pneumatic Suspension System.

VIII. SIMULATION RESULTS

The mathematical formulations as stated will be compared in term of performance of a normal passive suspension configuration versus an integrated pneumatic dampener suspension. The exerted vibration dampening and efficiency of the effect has been achieved. The road commotion will be assumed and the performance of the suspension will be measured in term of quality ride and control.

The data and parameters of the simulation was taken from a research done by P. Sathishkumar et al.,[13] and Pei-Sheng Zhang [10]as a basis of the test. The sensible values of the dampening effect from the pneumatic configuration of the hysteresis models are based on the trial and error during the simulation. The goal is to achieve a minimal amplitude values for the shock absorber travel, tire deflection and the car acceleration.

Given by the data parameter (Table 1) of a normal passive suspension, the displacement of M_s (Sprung mass) transpires over a period of 2.5 seconds are based on the excitation signal produce starting from amplitude of zero. The maximum point of suspension sprung mass displacement is 0.06 m.



- Figure 12: Step Response of a Normal Passive Suspension System

Comparatively, the integrated pneumatic suspension has produced almost a similar pattern of amplitude response based on a new data configuration (Figure 13). The dampening coefficient of pneumatic cylinder (B_a), dampening factor of suspension (B_2), and suspension stiffness (K_2) has been adjusted to suit the similar outcome of a normal passive suspension system. The step response is as shown below.

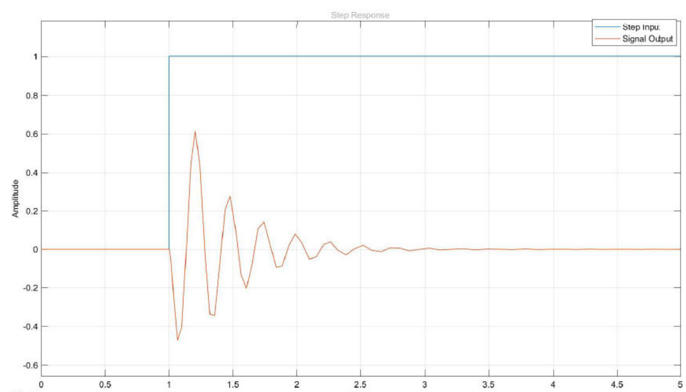


Figure 13: Step Response of a Pneumatic Integrated Suspension System

Based on data parameter (Table 2) of a pneumatic integrated suspension system, the displacement of M_s (Sprung mass) has transpires over a period of 3.0 seconds. The maximum point of suspension sprung mass displacement is 0.06 m. However, the dampening factor of the pneumatic coefficient is 400 Nm/s. The energy produce by the air compression will be calculated on a next section.

The energy harvested can be concluded based on the outcome of the volume and pressure produce by the suspension. The result established on chapter 3 has indicate that the volume produce at a nominal rate of 1.25 l/s and at a pressure of 4.2 barg.The performance of the pneumatic air motor is a dependent of the inlet pressure. The compress air of the suspension in this project is being regulated and control at the pressure of 4.2 barg. The air accumulator will be used to store the compress air before being introduced to the air motor. Hence, it provides a constant inlets pressure to support the system design.For the purpose of this project thesis, GAST air motor (Model 1AM-NCC-12) has been used to benchmark power produce by the suspension. It can be concluded that based on the pressure vs. compresses air consumption by the system, the speed will produce around 1000 RPM. The indication will lead to a power output of 40 watts.

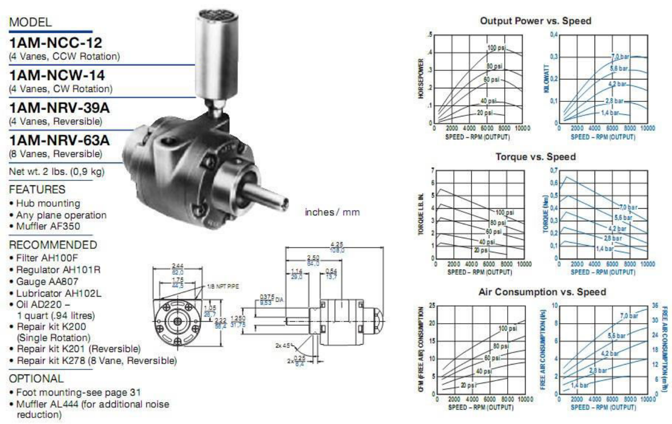


Figure 14: Product Data Sheet of GAST Air Motor Model 1AM

IX. Results and Summary

Simulink model has a capability to express the differential equation by a system. Simulink react to every pulse with a marginally underdamped output before return back to its equilibrium. On this project, Simulink translate the reaction of a bouncing vehicle. It can be noted that the first 2.5 seconds, the passengers will experience the uncomfortable ride due to higher oscillation amplitude but immediately decline. At this point, the comfort ride produce are at a nominal rate of amplitude. This is very important in order to ensure that the compressed air dampening will react as it supposed to function without compromising the suspension ability as a whole system.
The system does stabilize and the same time producing the volumetric compressed air required in order to produce the energy supply to the air motor. By compiling the calculation data produce below, the reference power output can be obtained.

No	Data Description	Parameter/Data
1	Air Flow rate	$1.25\,\frac{l}{s}$
2	Compressed Air Pressure	4.2 barg

Table **3**: Data Parameter of Air Compression

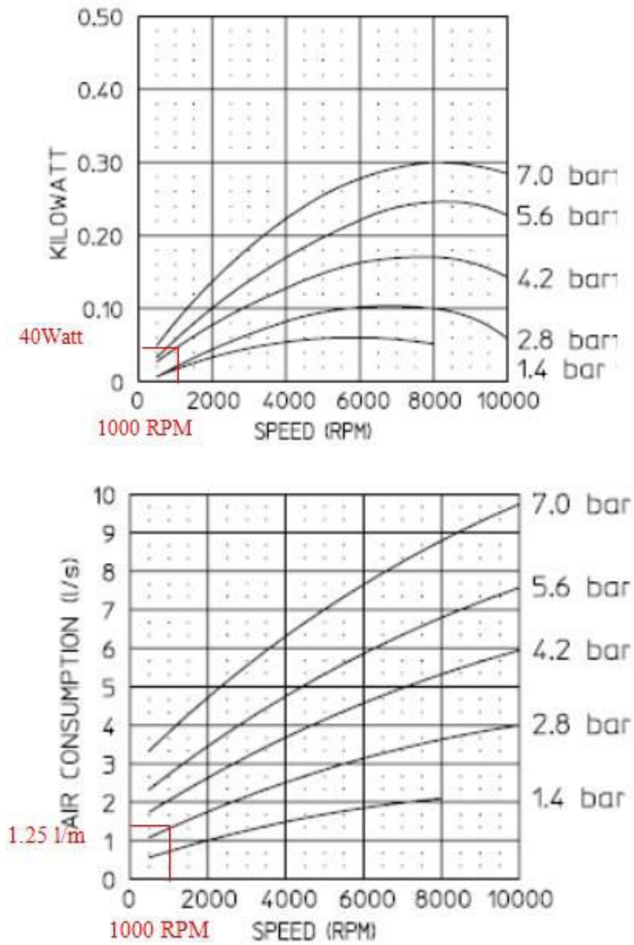


Figure 15: Air Consumption and Power of Air Motor

X. Conclusion

This chapter has elaborated the potential output performance of the harvested air compression integrated suspension system. By presenting the Simulink simulation, it has shown that the output performance of the design suspension will accommodate the required balance of comfort driving, which at the same time managed to have the desired capabilities to produce enough energy to be extracted. It is easy to be re-configures, displayed a very minimum risk and permit sensitive variation of analysis.

From the theoretical research being done, it can be concluded that the power output of 40 watts has been produced with an average suspension vertical velocity of 0.40 m/s. It also needs to be noted that the energy produced is based on quarter car model.

The model proposed is by integrating a standard suspension currently available in the market, which is a proven design in general. By combining the design with the pneumatic cylinder, it will not compromise the integrity of the basic shock absorber design that is within the code and standard of the ISO 43.040.50 (Transmission and Suspension). Hence, the strength of the structural analysis will be maintained in general. By balancing the dampening factor of the incorporated compressed air with the modified existing spring/oil dampening values, the design should accommodate the vibration response as required.

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