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Force line improvement analysis of a coil spring in macpherson strut using Altair Hyperworks

C.W. Lee[1], S.N. Sulaiman[1,2*], M.H. Abdul Rahman[1,2], A.A.M. Damanhuri[1]

[1)] Faculty of Engineering Technology, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[2)] Centre for Advanced Research on Energy, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia

*Corresponding e-mail: safulnaim@utem.edu.my

Keywords: Coil spring; macpherson strut; side force

ABSTRACT – One of the contributive factors to the worn damper in a Macpherson strut is the continuous side force exerted by the coil spring. In order to study and overcome the problem, the behavior of the coil spring during different loading conditions was simulated using Altair RADIOSS to obtain the spring force line. A new spring shape is proposed by morphing the original spring without changing other geometrical characteristics. Both spring designs are compared in strut level using simulation. As a result of the evaluation, the morphed spring design shows a side force reduction with a better spring force line.

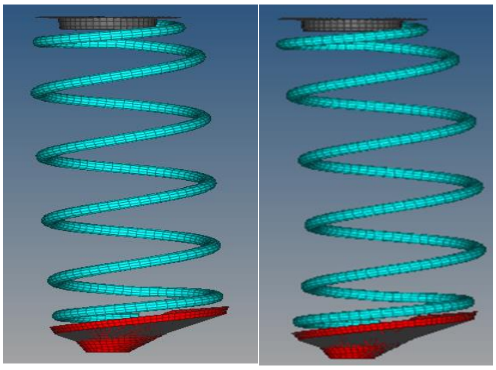
1. INTRODUCTION

Macpherson strut is the most common suspension system assembled in passenger front vehicles. The main reason to the popular use of Macpherson strut is due to its simpler structure compared to other common vehicle suspension system. This results in a low manufacturing cost and a low service cost for the consumer [1]. The main moving component in the suspension is the strut which consisting of a damper and a spring [2]. Despite of the advantages, the nature of the strut design creates an inevitable side force exerted at the top of the damper. This side force reduces the durability of the damper rod because it increases the inner friction between damper parts [3]. The conventional method used to minimize the side force is to incline the spring to a certain inclination angle to get a favourable spring force line but this is difficult to be implemented due to the limited installation space of the strut [4]. The modelling of the suspension system and the analytical process to find the best spring inclination angle was developed by [5] using multibody dynamics (MBD) analysis. For the same purpose, a mathematical model to calculate the spring force line using vector approach was introduced by [6]. Both shows that the spring plays an important role in the reduction of side force. In this study, the spring force line is to be determined using finite element (FE) method and the effect of spring shape towards the side force reduction of the Macpherson front suspension module is to be studied.

2. METHODOLOGY

There are two levels of approach in order to conduct the analytical evaluation on the coil spring namely the strut and suspension levels. In the strut level, an FE model consists of coil spring and both the upper and lower seats was constructed using the Altair Hyperworks software to represent the front strut of a vehicle. Before the FE model is to be created, all the required spring design parameters was measured and the hardpoints of the strut components in the vehicle were determined. The FE model was validated experimentally by comparing the spring stiffness.

The shape of the spring was altered by morphing the FE model. The morphing process involved a shift of the centre point of the coil spring axis along the side force axis by 20mm. Subsequently the spring mesh will force to adapt towards the new spring shape and it is said to be morphed. Both spring models are represented in Figure 1.



(a) (b)

Figure 1 Finite element spring design models: (a) original spring design and (b) Morphed spring design.

The behaviour of the spring during vehicle driving, namely full rebound, unladen and full bump position was simulated by imposing displacement value applied on the lower seat to compress the spring FE model. Altair RADIOSS solver was used to calculate, among other things, the value of forces and moments resulting from the simulation. These values were then used to calculate the spring force line at three different compression lengths according to the equations from [3]. The deviation of the force line from the vertical axis were evaluated and compared to conclude improvement made by the new morphed spring.

3. RESULTS AND DISCUSSION

The results of the spring force lines for both spring designs during full bump position are shown in Figure 2. The tangent trigonometric function, tan θ is applied to determine the deviation of spring force line from the vertical axis. At the full bump length of 142.4mm, the original spring force line deviates at 7.90^0 from the vertical axis compared to the deviation of new spring force line of only 1.15^0. This shows that the new spring force line is improved from the original design by offering a smaller deviation. When the deviation of spring force line is reduced, this will diminish uneven loading around each coil, causing the coil to have smaller warp, or sustain the form of true coil spring. Any spring warping will increase the maximum stress in the spring.

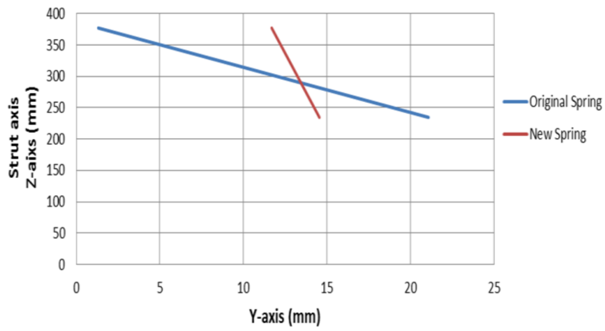


Figure 2 Comparisons of Spring Force Lines during Full Bump.

Similar results are found during the unladen and full rebound position as summarized in Table 1. At the unladen length of 217.0mm, the original spring force line deviates at 1.57^0 from the vertical axis while the new spring recorded approximately the same at 1.51^0. However, the new spring force line deflects in the negative direction compared to the usual positive direction of the original spring force. This can be due to the situation, whereby under unladen condition the spring is not much compressed, and additionally the spring rate is not large, thus the center of distribution of spring force possibly to be shifted.

The original spring force line is found to deviate at 3.32^0 from the vertical axis compared to the deviation of new spring force line at 1.39^0 during full rebound position. This indicates that the new spring force line deviation is enhanced by 1.93^0. Upon the returning motion of the spring after compressed to its original position, the side load can be reduced along the dynamic motion and prevent much side load transferred back to the original portion of the spring.

Table 1 Results of Spring Force Line Angle.

Spring Position	**Spring Length [mm]**	**Original Spring Force Line Deviation Angle [°]**	**New Spring Force Line Deviation Angle [°]**
Full Bump	142.4	7.90	1.15
Unladen	217.0	1.57	1.51
Rebound	292.0	3.32	1.39

4. CONCLUSIONS

The spring force line of original and new morphed spring has been evaluated in this paper. The new spring design is proven to reduce the deviation of spring force line by 6.75^0 and 1.93^0 from vertical axis in full bump and rebound condition. The improvement of spring force line deviations is considered large for a small morphing of 20mm spring. The new spring design can resolve the drawback of the vehicle manufacturers and users where the tire thread sustained in a good condition after a long travel. When the side load is reduced, the wear on the shock absorber can be minimized and preserve excellent force absorption rate from the tyres.

ACKNOWLEDGEMENTS

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REFERENCES

[1] S.C. Jain, P.K. Sharma and D. Vadodaria,"Mcpherson Suspension System - A Review", *International Journal For Technological Research In Engineering*, Vol. 1, No. 12, 2014.

[2] S. Suzuki, S. Kamiya, T. Imaizumi and Y. Sanada, "Approaches to minimizing side force of helical coil springs for riding comfort", *SAE Technical paper series* No. 960730, 1996.

[3] J. Liu, D. J. Zhuang, F. Yu and L. M. Lou, "Optimized Design For A Macpherson Strut Suspension With Side Load Springs." *International Journal of Automotive Technology*, Vol. 9, No. 1, pp. 29-35, 2008.

[4] J.P. Hastey, J. Baudelet, E. Gerard, C. Jones and C. Viel, "Optimization on MacPherson suspensions with a spring", *SAE Technical paper series* No. 970100, 1997.

[5] Y.I. Ryu, D.O. Kang, S.J. Heo, H.J. Yim and J.I. Jeon, "Development of Analytical Process to Reduce Side Load in Strut Type Suspension. *Journal of Mechanical Sciences and Technology*. 24: 351 – 356, 2010.

[6] A. Joshi and H. Singh Chhabra, "Mathematical Model to Find Piercing Point in McPherson Strut Suspension and Design of profile for Side Force Control Spring," *SAE Technical Paper* 2012-28-0014, 2012.

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Spot welding quality evaluation of reworked galvannealed metal sheet with high strength steel (HSS)

Mohd Harris Fadhilah[1,2,*], Najiyah Safwa[1,3], Mohd Saiful Ridzuan[1]

[1)] Faculty of Engineering Technology, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[2)] Centre for Advanced Research on Energy, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[3)] Advanced Manufacturing Centre, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia

[*]Corresponding e-mail: mohdharris@utem.edu.my

Keywords: Reworked galvannealed; HSS; spot welding

ABSTRACT – Weld quality is one of the factors that determine crash worthiness of a vehicle. The aim of this study is to investigate the welding quality of 2-layer welding sheet containing reworked galvannealed metal sheet with high strength steel (HSS). The spot welding quality is then compared with the samples containing non reworked galvannealed metal sheet with HSS. The samples were prepared according to ISO 14273:2000. Obtained results indicated that spot weld between reworked galvannealed metal sheet and HSS shows a similar result with the samples containing non reworked galvannealed metal sheet and HSS.

1. INTRODUCTION

Resistance spot welding is the most common metal joining technique can be found in a vehicle. The quality of the spot welding is crucial since it determines the safety of occupants. It is reported that the welding quality determines the crash worthiness of a vehicle [1]. The metal sheets used in a vehicle are selected based on the function and necessities. Galvannealed metal sheets are used at the outer panel while HSS is used for vehicle structure. Excellence in corrosion resistance property, galvannealed metal sheet also has excellent weld ability and formability property [2]. HSS in another hand is being widely used in the structure part due to the excellence of its properties to withstand impact during crash test. It is also widely used as it helps reducing the body weight of a vehicle.

It is quite normal in automotive industry that the panel, especially outer panel is being stoned using whet stone when the worker is trying to find the defect at the panel [3]. The stoning process will remove the thin layer of Zinc coating existing at the surface of the panel. Many researchers focus the study of dissimilar material in spot weld since the materials used in a vehicle varies. However, the investigation on spot weld of reworked panel can rarely be found. This paper is to evaluate the welding quality of 2-layer spot welding, containing reworked galvannealed metal sheet with HSS and non-reworked galvannealed metal sheet with HSS. The welding characteristic is compared with the available standard by ANSI/AWS/SAE.

2. METHODOLOGY

The samples used for this research were galvannealed steel, and HSS, with thickness for each material, 0.7mm. The galvannealed metal sheet was first reworked using abrasive paper to remove the coating layer of the galvannealed metal sheet. Once the surface finish became shiny they were then welded with HSS material. Another set of welding samples were the combination of non-reworked galvannealed metal sheet and HSS. Figure 1 shows the schematic drawing of the sample size and welding orientation used in this research.

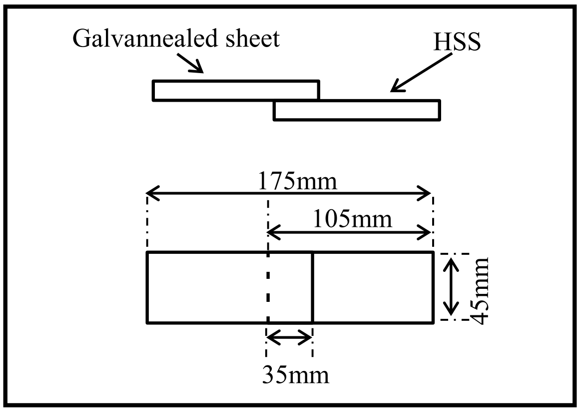

Figure 1 Schematic drawing of sample size and orientation for spot welding.

The welding parameters for both specimens were set to the same parameters. Welding parameters for both samples are shown in Table 1.

Table 1 Welding parameter.

Parameter	Value
Current	1.5kA
Pressure	3 bar
Weld time	32 cycle

The type of failure, nugget size and tensile strength for each sets of samples were then checked and compared.

3. RESULT AND DISCUSSION

Table 2 above displayed the nugget size of sample with reworked galvannealed metal sheet and HSS, and sample with galvannealed metal sheet and HSS. The average nugget sizes for both samples were 6.06 ± 0.3(mm) and 6.22 ± 0.13(mm) respectively. The obtained results adhere to the minimum nugget size recommended by ANSI/AWS/SAE as per equation 1; where *t* is the total thickness of the welded materials.

$$d=4\sqrt{t} \tag{1}$$

Table 2 Nugget size for both sets of samples.

Sample	Nugget diameter(mm) (sample with galvannealed)	Nugget diameter(mm) (sample with reworked galvannealed)
1	6.1	6.0
2	6.2	6.2
3	6.4	5.6
4	6.3	6.1
5	6.1	6.4

Recommended nugget size for these samples based on the ANSI/AWS/SAE standard is 4.73mm. The difference in the nugget size between both samples is due to the existence of the zinc coating in the galvannealed metal sheet. Zinc coating increased electrical resistivity thus resulting the higher volume in fusion zone [4]. As for the tensile test, the results are indicated in table 3.

Table 3 Maximum stress during tensile test.

Sample	Max stress MPa (sample with galvannealed)	Max stress MPa (sample with reworked galvannealed)
1	491.78	515.504
2	535.686	507.241
3	502.868	516.812
4	499.914	517.97
5	505.529	565.013

The average reading for samples with reworked galvannealed metal sheet and HSS, and sample with galvannealed metal sheet and HSS are 524.51 ±23.03MPa and 507.16 ±16.76 MPa respectively. The difference of maximum stress, might be due to the difference in the width of HAZ (Heat Affected Zone) area between the samples.As both types of samples exhibited pullout failure, the fracture during tensile test occurred at the HAZ area. Thus, a smaller nugget at reworked galvannealed metal sheets indicates the wider area of HAZ, where it became the location of the occurring fracture [5]. The size of HAZ for reworked galvannealed metal sheet and HSS, and sample with non-reworked galvannealed metal sheet and HSS are 2.404mm and 1.081mm respectively.

4. CONCLUSION

Based on the research, both sets of samples had shown pullout failure. The size of the nugget for samples containing reworked galvannealed steel is 2.5% lower from the samples containing galvannealed steel. Nonetheless, both samples adhered to the recommendation of nugget size recommended by ANSI/AWS/SAE. The tensile strength of the sets containing reworked samples conversely displayed an opposite result of the nugget size. Though the difference is less than 5 %, the difference in the result might be due to the wider area of HAZ in samples containing reworked galvannealed metal sheet. Based on these 3 criteria, the reworked of galvannealed panel can be said of having a less impact to the quality of spot weld.

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REFERENCES

[1] S. Donders, M. Brughmans, L. Hermans, C. Liefooghe, H. Van der Auweraer and W. Desmet, "The robustness of dynamic vehicle performance to spot weld failures," *Finite Elements in Analysis and Design*, Vol. 42, No. 8–9, 670-682, 2006.

[2] A. Chakraborty, A. Mondal, S. Agnihotri, R. Pais, and M. Dutta, "Investigation of a surface defect and its elimination in automotive grade galvannealeded steels," *Engineering Failure Analysis*, Vol. 66, 455-467, 2016.

[3] S. Hazra, R. Roy, D. Williams, R. Aylmore and D. Hollingdale, "A novel inspection method for determining the cosmetic quality of automotive skin panels," *Journal of Materials Processing Technology*, Vol. 213, 2049-2063, 2013.

[4] M. Pouranvari, S.P.H. Marashi and S.M. Mousavizadeh, "Influence of fusion zone size and failure mode on mechanical performance of dissimilar resistance spot welds of AISI low carbon steel and DP600 AHSS," *Materials & Design*, vol. 32, No. 3, 1390-1398, 2011.

[5] V.H. Baltazar Hernandez, M.L. Kuntz, M.I. Khan and Y. Zhou, "Influence of microstructure and weld size on the mechanical behaviour of dissimilar AHSS resistance spot welds," *Science and Technology of Welding and Joining*, Vol. 13, 769-776, 2008.

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Investigation on hBN Nanoparticle dispersion stability in liquid phase with difference dispersion agents

A.A.E.M. Nawi[1], M.I.H.C. Abdullah[1,3,*], M.F.B. Abdollah[2,3], N.R.M. Nuri[1,3], M.A. Samsudin[1,3]

[1)] Faculty of Technology Engineering, Universiti Teknikal Malaysia Melaka,
Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[2)] Faculty of Mechanical Engineering, Universiti Teknikal Malaysia Melaka,
Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[3)] Centre for Advanced Research on Energy, Universiti Teknikal Malaysia Melaka,
Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia

[*]Corresponding e-mail: ilmanhakimi@utem.edu.my

Keywords: Stability; surfactant; nanoparticles; uv-spectrometer

ABSTRACT – The aims of this study is to investigate the effect of difference surfactant agent for better dispersion of hBN solid nanoparticles inside the liquid phase. A set of sample was prepared by using ultrasonic homogenizer. Three types of surfactant agent were used which are Sodium Cholate, Oleic acid and Sodium Benzene Dodecyl Sulphonate (SDBS). The absorbance values of the nano-oil were measured using UV-spectrometer. The result shows that suspension of conventional engine oil SAE 15W-40 with the addition of hBN nanoparticles and surfactant agent of SDBS manage stable over the period of 56 days.

1. INTRODUCTION

Today, a lubricant is the most important things for any machinery parts that appearing in this world. The main features of lubricant are to controls resistance and friction between surfaces through supplying a durable film [1]. Even though lubricant can help reduce the friction and wear, lubricant still have their disadvantageous which is it will degrade and deteriorate very rapidly in some operating conditions. According to recent numerous studies [2-4], the nanotechnology can indeed improve the lubrication properties oils. By mixing lubricant with nanoparticles (additives) it can maximizing the performance of the lubricant. Nanoparticles did not dissolve with lubricant. As reported by Yu and Xie [2] the agglomeration of nanoparticles resulting not only on the settlement and clogging of micro-channels but also the decreasing of thermal conductivity of nano-lubricant.

Addition of dispersants agent inside the oils with mixed of nanoparticles is an easy and economic method to enhance the stability of nano-oil [3]. According to Paramashivaiah and Rajashekhar [4] the use of surfactant agent is to provide an effective and efficient coating to induce electrostatic or steric repulsions that can counterbalance van der Waals attractions and also give better dispersion on the nanoparticles additive inside the oil [5]. The uncontrolled factor for the dispersion stability of nanoparticle in the lubricant is the sedimentation. Sedimentation means settling of particle or floccules occur under gravitational force in liquid dosage form. To decrease the sedimentation, the dispersion or suspension agent must be added into the lubricant. Furthermore, this both agents also can help to increase the stabilization of nano-lubricant and enhance the thermal conductivity. Therefore, it is importance to investigate the effect of difference surfactants agent of the nano-oil sample on the stability of nanoparticles. The stability of nano-oil will be evaluated using quantitative and qualitative analyses.

The objective of this studies is to investigate the effect of different surfactant agent and suspension agent for better dispersion of solid nanoparticle inside the liquid phase condition.

2. METHODOLOGY

A set of sample was prepared according to the Table 1 with an optimal composition of 0.5 vol.% hBN nanoparticles in SAE 15W-40 conventional engine oil and with addition 0.3 vol.% surfactant agents as a nano-oil. The samples were mixed using ultrasonic homogenizer (Sartorius Labsonic P) with 50% amplitude and 0.5 active time interval as shown in Figure 1. The absorbance value was recorded for approximately two months by using UV-Spectrometer and the image of samples sedimentation was captured.

Table 1 Sample preparation.

Test No.	Factor		
	Surfactant	**Nanoparticle**	**Homogenize time (minutes)**
1	Oleic acid		
2	Sodium cholate	hBN	30
3	SDBS		

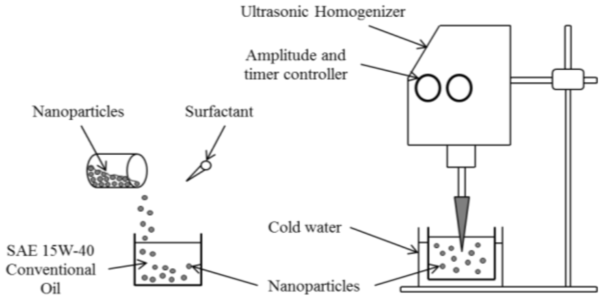

Figure 1 Schematic diagram sample preparation by using ultrasonic method.

3. RESULT AND DISCUSSION

Table 2 shows picture of nano-oil dispersed with SDBS, Sodium Cholate and Oleic acid for approximately two months. Based on the image observation, there are no changes occur from the 1st day until the 4th day for SDBS agent compared to Sodium Cholate and Oleic acid. The changes can be seen after a week which the colour of the oil sample become pale yellow compare to the image in the previous day. According to Rosicka and Sembera, [3] due to attractive magnetic forces, the rate of aggregation is significantly higher, whereas the repulsive electrostatic forces are almost negligible which it can resulting in occurring of sedimentation process. The stability trend for overall 56 days of surfactant agent shows in Figure 2, where the higher absorbance obtained by SDBS agent.

Table 2 Nano-oil with SDBS, Sodium Cholate and Oleic acid.

Picture of SAE 15W 40 + Additive + Surfactant						
Days	**abs**	**SDBS**	**abs**	**Sodium Cholate**	**abs**	**Oleic acid**
1	3.984		3.385		3.999	
2	3.981		3.325		3.989	
3	3.784		3.320		3.892	
4	3.705		3.289		3.789	
7	3.689		3.282		3.690	
8 (Week 1)	3.681		3.099		3.644	
15 (Week 2)	3.536		3.011		3.587	
22 (Week 3)	3.515		2.986		3.237	
28 (1 Month)	3.461		2.959		3.206	
56 (2 Month)	3.1953		2.761		3.110	

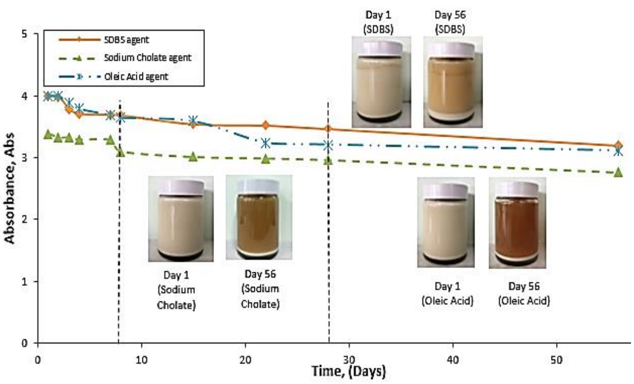

Figure 2 Comparison of SDBS, Sodium Cholate and oleic acid stability for 56 days.

SDBS commonly used inside washing detergent, which the main purposed is to disperse the liquid detergent from liquid separation. In primary conclusion, the electrostatic mechanism between the SDBS and hBN nano particles maybe increase the repulsive force and result in thicken the electrical double-layer that provides more dispersion stability. By referring to main function of SDBS agent, there are possibility that SDBS can improve the stability of the nano-oil which resulting to highest absorbance value until 56 days. Further investigation on the intermolecular reaction between the SDBS and hBN maybe key factor for the success of the nano-oil stability which not going to be discuss here.

4. CONCLUSION

As conclusion, the result shows that the effect of SDBS agent towards hBN nanoparticles in nano-oil is more stable compare to Sodium Cholate and Oleic acid. The electrostatic mechanism which increase the repulsive force and result in thicken the electrical double-layer that provides dispersion stability maybe the key solution for future work.

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REFERENCES

[1] M.I.H.C. Abdullah, M.F.B. Abdollah, H. Amiruddin, N. Tamaldin, N.R. Mat Nuri, "The Potential of hBN nanoparticles as friction modifier and antiwear additive in engine oil," *Mechanics and Industry,* vol.17, pp. 104, 2015.

[2] W. Yu, H. Xie, "A Review on Nanofluids: Preparation, Stability Mechanisms, and Applications," *Journal of Nanomaterials*, pp. 1-17, 2012.

[3] D. Rosická, J. Šembera, "Changes in the nanoparticle aggregation rate due to the additional effect of electrostatic and magnetic forces on mass transport coefficients," *Nanoscale Res Lett Nanoscale Research Letters,* 8(1), 20, 2013.

[4] B.M. Paramashivaiah, C.R. Rajashekhar, "Studies on effect of various surfactants on stable dispersion of graphene nano particles in simarouba biodiesel," *Materials Science and Engineering*, 149, 2016.

[5] H. Amiruddin, M.F.B. Abdollah, A.M. Idris, M.I.H.C. Abdullah, N. Tamaldin, "Stability of nano-oil by pH control in stationary conditions," in *Proceedings of Mechanical Engineering Research Day 2015*, pp. 55-56, 2015.

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Prediction of engine performance for different cam profile by using GT-power simulation

A.B. Shahriman[1,2], A.K. Mohamad Syafiq[1,2,*], M.S.M. Hashim[1,2], Zuradzman M. Razlan[1,2], W. Khairunizam[1,2], D. Hazry[2], M. Afendi[2], R. Daud[2], M. Tasyrif Abdul Rahman[2]

[1)] School of Mechatronic Engineering, Universiti Malaysia Perlis, Kampus Ulu Pauh, 02600 Arau, Perlis, Malaysia
[2)] Motorsport Technology Research Unit, Universiti Malaysia Perlis, Kampus Ulu Pauh, 02600 Arau, Perlis, Malaysia

[*]Corresponding e-mail: syafiq.akbar89@gmail.com

Keywords: Automotive; cam profiling; engine performance

ABSTRACT Camshaft is one of the main parts in the automotive engines inside internal combustion engine. In order to enhance the performance of MODENAS motorcycle engine, a new design of cam profile is introduced. The design stage for numerous cam lift values is applied using GT-SUITE. GT-SUITE software was also used in order to plot engine performance curve, which gives torque graph. These values will be compared with the actual values. The best design with optimum parameters will be chose for industry implementation. The cam with 20 mm for cam base circle diameter and width plotted the best performance curve and was recommended to be manufactured with considering another external factors.

1. INTRODUCTION

There are a lot of concerns on engine performance improvement. Many researches had been carried out in order to occupy this objective, by trying to get the improvement of the number of work done with the same amount of fuel used. The parameters that always contribute to the improvement of engine performance are compression ratio, power, torque, efficiency and the amount of work done for an engine [1]. One of the methods to improve the engine performance is by optimizing the cam events. Wizard [2] listed all the elements needed to achieve high output by optimizing cam events. The first element is by optimize the quality and ratio of mixture besides the turbulence and mixture swirl. Afterwards is trying to expand expansion/compression ratio and diminish heat loss. Lastly try to give the maximum rpm. All these elements need to be achieved for the optimum engine performance [2].

Theoretically, by increasing the compression ratio, the output of the engine can be improved but there is a need to consider other parameter such as amount of fuel sprayed. This is because if the compression ratio is too high, and not enough fuel was supplied, it can cause the cylinder to be overheated and can burn out the cylinder itself. Therefore, an optimum point needs to be identified, which can help to get the best performance curve and avoid overheated phenomenon [1].

One of the main components that will be investigated to improve the engine performance is camshaft. Camshaft is one of the most important control parts of automobile engines and other auto machines. In the engine, the cam pushes the valve in order to open and the camshaft rotates the spring, which function to return the valve to closed position. Meanwhile, valve timing can be described as the usage system to calculate the exact time for valve to open and close, for both intake and exhaust valve. Every single design of cam profile and the angle between these two cam, intake and exhaust directly impact the engine performance.

Thus, the design of cam profiling and the timing are critical parameters since both affect the compression and combustion of the engine. The best quality of compression and combustion will result in a better performance in term of torque and the speed [3-5]. In order to get the best cam profile, a simulation will be conducted to simulate the expected result. The simulation result will be compared with the real application. The objective of simulation is trying to predict the engine behaviour corresponding to various cam designs. Every single design will give different results. In order to reduce the time consumption and manpower, the simulation is conducted.

2. METHODOLOGY

2.1 Hardware and software requirement

MODENAS genuine sample camshaft was selected and referred as a benchmark and guideline, which was obtained from MODENAS and had been approved by MODENAS. By applying vernier caliper and ruler, the detail dimensions were obtained in order for a new design development. After the dimension and required parameters were obtained, GT- SUITE valve train was the tool for designing and simulation stage. Due to MODENAS restriction, the assumption had been used for all the parameters values. Figure 1 shows the project map for the camshaft simulation. Project map is the schematic diagram which is the combination of series of icons and connectors. The combination of icons and connectors in the 1-D mapping was implemented in describing the real engine condition and transferring the real engine condition into 1-D simulation mapping. All the icons, objects or components must be defined first according to the real or existing engine arrangement before all were linked together as a system. After that, the simulation parameters were selected and designated as accurate as the real engine condition.

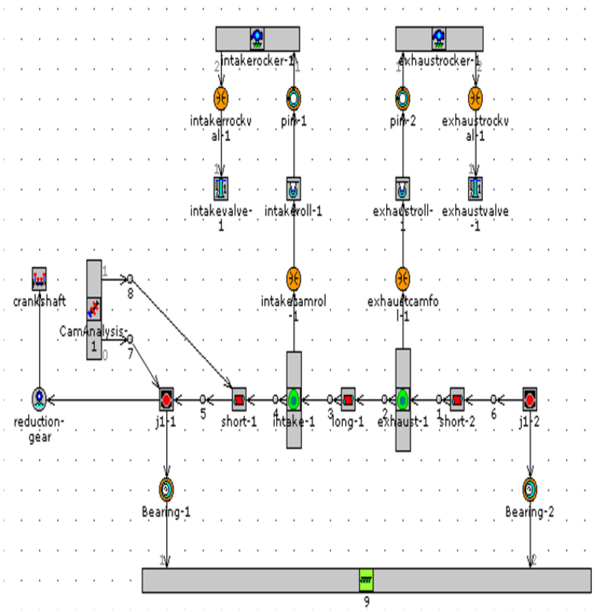

Figure 1 Project map

2.2 *Design stage*

After all the compulsory parameters had been inserted, the next step was design stage. By using GT-SUITE software, single overhead camshaft was designed first based on the parameters listed. The parameters were engine crankshaft, intake and exhaust valve size, cam lobe design, cam base circle diameter and the width. The last two parameters were manipulated parameters, which being varies since to determine the best cam with the optimum values of width and cam base circle diameter. Meanwhile, the other parameters were constant for each variation. The design stage started with identifying the cam or lift profile, where several alternatives are available either from ASCII formatted text file or by copying and pasting any spreadsheet software packages. Table 1 shows different parameters for intake and exhaust width cam and cam base circle diameter for three different designs.

Table 1 Design parameters.

Components	Parameters	Intake	Exhaust
First Design	Width (mm)	20	20
	Cam base circle diameter (mm)	20	20
Second Design	Width (mm)	22	22
	Cam base circle diameter (mm)	22	22
Third Design	Width (mm)	20	18
	Cam base circle diameter (mm)	20	18

2.3 *Simulation*

Once the model with different parameters had been identified and listed, the next step was simulation stage. All the required information need to be described properly. Among the required information were type of simulation, case specific input and desired output. This information was described properly by making the selection under Run setup, which a lot of folders are available and also consist of the required values to run the model.

3. RESULTS AND DISCUSSION

3.1 *Valve Lift*

Figure 2 showed the result of valve lift for three different designs. For the first design, the values for both width intake, exhaust and cam base circle diameter intake and exhaust are similar, 20 cm. Based on the graph, this design is plotted as the lowest number of valve lift, 6mm. For the second design, all the values were increased by 2 mm, from 20 mm to 22 mm meanwhile the third one is a bit different compared to the first two designs. The value for intake was set to 20 mm and 18 mm for the exhaust. The values of valve lift for both design 2 and design 3 indicated the same value, 10 mm, which was greater than the first design. Wenjie stated that the lift area should be optimum value to boost the capacity of gas exchange [6].

Thus, it can be concluded that design 2 and design 3 open further compared that to design 1. Unfortunately, by increasing the value of valve lift, the duration of time also increased. The relationship between the lift values and the time was linear proportional. This was because of the ramp was limited in term of shape. Therefore, to minimize this effect, the type of lifters need to be considered either flat or roller. Besides, as can be seen, the ramps for all three designs did not steep to each other. Sandor mentioned that these values could not been steep to each other because to minimize the opening and closing speed for the valve, in order to prevent damage either the valve or the cylinder head.

3.2 *Performance curve*

Figure 3 was the graph of performance curve, torque for all three different designs. The graph represented the torque value versus crank speed, rpm. From the simulation result, all these three lines were plotted and compared to each other in order to obtain the best design in term of torque aspects. All these three design gave three different trends and different values of maximum torque.

Firstly, the shape of graphs was not so smooth and gave different trends. Despite non uniform shape, all three designs achieved the maximum torque value at the same speed, 8000 revolutions per minute, rpm.

The first design gave the highest torque value at 0 rpm with 1.2 N-m. It keeps increasing until 3800 rpm before experiencing sudden drop with torque value was 2.8 N-m. At 6000 rpm, the graph inclines till 8000 rpm with a maximum torque value was 7.4 N-m before the graph decline again. This trend was because of the lift produced allowed more air fuel mixture entering the cylinder at low speed of engine.

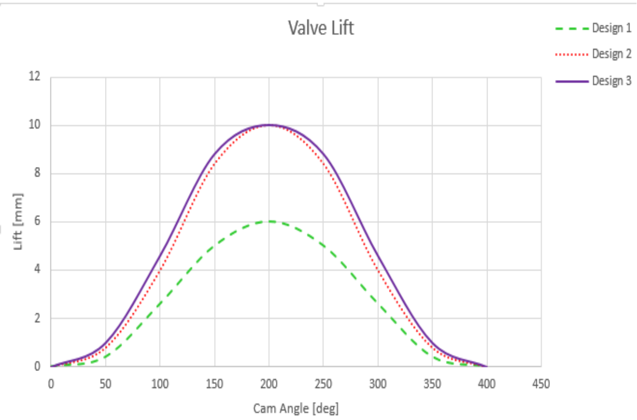

Figure 2 Valve lifts for three different designs.

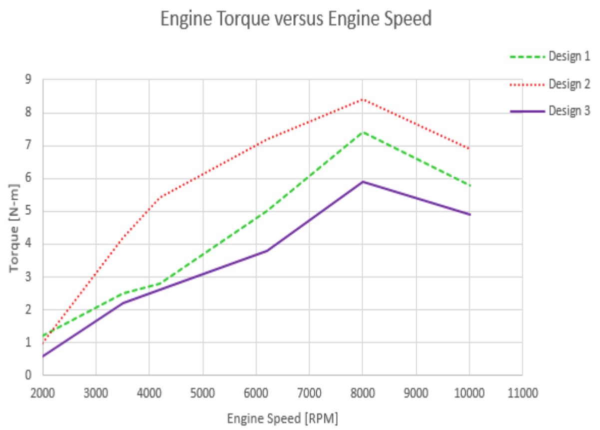

Figure 3 Graph torque versus crank speed.

The trend of all three designs illustrated the same pattern, which the graph increased until the peak point at 8000 rpm before decline to a lower value. The peak value for the second design was 8.2 N-m and the maximum value for third design was 5.9 N-m, which was the lowest values compared to another two designs but all these three designs demonstrated the same characteristics, which all of them build low-end torque.

The third design gave lowest torque value because of the generated lift profile. Even the maximum lift profile for design 2 and design 3 gave the same value, 10 mm but the lift profile itself slimmer compared than the second design. One of the factor was the size of circle base radius for exhaust cam smaller than the second design. Therefore, the timing for third design shorter in allowing air fuel mixture entering the cylinder for combustion process occurred in producing torque value. A part from that, the timing of valve opening for third design was not efficient as the second design, which the gas passage that entering the cylinder and the exhaust gasses transmitted outside the cylinder was not at the best condition. The amount of exhaust gasses that trapped inside the cylinder were greater and caused the amount of air fuel mixture that entering the cylinder was drop.

Table 2 Result of Maximum Valve Lift and Torque.

Components	**Max Valve Lift (mm)**	**Max Torque (N-m)**
First Design	6	7.4
Second Design	10	8.4
Third Design	10	5.9

4. SUMMARY

In this study, it can be summarized that the best design among three different designs that have been proposed was design 2, which performed the best result either in term of valve lift values or torque versus crank speed. The second design with the parameter values of 20 mm for both cam base circle diameter and width of cam, for both intake and exhaust. Both second and third cam design gave the highest valve lift of 10 mm. Unfortunately, the third design performed poorly in terms of torque, while the second design gave the greatest value for torque values with 8.2 N-m at 8000 rpm. Therefore, the second design parameters were strongly recommended for the manufacturing level but bear in mind that there were other external aspects that can also affect the engine overall performance.

REFERENCES

[1] P. Tamil Porai, N.V. Mahalakshmi, V. Ganesan, "Effect of compression ratio on performance of a four-stroke spark ignition engine - A theoretical and experimental study," *Advanced Materials Research,* vol. 984-985, pp. 945-949, 2014.

[2] D. Wizard, "What it takes to make power," 14-16, 2012.

[3] M.T. Muslim, H. Selamat, A.J. Alimin, N.M. Rohi, M.F. Hushim, "A review on retrofit fuel injection technology for small carburetted motorcycle engine towards lower fuel consumption and cleaner exhaust emission," *Renewable and Sustainable Energy Review*, vol. 35, no. 35, 279-284, 2014.

[4] Z. Zhang, R.B. Jia, Q.H. Yu, M.L. Cai, "Study on design criteria and methods for the valve train of the compressed-air engine," *Applied Mechanics and Materials, vol.* 278, 159-164, 2013.

[5] M.F. Hushim, A.J. Alimin, M.F. Mansor, "Effect of intake manifold angle of port-fuel injection retrofit-kit to the performances of an SI engine," *Applied Mechanics and Materials*, vol. 165, pp. 31-37, 2012.

[6] W. Qin, Y. Chen, "Study on optimal kinematic synthesis of cam profiles for engine valve trains," *Applied Mathematical Modelling*, vol. 38, no. 17, pp. 4345-4353, 2014.

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Optimizing the emission effects of continuous hydrogen injection on diesel engine intake using Taguchi method

M.N.M. Norani[1,*], B.T. Tee[1,2], Z.M. Zulfattah[1,2], M.N. Mansor[3]

[1)] Faculty of Mechanical Engineering, Universiti Teknikal Malaysia Melaka,
Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[2)] Centre for Advanced Research on Energy, Universiti Teknikal Malaysia Melaka,
Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[3)] Faculty of Mechanical and Manufacturing Engineering, Universiti Tun Hussein Onn Malaysia
86400 Parit Raja, Batu Pahat, Johor, Malaysia

[*]Corresponding e-mail: mohamadnordin91@gmail.com

Keywords: Diesel hydrogen; emission; diesel engine

ABSTRACT – Diesel engine nowadays is the most efficient prime mover in automotive industry. It is also a significant power generator utilizing mechanical and chemical reaction, powering electricity generation and locomotive which is economically better than gasoline engine. However, it produces a larger environmental effect due to higher yield of emission. Thus, the research is about analyzing the optimized parameters that affect emission level by supplying hydrogen gas inside the intake of a single cylinder diesel engine. The criteria such as nitrogen oxide, carbon monoxide, hydrocarbon and carbon dioxide have analyzed using Taguchi's method. Thus, the optimized parameters have been compared with diesel fuel baseline. It is discovered that the optimized condition reduces emission by 44.85% of CO, 2.19% of NO_x, 79.53% of HC and 67.56% of CO_2.

1. INTRODUCTION

Diesel engine is utilized in almost everything that involves the need for an energy supply such as power generation, automotive truck and locomotive. However, diesel engine comes with a drawback which is exhaust gas or emissions pollutant. Exhaust gases contain hydrocarbons, nitrogen oxides and carbon monoxide, which is harmful to the environment and mankind [1]. Other than that, the exhaust emission also releases the carbon dioxide gas that can cause the greenhouse effect.

Taguchi method design is used significantly to simplify the number of task experiments greatly and able to reach objective by using orthogonal arrays. Combination effect of fuel injection timing, exhaust gas circulation (EGR) ratio, and fuel injection pressure to control NO_X emission using Taguchi's L_9 orthogonal arrays were examined by Saravanan et al [2]. Adding a little amount of hydrogen gas into diesel can enhance fuel efficiency during the process of combustion in diesel engines. Horng et al. [3] studied optimal condition for diesel/biodiesel blend using hydrogen gas and EGR at inlet port using Taguchi's method. He finds best for brake thermal efficiency (BTE), brake specific fuel consumption (BSFC), nitrogen oxide and smoke at combination B20 biodiesel blend, 30% of hydrogen and 40% EGR ratio. It gives reduction 25.4% for BSFC, 74.1% for NOX and 29.6% for smoke at 60% load. Furthermore, the hydrogen injection inside the diesel engine at low load condition saves fuel of up to 16.82% [4]. Thus, the research is about analyzing the optimized parameters that affect emission level by supplying hydrogen gas inside the air intake of diesel engine.

2. METHODOLOGY

The single cylinder, 4 stroke engine was used for this investigation. Figure 1 shows the general schematic diagram for this experiment.

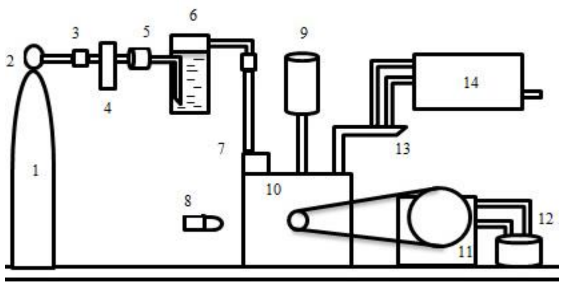

Figure. 1. Schematic layout.

Description: 1-Hydrogen tank, 2-Pressure regulator, 3-Flashback arrestor, 4-Flow meter, 5-One ways flow valve, 6-Flame trap, 7- Air intake, 8-Tachometer, 9-Diesel tank, 10-KM 170-F engine, 11-Hydraulic pump, 12-Hydraulic tank, 13-Exhaust manifold, 14- Gas analyzer.

Table 2 and Table 3 show factors with level in Taguchi's method array L9, those parameters are speed of engine (A), hydraulic loading (B) and hydrogen rate (C). Response data from Table 3 is used to optimize the parameter respect to nitrogen oxide, carbon monoxide, hydrocarbon and carbon dioxide emission production (low is better). This is achieved via Minitab software in which after obtaining the highest dominating factor effecting the emission. In order to verify the prediction from Taguchi method the optimized parameters are tested again. Following the responses, optimized parameters were compared with diesel fuel baseline at same condition without undergoing hydrogen gas addition.

Table 2 Factors with level.

	Factors	Level 1	Level 2	Level 3
A	Speed engine (RPM)	Low (2100±100)	Medium (2800±100)	High (3700±100)
B	Hydraulic loading (kPa)	1000	2000	3000
C	Hydrogen rate (l/min)	0	2.5	5

Table 3. Taguchi's method L_9 (3^3) orthogonal arrays.

No.	A	B	C
1	1	1	1
2	1	2	2
3	1	3	3
4	2	1	2
5	2	2	3
6	2	3	1
7	3	1	3
8	3	2	1
9	3	3	2

3. RESULTS AND DISCUSSION

This paper identifies the optimal operating factor for low nitrogen oxide, carbon monoxide, hydrocarbon and carbon dioxide. The average responses and the signal noise ratios (S/N) have been calculated in Taguchi's process. An analysis of variance (ANOVA) identifies the most effective factors and quantifies their effects. Table 4 shows the data on how to determine optimal conditions throughout factor contribution.

Figure 2 displays CO level between diesel fuel baseline and optimized parameter at the same condition. At optimum condition which is at low engine speed, loading of 1000 kPa and 5 l/min shows that 248.711ppm of carbon monoxide while at baseline produce 451ppm of CO. NOx emission level between baseline diesel and optimized parameter is illustrated in Figure 3. Optimum condition shows reduction 2.19% of NOx which is better than baseline diesel fuel. In Figure 4, hydrocarbon emission is withdrawn with optimized condition about 79.53% against original diesel baseline. The carbon dioxide is clearly decreases with optimized condition up to 67.56% beside baseline as shown in Figure 5.

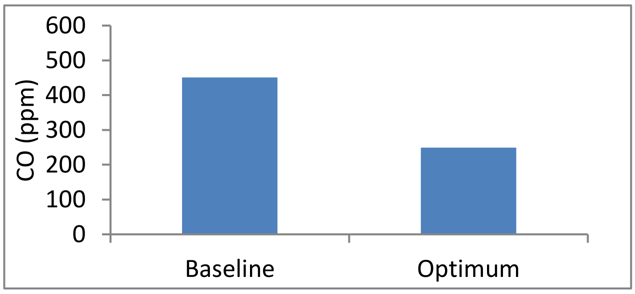

Figure 2. Comparison of CO level between baseline diesel engine and optimized engine.

Table 4. Contribution of factors S/N ratio for emission.

Factors	levels	S/N ratio for CO	S/N ratio for NOX	S/N ratio for HC	S/N ratio for CO2	Overall optimum
A	1	-47.62		-10.87	-4.59	
	2	-52.22		-17.32	-7.32	A_1
	3	-56.82		-22.18	-11.46	
B	1		-39.27			
	2		-43.48			B_1
	3		-43.96			
C	1	-55.16		-20.04	-10.384	
	2	-50.77		-16.20	-6.814	C_3
	3	-50.72		-14.14	-6.170	

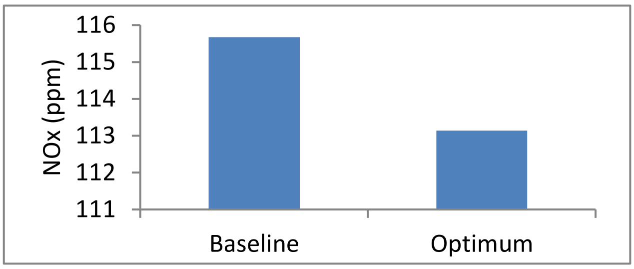

Figure 3 Comparison of NOx level between diesel engine and optimized engine.

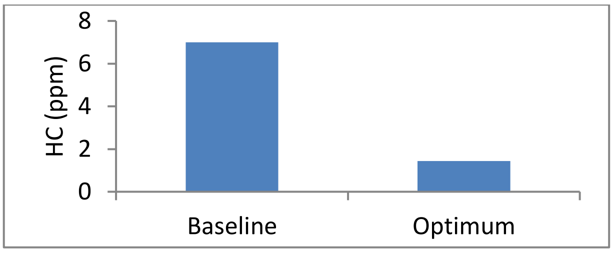

Figure 4 Comparison of HC level between baseline diesel engine and optimized engine.

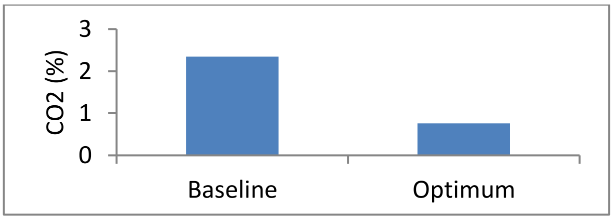

Figure 5 Comparison of CO_2 level between baseline diesel engine and optimized engine.

4. CONCLUSION

Through Taguchi method, the best CO, NO_X, HC and CO_2 emission is determined at the combination of low engine speed (A_1), 1000 kPa hydraulic loading (B_1) with 5 l/min of hydrogen rate (C_3). The predictions from Taguchi shows good agreement with confirmation interval between 86% to 90%. Optimized conditions reduce about 44.85% of CO level, 2.19% of NO_x, 79.53% of HC and 67.56% of CO_2 compare with the baseline diesel condition.

REFERENCES

[1] J. Young, “Hydrogen injection in diesel engine for fuel efficiency improvement,” 2008.

[2] S. Saravanan, G. Nagarajan, S. Sampath, Multi response optimization of NO_X emission of stationry diesel engine,” *Fuel*, vol. 89, no. 11, pp. 3235-3240, 2010.

[3] H.W. Wu, Z.Y. Wu, “Using Taguchi method on combustion performance of diesel engine with diesel/biodiesel blend and port inducting,” *Applied Energy*, vol. 104, pp. 362-370, 2013.

[4] M.N.M. Norani, B.T. Tee, M.Z. Zakaria, M.N.A. Saadun, A. Hussain, M.N. Mansor, “Effect of hydrogen injection on diesel engine performance intake: preliminary result,” in *Proceedings of Mechanical Engineering Research Day 2016*, pp. 21-22, 2016.

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Anti-roll suspension system using parallel type PID

Fathiah Mohamed Jamil[1], Mohd Azman Abdullah[1,2,*], Mohd Rasyidi Ridzuan[1,3], Munaliza Ibrahim[1], Fauzi Ahmad[1,2]

[1)] Faculty of Mechanical Engineering, Universiti Teknikal Malaysia Melaka,
Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[2)] Centre for Advanced Research on Energy, Universiti Teknikal Malaysia Melaka,
Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[3)] Department of Automotive Engineering Technology, Kolej Kemahiran Tinggi MARA Masjid Tanah, Jalan Ramuan China Besar, 78300 Lubok China, Melaka, Malaysia

[*]Corresponding e-mail: mohdazman@utem.edu.my

Keywords: Anti-roll suspension; parallel type PID; 4DOF vehicle model

ABSTRACT – This paper focusing on the investigation of parallel type Proportional-Integral-Derivative (PTPID) controller for active anti-roll suspension system in order to provide ride comfort and stability while handling the vehicle on uneven road surface. The previously verified 4 degrees of freedom (4DOF) vehicle roll model is used with the application of PTPID. The performance comparison is made between passive suspension and active anti roll suspension through the value of root mean square (RMS). It shows that PTPID control system produces better output responses thus providing vehicle stability, good controllability and ride comfort.

1. INTRODUCTION

The need for passenger comfort, road handling abilities of tires, vehicle handling characteristics, have been the major challenges in the design of suspension system over the years. However, the ride comfort is the main concern in order to give cosiness feeling of the passengers in the running conditions [1]. However, vertical displacement and vehicle roll motion arise from various sources of vibrations of the vehicle body [2] and the road surface irregularities affect both on vehicle comfort and driving safety [3-4].

Active suspension is one of the promising technologies to overcome the body vertical vibration and roll motion. In active suspension system, the body vibration which is developed due to uneven road surface can be suppressed with the help of actuator. Furthermore, the amount of suspension deflection is affected by the vehicle structural features which help in reducing car body acceleration [5]. Higher level of suspension is reachable by designing suspension which can control and focus on minimizing car body acceleration and the suspension deflection is small and approached its limit. Moreover, an active suspension not only can offer good vibration isolation for the sprung mass to improve pitch, roll and have response of the vehicle body but also can achieve better handling stability in critical manoeuvring, accelerating and braking [6-7].

In this research, a PTPID controller is designed and tested on a vehicle ride model. A simple sinusoidal inputs are applied on both left and right tires at different frequencies. The PTPID is tuned and the optimised parameters for the controller are used. The feedbacks are applied for all states of data (vertical displacement, velocity and acceleration, roll angle, rate of roll and roll acceleration). RMS is used to measure the output performance of the PTPID comparing to passive suspension.

2. METHODOLOGY

The verified model of 4DOF half car roll plane model (Figure 1) is used in this paper. PTPID controller (Figure 2) was designed and simulated by using Matlab/Simulink software on the 4DOF model. The control strategies were evaluated on its capability in reducing body displacement and angular velocity in order to increase the ride comfort [7] and vehicle stability. Performance evaluation of active anti-roll suspension with controller is compared with passive suspension through RMS value.

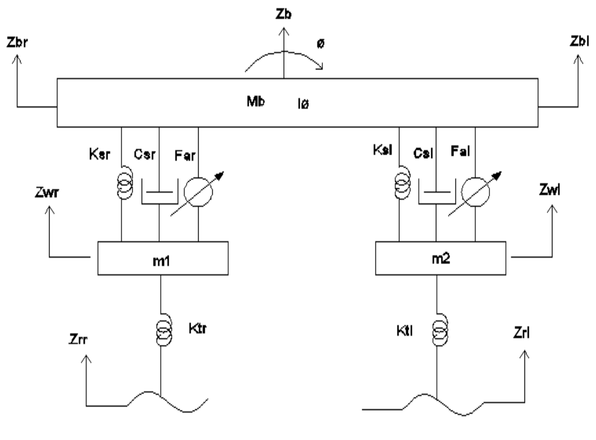

Figure 1 4DOF vehicle model with active suspension.

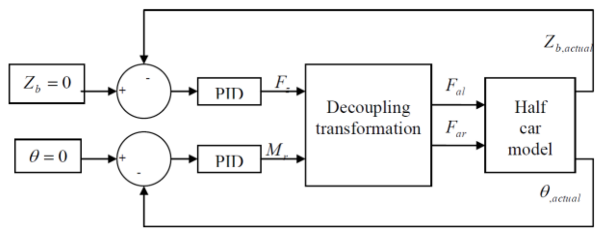

Figure 2 PTPID controller design.

3. RESULTS AND DISCUSSION

The comparison is made between passive suspension and PTPID with step of 0.1 m road input on the left tyre. Figure 3 shows that the passive suspension experienced larger value of vertical displacement which is 0.06 m compared to PTPID which returned to 0 m displacement. Figure 4 illustrates the performance of angular roll body motion. The amplitude of angular roll displacement for PTPID is stable to zero rad after 2 seconds compared to passive suspension with the value of 7 radian. Figure 5 shows the counter actuator force on the left suspension to recover the step disturbance. The performance analysis is measured in term of RMS (Table 1). Thus, it's clearly shows that an active suspension with PTPID provide better vehicle control and improve its stability and controllability on the body vehicle while it moved on uneven road surface.

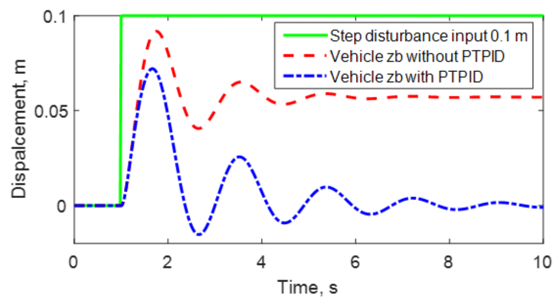

Figure 3 Vertical displacement.

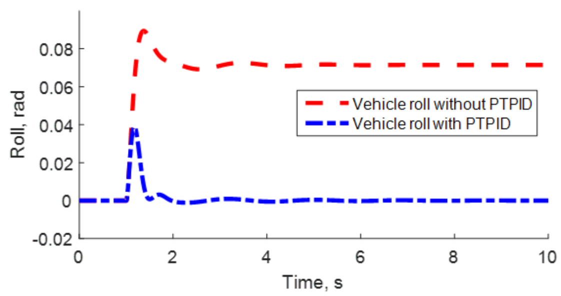

Figure 4 Roll motion.

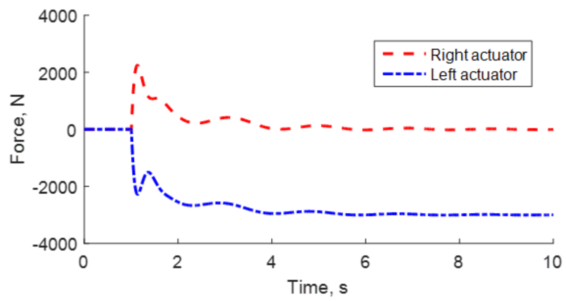

Figure 5 Actuator force.

Table 1 RMS value for passive suspension and PTPID.

Criteria	RMS Value	
	Passive	**PTPID**
Vertical displacement, m	0.0571	0.000953
Angular Displacement, rad	0.07142	1.478×10^{-5}

4. CONCLUSION

In this research, PTPID controller was proposed for active anti roll suspension system in order to reduce the effect of the vertical and angular movement on vehicle body. The improvement on vehicle body stability can be seen through the results obtained with the implementation of PTPID. The suspension system worked effectively to reduce vertical and angular body movement compared to passive suspension system. The PTPID control structure is also able to increase ride comfort and provide better vehicle handling quality.

ACKNOWLEDGEMENT

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REFERENCES

[1] M.A. Abdullah, M.R. Mansur, N. Tamaldin, and K. Thanaraj, "Development of formula varsity race car chassis", *IOP Conference Series: Materials Science and Engineering*, vol. 50, no. 1, pp. 012001, 2013.

[2] M.A. Abdullah, N. Tamaldin, M.A. Mohamad, R. S. Rosdi, and M.N.I. Ramlan, "Energy harvesting and regeneration from the vibration of suspension system", *Applied Mechanics and Materials*, vol. 699, pp. 800-805, 2015.

[3] M. Coric, J. Deur, J. Kasac, H.E. Tseng, D. Hrovat, "Optimisation of active suspension control inputs for improved vehicle handling performance," *Vehicle System Dynamics, International Journal of Vehicle Mechanics and Mobility*, vol. 5, pp. 1574-1600, 2016.

[4] M.A. Abdullah, M.A. Salim, M.Z. Mohammad Nasir, M.N. Sudin, and F.R. Ramli, "Dynamics performances of Malaysian passenger vehicle," *ARPN Journal of Engineering and Applied Sciences*, vol. 10, no. 17, pp. 7759-7763, 2015.

[5] M.A. Abdullah, J.F. Jamil and M.A. Salim, "Dynamic performances analysis of a real vehicle driving", *IOP Conference Series: Materials Science and Engineering*, IOP Publishing, vol. 100, No. 1, p. 012017, 2015.

[6] M.A. Abdullah, J.F. Jamil and A.E. Mohan, *Vehicle dynamics modeling & simulation*, Malacca: Centre for Advanced Research on Energy (CARe), Faculty of Mechanical Engineering, Universiti Teknikal Malaysia Melaka; ISBN: 978-967-0257-78-5; 2016.

[7] M.A. Abdullah, M. Ibrahim, M.A.H. Abdul Rahim, "Experimental and analysis of vehicle dynamics performance based on driving behavior", *Journal of Mechanical Engineering*, vol. SI1, no. 1, pp. 193-206, 2017.

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Bodywork design and analysis of FV Malaysia race car

Mohd Azman Abdullah[1,2,*], Naqiuddin Nasirin[1], Muhammad Zahir Hassan[3], Syahibudil Ikhwan Abdul Kudus[1,2]

[1)] Faculty of Mechanical Engineering, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[2)] Centre for Advanced Research on Energy, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[3)] Faculty of Engineering Technology, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100, Durian Tunggal, Melaka, Malaysia

[*]Corresponding e-mail: mohdazman@utem.edu.my

Keywords: FV Malaysia; bodywork; coefficient of drag

ABSTRACT – This paper presents the aerodynamics properties of the Formula Varsity (FV) Malaysia race car through computational fluid dynamic (CFD) simulation using ANSYS software. The aerodynamics properties include coefficient of drag, coefficient of lift, and their values at different speeds of travel. Results from the analysis can be used to observe the performance of the race car during racing event. The findings may be useful in optimizing the aerodynamic properties of the bodywork for future design and fabrication.

1. INTRODUCTION

Aerodynamic is the main factor when designing open-wheel race car. Shape of the vehicle is often the main factor that contributes to the aerodynamic performance [1]. In order to get the stability on open-wheel race car, aerodynamic drag and aerodynamic lift needed to be considered during design processes. During high-speed manoeuvre, open-wheel race car has significant effect on aerodynamic behaviour. Lower value of drag force will increase the speed and increase the stability of the open-wheel race car. However, the drag force can be compensated by manipulating drag force into down force. As a result, it will push car downward and provide higher traction force on tire. Multiples studies have been conducted regarding open wheel race car aerodynamic [2] and performance [3-4]. Installation of vortex generator on rear side of sedan car can control the separation of airflow and increase the aerodynamics performances. Few studies have been performed to observe the influence of elements inside the wheelhouse on the vehicle aerodynamic. The aerodynamics load on inside the wheelhouse can be possible by using a mathematical model. On a contrary, flow behaviour inside the wheelhouse and suspension [5] can be complex and have too much recirculation regions. In their research, they agree that analysis on box on wheels, wheel tire set are not suitable for aerodynamic performance research.

In this paper, the previously designed and fabricated FV Malaysia race car [6-8] is undergone CFD analysis to observe the aerodynamics properties such as coefficient of drag (CD) and lift (CL) [9] at different speeds and meshes. Commercial CFD software ANSYS is used for the analysis. The results from the analysis can be used for further bodywork design improvement and racing advantages.

2. METHODOLOGY

The computer aided design (CAD) software CATIA is used to design the race car bodywork (Figure 1). Tire design has been kept as real as possible. The sport rim is not being design and replace with plain rims due to fact it has minimum influence on aerodynamics. The wheel diameter is 15. The front spoiler is being design to have better air flow. Generative Surface Design Function has been used. On this function the outer surface being created by using a line projection and trimming method. ANSYS is used for the CFD simulation to produce coefficient of drag and lift. The pressure distribution on the bodywork is also studied (Figure 1). The shape of the vehicle body has high influence towards the performance and coefficient of drag. In this study, CFD analysis of flow over the open wheel race car is carried out for speed of 15, 25 and 35 m/s. Average speed is determined by considering the race car will be able to maintain at an average speed on straight course.

3. RESULTS AND DISCUSSION

Results obtained are plotted in Figure 3 and Figure 4. From Figure 2, it can be observed that on initial number of iterations, the value of (CD) is unstable. It has fluctuation from 0.1703 up to 0.74 since it still in converging process. The data was fully converged on 120th number of iterations. The average of CD is being compute from 90th iterations to 120th iterations which is right before the converging happens and the value is 0.4610. From Figure 3, the CL is in negative value; any positive value of coefficient of lift will provide lift force and reduce traction on control for each tire. The average CL is computed from 80th iteration towards 120th iterations and the value is -0.300. Three different speeds are set in order to see the reliability of CD and CL. Table 1 shows the results of CD and CL at different speeds. At 15 m/s the race car generates averagely 0.426 of CD and -0.28 for value CL. When the race car travels at 25 m/s the CD is slightly increased to 0.461 and CL is reduced to -0.31. Same pattern is obtained when speed is increased up to 35 m/s.

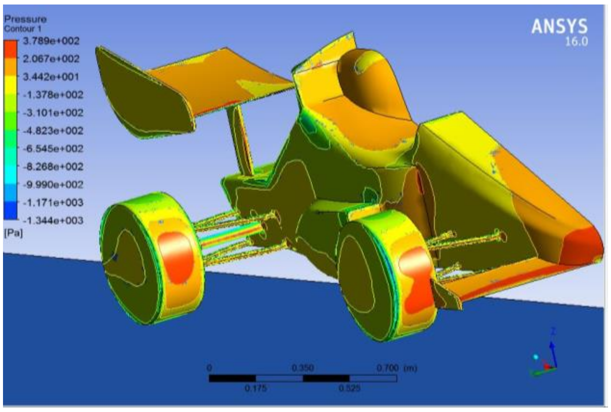

Figure 1 Pressure distribution of FV bodywork.

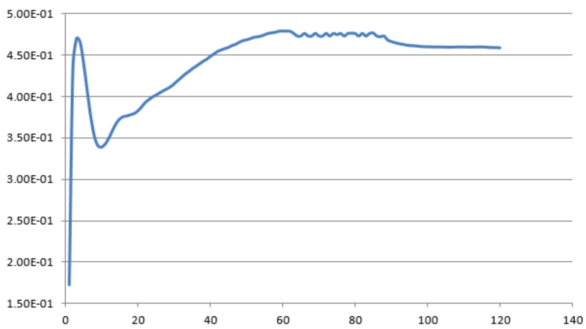

Figure 2 Coefficient of drag.

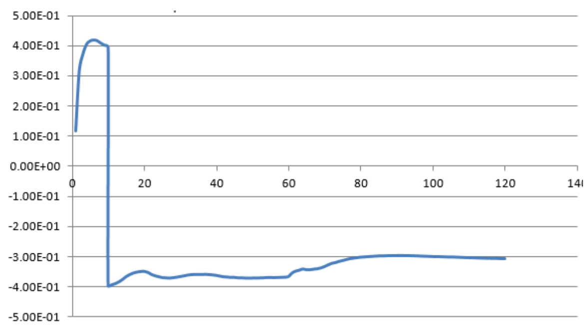

Figure 3 Coefficient of lift.

Table 1 CD and CL at different speeds.

Speed	Coefficient of Drag	Coefficient of Lift
15 m/s	0.426	-0.28
25 m/s	0.461	-0.30
35 m/s	0.479	-0.31

4. CONCLUSION

In this research, bodywork has been designed using Generative Design Function and being solidified by using part design. In order to make the analysis more realistic, the design was included with the tire, front spoiler and rear spoiler. The interface of rim tire has been simplified in order to have better flow on tire since. Complete bodywork analysis was carried out through CFD simulation. Whenever the vehicle is accelerating, the CD will increase and the CL will reduce, thus increase down force and provide extra traction force. The value of CD and CL are comparatively verified [9] at different speeds.

ACKNOWLEDGEMENT

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REFERENCES

[1] M.N. Sudin, M.A. Abdullah, F.R. Ramli, M.T. Musthafah and S.A. Shamsudin, "Review of research on vehicles aerodynamic drag reduction methods," *International Journal of Mechanical and Mechatronics Engineering*, vol. 14, no. 2, pp. 35-47, 2014.

[2] J. Newbon, R. Domin and D. Sims-Williams, "CFD investigation of the effect of the salient flow features in the wake of a generic open-wheel race car," *SAE International Journal of Passenger Cars-Mechanical Systems*, vol. 14, no. 8, 2015-01-1539, pp. 217-232, 2015.

[3] M.A. Abdullah, M.A. Salim, M.Z. Mohammad Nasir, M.N. Sudin, and F.R. Ramli, "Dynamics performances of Malaysian passenger vehicle," *ARPN Journal of Engineering and Applied Sciences*, ISSN 1819-6608, vol. 10, no. 17, pp. 7759-7763, 2015.

[4] M.A. Abdullah, M. Ibrahim, M.A.H. Abdul Rahim, "Experimental and analysis of vehicle dynamics performance based on driving behavior", *Journal of Mechanical Engineering*, vol. SI1, no. 1, pp. 193-206, 2017.

[5] M.A. Abdullah, N. Tamaldin, M.A. Mohamad, R. S. Rosdi, and M.N.I. Ramlan, "Energy harvesting and regeneration from the vibration of suspension system", *Applied Mechanics and Materials*, vol. 699, pp. 800-805, 2015.

[6] M.A. Abdullah, M.R. Mansur, N. Tamaldin, and K. Thanaraj, "Development of formula varsity race car chassis", *IOP Conference Series: Materials Science and Engineering*, vol. 50, no. 1, pp. 012001, 2013.

[7] M.A. Abdullah, M.R. Mansor, M.Mohd Tahir, S.I. Abdul Kudus, M.Z. Hassan and M.N. Ngadiman, "Design, analysis and fabrication of chassis frame for utem formula varsitytm race car", *International Journal of Mining, Metallurgy & Mechanical Engineering*, vol. 1, no. 1, pp. 75-77, 2013.

[8] M.A. Abdullah, A.H. Mohamad and F.R. Ramli, "Design, Analysis and Fabrication of Fixed-Base Driving Simulator Frame", *The Journal of Engineering and Technology*, vol 4, no. 2, pp. 85-102, 2013.

[9] S. Hetawal, M. Gophane, B.K. Ajay, Y. Mukkamala, "Aerodynamic study of Formula SAE car," *Procedia Engineering*, vol. 97, pp. 1198-1207, 2014.

Effect of temperature on friction of bio-lubricant under high loading capacity

A.R. Zulhilmi[1], N.A.B. Masripan[1,2,*], S.O.A.N. Hayder[1], M.F.B. Abdollah[1,2], G. Omar[1,2], M.I. Syakir[1,2], A.H. Nurfaizey[1,2], S.H.S.M. Fadzullah[1,2]

[1)] Faculty of Mechanical Engineering, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[2)] Centre for Advanced Research on Energy, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia

[*]Corresponding e-mail: norazmmi@utem.edu.my

Keywords: Bio-lubricant; banana peel; tribological properties; high loading capacity

ABSTRACT – The purpose of this study are to produce crude oil from Banana Peel (BP) as bio additives in paraffin oil, as well as to determine their physical and tribological properties as bio-lubricant under severe operation conditions to identify their ability for lubricants. Tribological performance of Banana Peel (BP) as a bio-lubricant was tested using four-ball test machined under extreme pressure conditions, according to ASTM D2783-03. Experimental results showed significant improvement in overall performance with increased BP content compared with paraffin oil (PO) through Coefficient Of Friction parameter (COF) at 100˚C, lower value of COF which 0.086 for 50%BP followed by 20%BP, 5%BP and 100%PO at values 0.089, 0.456 and 0.595 respectively.

1. INTRODUCTION

Tribology can be defined as the science and technology of interacting surface in relative motion which are present in various machined elements [1]. In almost every aspect of our daily lives, some appearances of tribology such as sliding, brushing, gripping, holding, machinery works, friction between skin and clothes, movement of artificial hip joints etc. [2]. Friction is the force that resisting the relative motion of solid surface, fluid layers and material elements sliding against each other. There are many types of friction like, lubricated friction, fluid friction and dry friction. An important consequence of many types of friction is wear which lead to decline in performance and/or damaged to components. Wear can be defined as undesired removal of material due to mechanical action [3]. The rough surface (deep valley of asperities) that formed helped to create an oil reservoir of the lubricant and prevented metal to metal contact [4].

Lubrication is the process or technique employed to reduced friction between two surfaces. Most of friction and wear are created during the start-up and shut down of engines, whereas Boundary Lubrication (BL) occur at low speed [5]. The major reasons of using lubricants in engines are to control friction properties, reduce wear, and improve the efficiency. The bio-based lubricant is promising to protect the surfaces from wear and damage in comparison with the mineral oil due to lower value of dynamic pressure [6].

In this study, Banana Peel (BP) had been investigated as an additive in lubrication system. This is a novel attempt to use banana peel in lubrication system. Hence, it is important and necessary to evaluate the characteristics of BP to show their effect of temperature on friction performance to test their validity in industry application. The dispersion of banana peel in paraffin is stable and smooth without any sedimentation problem. Moreover, oil shows good and promising tribological characteristic of lubricant [7].

2. METHODOLOGY

2.1 Material preparation

Cavendish banana skin or banana peel (BP) which is pericarp (outside skin) had been chosen as natural additives in paraffin oil. Paraffin oil as based-oil has been mixed with banana peels because of simple structure, unique tribological behavior and flexible for use under different percentage in preparation of lubrication samples.

2.2 Material and methods

There were four types of lubricant samples which are state in Table 1 below.

Table 1 Composition of lubricant samples.

Lubricant samples	Composition of Lubricant sample
Sample A	100% Pure Paraffin oil
Sample B	Paraffin oil +5% Banana Peel
Sample C	Paraffin oil +20% Banana Peel
Sample D	Paraffin oil +50% Banana Peel

Preparation of lubrication samples was determined by using Equation 1. Volume percentage was referred after solution was made by mixing two liquids. Total volume for each lubricant sample fixed to 100 ml that contained of banana peel and paraffin oil.

$$C\%\ v/v = \frac{V\ Substance}{V\ Solution} * 100\% \qquad (1)$$

2.3 Friction Test

Three design parameter were performed which are percentage of lubricant, temperature and load. The four sample of lubricant are test under the temperature of 27, 80 and 100℃. The factor of coefficient of friction had been taken into account as results to evaluate significant performance for different concentration of lubricant samples. Friction test were carried out according to standard test methods for measurement of coefficient of friction (COF) and extreme pressure (EP) properties of lubricants until obtaining welding point on four-ball testing, according to ASTM D2783-03 [8]. The test has been conducted for 30 minutes at 27, 80 and 100 ˚C on four samples.

3. RESULTS AND DISCUSSION

3.1 Analysis of coefficient of friction

The results of variation value of COF with applied loads for various lubricants under different temperature shown in Figure 1 below. Test at lower temperature could not supply reliable information about oxidative stability and sometimes cannot evaluate the presence of protective film or anti-oxidation compound [9]. At high temperature, a film was formed on the metal surface during thermal decomposition, containing effective compounds, resulting in the friction reduction thus COF reduction. Hence, for bio-lubricant, mechanism of friction reduction could be achieved by increasing the content of banana peel in paraffin oil, especially at 100 ˚C, for lubricant 20% BP and 50% BP.

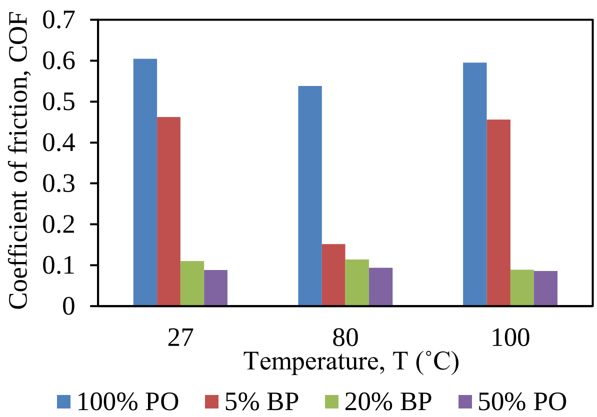

Figure 1: Effect of temperature (˚C) on value of Coefficient of friction, COF.

4. CONCLUSION

The Coefficient of friction was found to decrease with increase of banana peel content. The behavior of the lubricant under extreme pressure conditions became better with increase the of banana peel content. From the result, banana peel as natural additives have ability to improve physical and tribological properties of paraffin oil.

REFERENCES

[1] M. Nosonovsky and B. Bhushan, "Green tribology: principles, research areas and challenges.," *Philos. Trans. A. Math. Phys. Eng. Sci.*, vol. 368, pp. 4677–4694, 2010.

[2] L. Mattei, F. Di Puccio, B. Piccigallo, and E. Ciulli, "Lubrication and wear modelling of artificial hip joints: A review," *Tribol. Int.*, vol. 44, no. 5, pp. 532–549, 2011.

[3] I. Golshokouh, S. Syahrullail, and F. N. Ani, "Influence of normal load and temperature on tribological properties of jatropha oil," *J. Teknol.*, vol. 71, no. 2, pp. 145–150, 2014.

[4] M. I. R. Azmi, B. T. Tee, N. A. B. Masripan, and C. T. Chong, "Preliminary study of friction and wear on natural oil-based lubricants," *Proceedings of Malaysian International Tribology Conference*, pp. 220–221, 2015.

[5] W. Tuszynski, "An effect of lubricating additives on tribochemical phenomena in a rolling steel four-ball contact," *Tribol. Lett.*, vol. 24, no. 3, pp. 207–215, 2006.

[6] Z. H. Nazri, M. Z. M. Rody, M. F. B. Abdollah, S. A. Rafeq, H. Amiruddin, N. Tamaldin and N. A. B. Masripan, "Elastohydrodynamics lubrication for bio-based lubricants in elliptical conjunction," *Procedia Eng.*, vol. 68, pp. 123–129, 2013.

[7] A. H. Hamid, N. A. B. Masripan, M. M. B. Mustafa, R. Hasan, M. F. B. Abdollah, and R. Ismail, "Effect of banana peels as an additive on the tribological proporties of paraffin oil," *J. Teknol.*, vol. 77, no. 21, pp. 73–77, 2015.

[8] ASTM D 2783-03: Standard Test Method for Measurement of Extreme-Pressure Properties of Lubricating Fluids (Four-Ball Method), vol. 3. West Conshohocken: ASTM, 2003.

[9] F. B. Gonzaga and C. Pasquini, "A new method for determination of the oxidative stability of edible oils at frying temperatures using near infrared emission spectroscopy," *Anal. Chim. Acta*, vol. 570, no. 1, pp. 129–135, 2006.

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Experimental investigation on the relationship between current and alternator temperature and battery voltage

R.K. Mazlan[1,2,*], R.M. Dan[1,2], A. Putra[1,2], M.Z. Zakaria[1,2], A.H.A. Hamid[1,2]

[1)] Faculty of Mechanical Engineering, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[2)] Centre for Advanced Research on Energy, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia

[*]Corresponding e-mail: rozdman.edir@gmail.com

Keywords: Direct current (DC); battery voltage; alternator temperature

ABSTRACT – The objective of this research is to find the effect of alternator current output to the alternator temperature and battery voltage. The electrical power demands in an automobile have been rising rapidly for many years. This flow of direction is driven by the introduction of a wide range of new functionality in vehicles. Alternator produces DC and generates heat. The more power there is, the more heat is generated as a by-product. Also, as more current it produces, more voltage will drop at the battery. After 15 minutes, the more current output generates more temperature and drop more voltage at the battery.

1. INTRODUCTION

In our modern world, electricity plays a pivotal role in driving industry and powering the systems used to meet our agricultural, health care, educational, and commercial needs. The same goes for automobile industry. The electrical power demands or requirements in vehicle transportation have been rising rapidly for many years and are expected to continue to rise [1].

Since the invention of the automobile, power has been an issue. The evolving of technology has produced many research and inventions in order to provide more and more power to the car. The race in the automobile industry to invent product which can put out more power has been stimulated and encouraged by the latest and advanced technologies, and also the introduction and establishment of wide range of functionality in vehicles such as hi-fi speaker, LCD touch screen, LED, DVD player etc. which required more power from the car electrical power source.

More power means more work is done. And the more work is done, means the more heat it produces. Just like any other machinery, alternator also produces heat as a by-product [2]. Alternator produces alternate current (AC) which will then convert to direct current (DC) by diodes as the electrical consumer in the car required DC to operate [3]. When there is more demand, the more DC the alternator will generate and supplies. The input of the alternator is speed-dependent. The faster the speed of the alternator, the more current it can generate [4]. But the output of the alternator is demand-dependent. The alternator will only supply according to the total of demand. The more the alternator supply, the higher the efficiency of the alternator. Better efficiency can help in reducing fuel cost [5]. But as mentioned earlier, the alternator will produce more heat when the more current it supplies to the vehicle's electrical components.

The aim of this research is to prove the theory and investigate the relationship between current and temperature of the alternator, and also the effect of current to the battery voltage.

2. METHODOLOGY

For this research, a Proton Preve 1.6 Manual was used as the test subject. The car has an alternator with a rating of 12V 90A. The experiments were done at idle speed, repeatedly to get consistency and also average result.

The speed of the alternator is the constant variable, as the experiment was done on idle speed of the engine. A tachometer was used to measure the speed of the alternator. The time period for each experiment was set for fifteen minutes. The manipulated variable was the load being put to the alternator. The load was current demand from the electrical components in the car, which are front light and radio. The temperature is the responding variable. The alternator temperature was taken at the stator, and two points were measured.

The speed of the engine was measured by using timing light, while the speed of the alternator was measured by using a digital tachometer. The current (alternator output) was measured by using a AC/DC Clamp. Table 1 shows the electrical components as load, while Figure 1 is the schematic diagram of the experiment.

Table 1 Electrical components as load.

Properties	Load (Amp)
Idle (engine start)	12
Front lamp + Radio + Passenger light	25

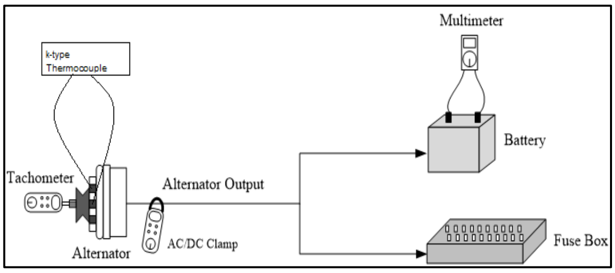

Figure 1 Schematic diagram of experiment.

3. RESULT AND DISCUSSION

Figure 2 shows the result for load = 12 A, while Figure 3 shows the result for load = 25 A.

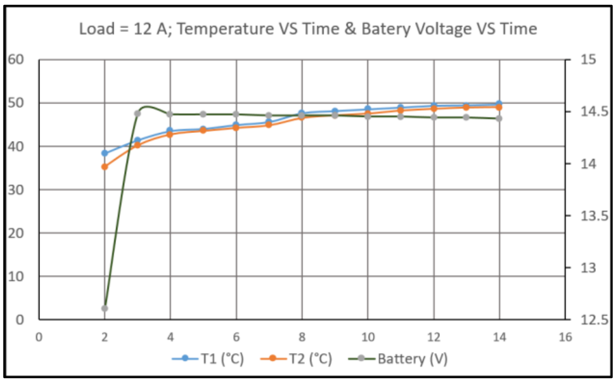

Figure 2 Temperature VS Time and Battery Voltage VS Time for current = 12 Ampere.

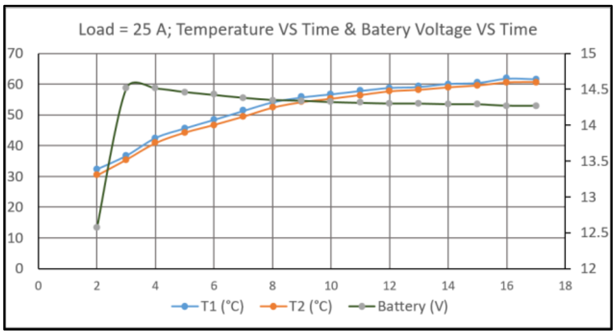

Figure 3 Temperature VS Time and Battery Voltage VS Time for current = 25 Ampere.

As shown in Figure 2 and Figure 3, it is proved that as more current being produced, the more heat was generated as well. The temperature at alternator stator rises over time until it reached a point where the temperature met normal operating temperature, which the temperature only increases with very slight differences. Environment factor is the main reason for the increase of temperature of alternator after it has reached a point where it has become equivalent to ambient temperature. The alternator has a tendency to absorb heat from the engine, in which will add temperature to itself. At time 14 minutes, the highest temperature at alternator stator for current equal to 12 A is 49°, while the highest temperature at alternator stator for current equal to 25 A is 59°.

Both Figure 2 and 3 also show that the more current drawn from the alternator, the more voltage will drop at the battery over time. At time 14 minutes, the battery voltage for current equal to 12 A is 14.37 V, while the battery voltage for current equal to 25 A is 14.23 V. Although the voltage drops, this reading is actually still good, as for a battery to be considered as fully charged, the voltage value should be above 13.8 V. Alternator will only charge the battery after it has fully supplied the demand from electrical consumers [6-7], and if the demand is high, it will take more time to fully charge the battery [8-9].

4. CONCLUSION

Proton Preve used in this research has an engine speed of 750 RPM during idle and has an alternator speed of 1560 RPM during idle. This tells us that the speed ratio of the engine to the alternator is 1:2. The fact that the battery voltage value is above 14.2 after 15 minutes at current equal to 25 A tells that the alternator can still support the demand of 25 A of current. But this action resulted in temperature rise and drops of battery voltage. This knowledge hopefully will bring awareness to the public about the consequences that will occur when they put loads to the alternator, as nowadays many people love to install high power devices such as fancy LED light and hi-fi stereo in the vehicle.

ACKNOWLEDGEMENT

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REFERENCES

[1] C. Cho and D. Crecelius, “Vehicle alternator/generator trends toward next millennium,” *Veh. Electron. Conf. 1999*, pp. 433–438, 1999.

[2] H. Sinks, B. M. Bradfield, H. Sinks, L. Cooling, H. Sink, L. Cooling, P. Conversion, T. Design, and P. This, “Thermal Design Challenges in Automotive Alternator Power Electronics,” pp. 2–5, 2002.

[3] E. J. Beyn, “The 12 Volt Doctor’s Alternator Book.”

[4] D. J. Perreault and V. Caliskan, “Automotive power generation and control,” *Power Electron. IEEE Trans.*, vol. 19, no. 3, pp. 618–630, 2004.

[5] M. Bradfield, “Improving Alternator Efficiency Measurably Reduces Fuel Costs,” *Remy, Inc*, pp. 1–31, 2008.

[6] T. Dc, “Automobile Electrics : The Charging System The Basics,” pp. 1–18.

[7] R. P. Sharma, *A Course In Automobile Engineering*. New Delhi: Dhanpat Rai Publication (P) Ltd, 2009.

[8] K. R. Sullivan, “Understanding the Alternator,” 2002.

[9] H. S. Reyat, *The Automobile*. New Delhi: S. Chand & Compant Ltd, 2007.

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Commercial vehicle anti-rollover suspension system

Munaliza Ibrahim[1], Mohd Azman Abdullah[1,2,*], Fathiah Mohamed Jamil[1], Fauzi Ahmad[1,2]

[1)] Faculty of Mechanical Engineering, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[2)] Centre for Advanced Research on Energy, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia

[*]Corresponding e-mail: mohdazman@utem.edu.my

Keywords: Anti-rollover suspension; PID; 7DOF vehicle model

ABSTRACT – This study investigated the performance of PID controller for active anti-rollover suspension system. Road disturbance such as the bump has been applied to disturb the suspension system. Based on the road disturbance, the performances of passive and active suspension systems were investigated. A simple tuning method was used to obtain the optimum values of the PID controller parameters. A comparative study was carried out between active and passive suspension systems. The active suspension system with PID controller is proven to perform better than the passive suspension system.

1. INTRODUCTION

Suspension system is one of the important components of a vehicle which physically separates the vehicle's body [1] from its wheels [2-3]. The suspension system has its own mechanism which gives a great comfort and safety to driver and passengers inside the vehicle especially when the vehicle hitting a bump or a hole and also due to cornering [4-6]. Active suspension system has been investigated in many different controllers previously. The control methods used in the automotive industry usually are simple control mechanisms, such as PID controller. The performance of these controllers managed to provide good handling and ride comfort as compared to the passive system. PID controller was also introduced and the result showed active system has reduced the peak of the overshoot of the sprung mass displacement as compared to passive system. The popularity of PID controllers can be attributed partly to their good performance in a wide range of operating conditions.

2. METHODOLOGY

The previous verified model of 7DOF full car ride model is used in this paper. A simple method of tuning PID controller is also applied. The controller parameters are tuned automatically to overcome the disturbance [7-8]. Taking the road profile of a bump as a disturbance, the value of proportional gain was obtained, therefore, using automatic tuning, the best PID controller is obtained with the best combination of P, I and D values. Equation (1) to (3) show the decoupling transformation equations to produce 4 individual actuator forces (F_{afl}, F_{afr}, F_{arl}, F_{arr}) and moments (M_p, M_r). The controller model as shown in Figure 1. The system was given a step input of 0.1 m at 1 second at front left wheel, z_{rfl} (Figure 2).

$$F_{afl} + F_{afr} + F_{arl} + F_{arr} = F_z \quad (1)$$

$$(F_{afl} + F_{afr})l_f + (F_{arl} + F_{arr})l_r = M_p \quad (2)$$

$$(F_{afl} + F_{afr})\frac{t}{2} + (F_{arl} + F_{arr})\frac{t}{2} = M_r \quad (3)$$

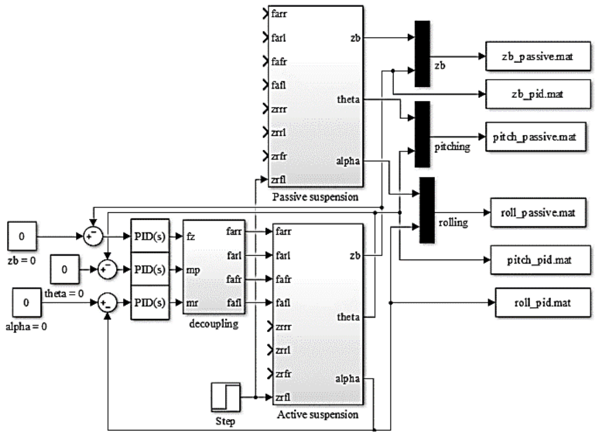

Figure 1 Controller model.

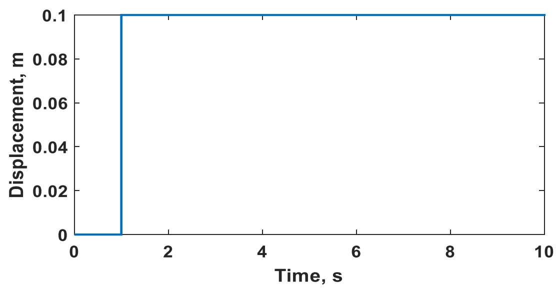

Figure 2 Step input of 0.1 m to the system.

3. RESULTS AND DISCUSSION

All figures below show the results comparison between active suspension (with PID controller) and passive suspension. The active suspension performance is plotted with red solid line while the passive suspension is plotted with blue dotted lines. The result displays three graphs for roll angle, pitch angle and vertical displacement. From the graph (Figure 3), it shows that the vehicle come to the level condition at 2 second after the input given at 1 second for active suspension compared to the passive suspension where the vehicle maintain declined at 0.03 m high of displacement or degree of roll and pitch.

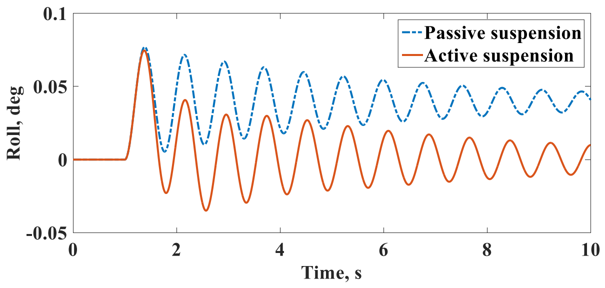

Figure 3 Degree of roll angle.

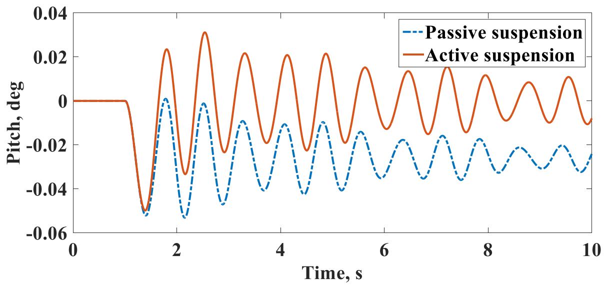

Figure 4 Degree of Pitch angle.

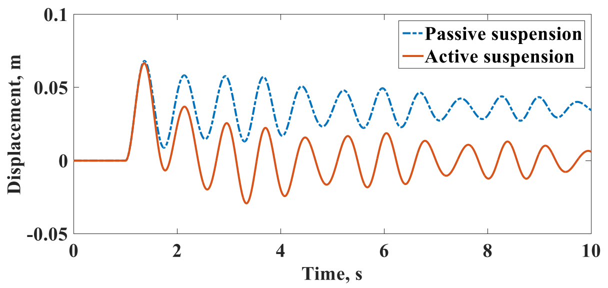

Figure 5 Front left wheel Vertical displacement.

The active (with PID controller) suspension system has indicated a better performance as compared to the passive when the vehicle hitting the bump. The results of roll angle and displacement of vehicle body oscillates with a same magnitude for both systems at 0.08 degree and meter respectively. While the result of pitch angle oscillates at -0.05 degree for both systems due to assumption for moment with clock wise direction will give positive magnitude.

Passive suspension system does not give enough stability for the passenger and driver when the vehicle experiences this road disturbance. By referring to the trend line of the graph, most probably for all figures, the passive suspension trend is inclining while the active suspension trend is declining to zero value. Table 1 shows the value for offset improvement for the suspension system.

4. CONCLUSION

The PID controller has been successfully implemented in an active anti-rollover suspension system through simulation study. Throughout the research, a simple method for tuning PID controller was applied in this system. The active suspension isolated the effect of the road condition from the vehicle body and maintain the comfort of the passenger by keeping the vertical displacement about zero level. The results of controller performance analysis demonstrate the potential of simple PID as a controller for anti-rollover system for commercial vehicle.

Table 1 Check improvement in offset.

Motion	Mean offset		% Improvement
	Passive	Active	
Roll angle	0.0395	0.0038	90.4
Pitch angle	-0.0259	-0.00184	92.9
Displaceme	0.0355	0.0035	90.3

REFERENCES

[1] M.A. Abdullah, M.R. Mansur, N. Tamaldin, and K. Thanaraj, "Development of formula varsity race car chassis", *IOP Conference Series: Materials Science and Engineering*, vol. 50, no. 1, pp. 012001, 2013.

[2] D. Chu, X.Y. Lu, C. Wu, Z. Hu and M. Zhong, "Smooth sliding mode control for vehicle rollover prevention using active antiroll suspension," *Mathematical Problems in Engineering*. 2015 May 31; 2015.

[3] X. Shao, "Modeling and model analysis of a full-car fitted with an anti-pitch anti-roll hydraulically interconnected suspension," *SAE Technical Paper*, 2014 Apr 1; 2014.

[4] M.A. Abdullah, N. Tamaldin, M.A. Mohamad, R. S. Rosdi, and M.N.I. Ramlan, "Energy harvesting and regeneration from the vibration of suspension system", *Applied Mechanics and Materials*, vol. 699, pp. 800-805, 2015.

[5] M.A. Abdullah, M.A. Salim, M.Z. Mohammad Nasir, M.N. Sudin, and F.R. Ramli, "Dynamics performances of Malaysian passenger vehicle," *ARPN Journal of Engineering and Applied Sciences*, ISSN 1819-6608, vol. 10, no. 17, pp. 7759-7763, 2015.

[6] M.A. Abdullah, M. Ibrahim, M.A.H. Abdul Rahim, "Experimental and analysis of vehicle dynamics performance based on driving behavior", *Journal of Mechanical Engineering*, vol. SI1, no. 1, pp. 193-206, 2017.

[7] M.F. Md Isa, M.H. Harun, M.A. Abdullah and N.S. Muhammad, "Force tracking control of MR damper and controller parameter optimization using sensitivity analysis method," *Proceedings of Mechanical Engineering Research Day*, 2015, pp. 61-62.

[8] M.A. Abdullah, J.F. Jamil and A.E. Mohan, *Vehicle dynamics modeling & simulation*, Malacca: Centre for Advanced Research on Energy (CARe), Faculty of Mechanical Engineering, Universiti Teknikal Malaysia Melaka; ISBN: 978-967-0257-78-5; 2016.

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Drag and lift evaluation on redesigned front area of 1990's land rover defender

A.M.H.S. Lubis[1,2,*], M.I.F. Rosley[1,2], M.S.A. Razak[1,2], A.A.M. Damanhuri[1,2], A.N.M. Amin[1,2]

[1)] Faculty of Engineering Technology, Universiti Teknikal Malaysia Melaka,
Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[2)] Centre for Advanced Research on Energy, Universiti Teknikal Malaysia Melaka,
Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia

[*]Corresponding e-mail: munir@utem.edu.my

Keywords: Drag coefficient; lift coefficient; land rover

ABSTRACT – The 1990's Land Rover Defender was designed with a feature that was not possess aerodynamics feature. However, people put more interest to this type of car for on-the road purpose. This project is meant to design and perform computational fluid dynamic (CFD) analysis of a redesigned front area of 1990's Land Rover Defender. The car was modeled using CATIA V5 software. Altairs Hyperwork Virtual WindTunnel software to simulate aerodynamic reaction of the car models. The wind tunnel model with the size of 39.177 m x 8.690 m x 8.690 m. The CFD simulation was performed using the Reynolds-Averaged Navier-Stokes (RANS) solver. Value of drag coefficient (CD) obtained from the analysis was 0.639 for the original model and 0.566 for the redesign model. The lift coefficient (CL) obtained was -0.234 for the original design and -0.216 for the redesign model. The modification made results in the change of wind velocity distribution and direction thus give reduction in drag and lift coefficient.

1. INTRODUCTION

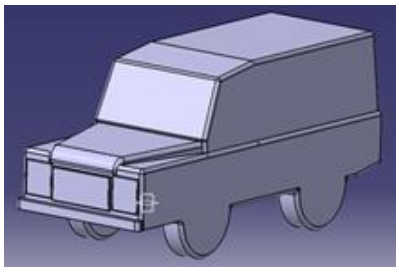
Front area of all cars, including the front bonnet, front grill, both sides of front eyebrows, are important for the aerodynamics features of the car. Land Rover vehicles were created for heavy road style (off-road) and specialist in four-wheel drive vehicles. Old Land Rover Defender design was designed with a feature that was not possess aerodynamics feature. This possibly due to their purpose to be used for off-road application. However, people have put more interest to this type of car for on-road purpose now.

The air flow around a vehicle is complex and nonlinear interactions between different parts of the vehicle are present. Drag and lift coefficient are important parameter vehicle design due to their direct effect to speed, fuel consumption, and comfort of a vehicle. Aerodynamic drag (D) depends on the size of a vehicle especially projected frontal area [1].

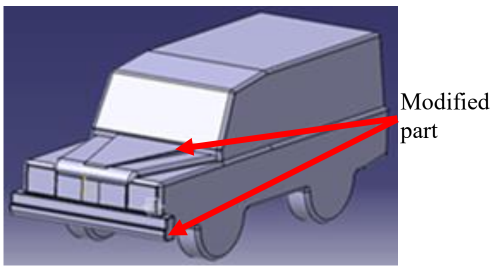
CFD analysis commonly used to evaluate a design related to fluid flow before full scale model produced. Wind tunnel simulations is employ CFD analysis to study aerodynamic loads thus can reduce the number of physical wind tunnel test during a vehicle development. This project is meant to design and perform computational fluid dynamic (CFD) analysis of a redesigned front area of 1990's Land Rover Defender.

2. METHODOLOGY

To perform effective control solution, it is efficient to work on simplified geometries which able to reproduce the physical phenomena happened on a real vehicle [2]. A simplified geometry of 1990's Land Rover Defender car was modelled using CATIA V5 software. A modified model of the car was made where modification to hood and front bumper was applied. Both of this parts are considered give major effect to wind to flow across the car when it moves forward.



(a)



(b)

Figure 1. Land rover defender model used in the analysis, (a) original design and (b) redesigned model.

The CFD analysis was performed on Altairs Hyperwork Virtual WindTunnel software to simulate aerodynamic reaction of the car models. The wind tunnel model was set to the size of 39.177 m x 8.690 m x 8.690 m and car model with ratio of 1:1 was set inside the wind tunnel as shown in Figure 2. The model was mesh so around 15e10^6 elements were created in this steps. The CFD simulation was performed using the Reynolds-Averaged Navier-Stokes (RANS) solver. The inflow speed of 25 m/s (90 km/h) was set to positive Z-direction and the outlet pressure was set to atmospheric pressure. After analysis performed, the software automatically calculates and report the drag and lift coefficient of the model.

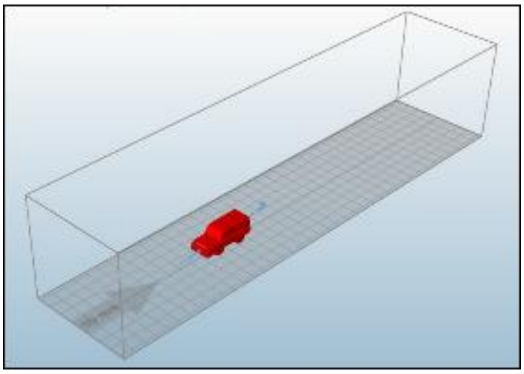


Figure 2. Wind tunnel model.

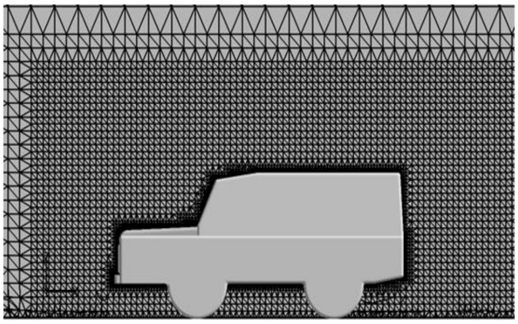


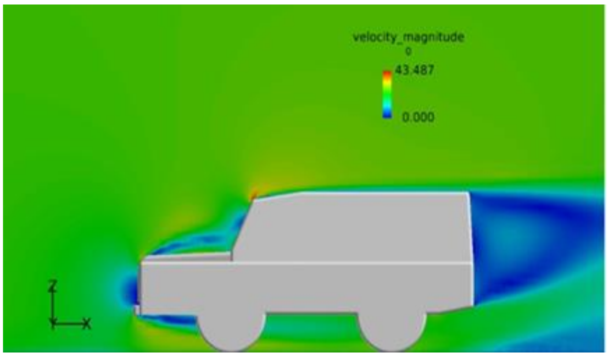
Figure 3. Meshed model.

3. RESULTS AND DISCUSSION

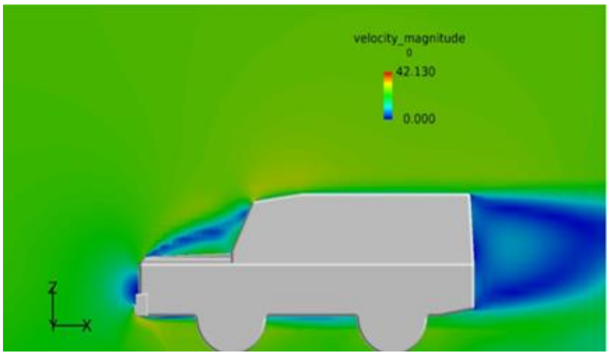
One of advantages of Altairs Hyperwork Virtual WindTunnel software is it directly report the value of drag and lift coefficient. In this analysis, the value of drag coefficient, C_D, reported was 0.639 for the original model and 0.566 for the redesign model. In term of lift coefficient, the lift coefficient C_L obtained from the analysis was -0.234 for the original design and -0.216 for the redesign model.

Figure 4 show velocity contour for the original model and the redesign model. From Figure 4a, it can have observed that the maximum velocity magnitude of 43.487 ms^{-1}. The maximum velocity magnitude was able to reduce to 42.130 ms^{-1} by the proposed design. It also observed that the direction of the streamline also altered due to modification of the hood. In the original design, the hood is relatively flat thus the wind was directly point toward the windscreen which affect to the aerodynamic drag of the car. By adding an inclined step to the hood, it is observed that the wind direction was altered to the top of the car thus reduce the drag coefficient of the car.

From the velocity contour shown in the Figure 4.5. It can be seen that the underbody flow was changed by modify the front bumper. In this project the front bumper was modified by increasing bend radius on lower side of the bumper. Modification of the front bumper show the change of underbody flow distribution and air vortex at the back of the car which give effect in the reduction of drag and lift coefficient as well. The drag caused by underbody flow passing through the underside of heavy vehicles deserves as much attention as that attributed to upper and front body flows [3]. The under body flow of a heavy vehicle such as tractor-trailer contributes approximately 30% to the total aerodynamic drag [4].



(a)



(b)

Figure 4. Velocity contour of (a) the original model and (b) redesign model.

4. CONCLUSION

A modification of the front side of and produced drag coefficient is 0.566 with 11.42% of changes from original design and lift coefficient -0.216 with 7.69% of changes from the original design. The modification made basically change incident area of the wind velocity which results in the change of wind velocity distribution and direction thus give reduction in drag and lift coefficient.

REFERENCES

[1] H. Chowdhury, H. Moria, A. Ali, I. Khan, F. Alam and S. Watkins, "Study on aerodynamic drag of a semi-trailer truck," *Procedia Engineering*, vol. 56, pp. 201-205, 2013.

[2] A. Kourta, and P. Gilliéron, "Impact of the Automotive Aerodynamic Control on the Economic Issues," *Journal of Applied Fluid Mechanics*, vol. 2, no. 2, pp. 69-75, 2009.

[3] B. G. Hwang S. Lee E. J. Lee, J. J. Kim M. Kim, D. You, S.J. Lee, "Reduction of drag in heavy vehicles with two different types o f advanced side skirts," *J. WindEng.Ind.Aerodynamics*, vol. 155, pp. 36–46, 2016.

[4] R.M. Wood, "A discussion of a heavy truck advanced aerodynamic trailer system". In *Proceedings of the 9th International Symposium on Heavy Vehicle Weights and Dimensions*, University Park, 2006.

Development of 3-cylinder composite engine: Analysis on strength and fuel efficiency

N. Tamaldin[1,2,*], S.R. Ruslan[1], A.K. Mat Yamin[1,2], A. Md Saad[1,2], M.F.B. Abdollah[1,2]

[1)] Faculty of Mechanical Engineering, Universiti Teknikal Malaysia Melaka,
Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
[2)] Centre for Advanced Research on Energy, Universiti Teknikal Malaysia Melaka,
Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia

[*]Corresponding e-mail: noreffendy@utem.edu.my

Keywords: Composite engine; ICE composite

ABSTRACT – Engine is the important part of automotive industry. From time to time, all automotive manufacturer tries to develop the very powerful engine due to high demand of performance of the engine. Basically, vehicle performance from speed analysis is depend on the weight of vehicle. Reduction on weight can give the best fuel efficiency. Averages car engine without the transmission system weights is about 158 kilograms. To give an example of typical weights, a small car engine and transmission weights around 151 kilograms, and a large car engine with transmission weights around 272 kilograms. Development of composite engine can reduce the weight about 30 - 40%. the type of composite material use in this research in fiber-reinforced polymer.

1. INTRODUCTION

A composite material is a material made from two or more constituent materials with significantly different physical or chemical properties. Composites are formed by combining materials together to form an overall structure with properties that differ from the sum of the individual component. The individual components remain separate and distinct within the finished structure. The new material may be preferred for many reasons, which are stronger, lighter, or less expensive when compared to the traditional materials. There are two main categories of constituent materials which is a matrix and reinforcement. The matrix material surrounds and supports the reinforcement materials by maintaining their relative positions. The reinforcement's material imparts their special mechanical and physical properties to enhance the matrix properties. Currently, there are so many type of composite materials such as mortars, concrete, reinforced plastics (fiber-reinforced polymer), metal composites and ceramic composites.

The use of composite material has been grown up because of their improvement properties over conventional material. Composite have high specific modulus and also the strength similar with the steel material. Otherwise, the composite is lighter and because of that, they can improve of fuel efficiency. The core of the engine is the cylinder, with the piston moving up and down inside the cylinder. The engine described in my research has three cylinder. That is typical of most lawn mowers, but most cars have more than three cylinder such a four, six and eight cylinders. In a multi-cylinder engine, the cylinders usually are arranged in one of three ways which is inline, V or flat [1].

The application of modern systems for pollution protection increases the mass of the vehicles which require the use of new (lighter) materials for the manufacturing of the engine and its constituent elements. The engine weight reduction leads to a decrease of fuel consumption, and therefore the environmental pollution. For production of engine blocks, cylinder liners, connecting rods, crankshafts and camshafts, the conventional material are commonly use such an aluminum, magnesium, titanium and their alloys. Engine parts, such as the cylinder casing, could shed up to 20% of its weight if it were made of fiber-reinforced plastic rather than aluminium, without additional costs.

A car's powertrain system, which includes the engine, accounts for a large proportion of the vehicle's weight. Until now, car makers have relied on aluminium to reduce the weight of engine components such as the cylinder block. However, in the future, car manufacturers will be able to achieve further weight savings by designing cylinder blocks in which certain parts are made of fiber-reinforced plastics. The materials used have to be able to withstand extreme temperatures, high pressure and vibrations without suffering damage. Depending on the application need, how the new composite material is to be fabricated and used to solve the performances issues is the main focus of this research.

2. METHODOLOGY

There are so many focus on research about composite material in automotive component. The objective of all researcher done for composite analysis is to reduce the weight then increase the performance. In automotive industry, composite material already uses in the external part, such a bumper, dashboard and accessories part. However, for engine part, conventional material like cast iron and aluminium was use due to the high temperature and strength. With new technology, the usage of composite material in engine part can be apply with some research from automotive manufacturers.

The paper written by Lee [2] describes the development of a hybrid valve lifter composed of composite and steel for use in a valve train in an automotive internal combustion engine. Valve lifter is a mechanical part that transmits the motion of the camshaft to the rest of valve train. Conventional valve lifter is produce from steel and aluminium and the weight is about 0.048kg. The weight of prototype hybrid valve lifters is 0.017kg. The composite hybrid valve lifter is separate into two pieces, which is steel cap and composite skirt. The important thing to consider and control is the method to join the steel cap and composite skirt. There are too many process of joining between two materials. The researcher uses the interference fit by radial interference as a method to assemble both two material parts. The tests showed that hybrid valve lifter were sufficiently durable to withstand the test loads.

The paper written by Tiruvenkadam et al. [3] was presented about the development a material to replace the current conventional cast iron cylinder liner (CL) with an improved performance and reduced emission biodiesel engine. They divided the experiment into two section. The first section is discussed about the Al nano hybrid composite sample preparation and selection based on frictional characteristics analysis. The second section is discussed about the development of a lightweight NL using suitable sample. They also tested the sample under real time experiment to find the accurate data. During the experimental process, they completed it by follow the relevan step include matrix and reinforcement’s preparation, fabrication of nano hybrid composite, photomicrograph examination, evaluation of tensile strength, pin on reciprocating plate test and plan experiments. At the end of this research, the percentage reinforcement and sliding distance do not affect the friction coefficient of the fabricated composite sample. Otherwise, the tensile strength and coefficient of friction are the most important properties for NL to reduce energy losses.

The paper written by Arsha et al. [4] was presented investigation aims at design, fabrication and evaluation of functionally graded automotive piston using in-situ primary silicon reinforced A390 aluminium composite by centrifugal casting technique with a view of obtaining improved thermo mechanical properties at specific locations. The dies are designed and fabricated so as to obtain the primary silicon rich region towards the head portion of the piston. FGM pistons with A390 and A390-0.5%Mg are produced. They are characterised along the vertical cross section of the piston from piston head towards the skirt by microstructural, chemical, mechanical, thermal and tribological characterisations methods. The results are also compared with that of gravity cast piston. Microstructure and chemical composition analysis of FGM piston shows graded distribution of primary silicon from the head portion of the piston towards skirt and a eutectic composition in the skirt region. That yields an increase in hardness towards the head region. The wear testing revealed that the gradation also resulted in a remarkable enhancement of the wear properties of the piston head.

The paper written by Hayashi [5] was discussed about the application of metal matrix composite (MMC) in engine cylinder blocks and also brake disks. The researcher designed the cylinder block with cylinder bore reinforcement by alumina and carbon fibers. Basically, there have to way to produce MMC by using new intermediate-pressure die-casting (NDC) method and another one is high-speed, high-pressure die-casting (HPDC) method. Then, the researcher **uses** the conventional HPDC for MMC cylinder block production due to the accelerated application problem. The HPDC product show a value several points higher than its predecessor. Then, the MMC cylinder block allow the combination of high performance with weight reduction compared to the conventional cylinder block. Conventional brake disks id designed from cast iron material. The researcher developed the aluminium MMC disk brake to achieve a reduction in weight. He **uses** an infiltrating method to produce MMC disk brake. From the test result, the MMC disk brake have advantages. Even, with weight reduction, the MMC disks brake have a good cooling performance. It also **has** high stability against deceleration and temperature. Then, the MMC disk brake can improved resistance to brake judder. It also easy to design, which is can prevent brake noise.

The paper written by Ashori [6] was discussed about wood plastic composite (WPC) which is a very promising and sustainable green material to achieve durability without using toxic chemicals. WPC refers to any composites that contain plant fiber and thermosets or thermoplastics. In comparison to other fibrous materials, plant fibers are in general suitable to reinforce plastics due to relative high strength and stiffness, low cost, low density, low CO_2 emission, biodegradability and annually renewable. Plant fibers as fillers and reinforcements for polymers are currently the fastest-growing type of polymer additives. From a technical perspective, these bio-based composites will enhance mechanical strength and acoustic performance, reduce material weight and fuel consumption, lower production cost, improve passenger safety and shatterproof performance under extreme temperature changes, and improve biodegradability for the auto interior parts. Automotive components including plant fibers are currently being used by many vehicle manufactures. The researcher found the problem due the poor compatibility exhibited between the fibers and the polymeric matrices. However, the researcher solves the problem by chemical coupling and compatibilizing agents. This researcher of WPC that only use for external component of automotive part such front and rear door linens, boot linens, seat back, sunroof sliders and headliners. There is no discussed about the potential of WPC into engine part [7].

The potential automotive parts suitability from various composite materials are engine block, pisto, crankshaft, camshaft.

3. SUMMARY

In this project of develop the 3-cylinder composite engine that can reduce the weight about 30 – 40%. The MMC engine block already designed according to the T. Hayashi. But in this research, the researcher tries to develop engine block by using fiber-reinforced polymer that can give lighter engine block. Otherwise, the piston, connection rod, crankshaft and camshaft also will develop using the fiber-reinforced polymer. The materials used have to be able to withstand extreme temperatures, high pressure and vibrations without suffering damage. This is done properly with strength analysis.

REFERENCES

[1] S.P. Parker, “Engine; Concise Encyclopedia of Science and Technology,” 3rd Edition, McGraw-Hill, Inc., 1994.

[2] S.W. Lee, “Composite hybrid valve lifter for automotive engines,” *Composite structures*, vol. 71, no. 1, pp. 26-33, 2005.

[3] N. Tiruvenkadam, P.R. Thyla, M. Senthilkumar and M. Bharathiraja, “Development of optimum friction new nano hybrid composite liner for biodiesel fuel engine,” *Transportation Research Part D: Transport and Environment*, vol. 47, pp. 22-43, 2016.

[4] A.G. Arsha, E. JayakumarT.P.D. Rajan, V. Antony and B.C. Pai, “Design and fabrication of functionally graded in-situ aluminium composites for automotive pistons,” *Materials & Design*, vol. 88, pp. 1201-1209, 2015.

[5] T. Hayashi, “Application of MMCs to engine cylinder blocks and brake discs,” Honda R&D Co. Ltd., Tochigi, Japan

[6] A. Ashori, “Wood–plastic composites as promising green-composites for automotive industries!” *Bioresource Technology*, vol. 99, no. 11, pp. 4661-4667, 2008.

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New approach of designing braking system for electric vehicle using artificial intelligence

N. Tamaldin[1,2,*], M. Saifuddin[1], A.K. Mat Yamin[1,2], G. Omar[1,2]

1) Faculty of Mechanical Engineering, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia
2) Centre for Advanced Research on Energy, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia

*Corresponding e-mail: noreffendy@utem.edu.my

Keywords: Breaking system; artificial intelligence

ABSTRACT – Braking system is one of the most important parts in vehicle. Some improvement has been made to improve braking performance such as stability control, maneuverability, braking condition and many others. This paper reviewed about research in term of braking system that been used by vehicle at present. New approach has been proposed in this paper about braking system using artificial intelligence. The approach is using fuzzy logic controller and sensor to trigger the braking system can automatically when some obstacles approaches in certain distance. Expected result from this study is a review of technology in braking system which could break automatically when some criteria are fulfill.

1. INTRODUCTION

The study on new evolution of vehicle is continuous and still growing until today as part of the active and passive safety vehicle research. There are so many researchers investigating on upgrading the vehicle braking capabilities and recently involving electric vehicle. The electric vehicle is totally powered by electric produced by batteries inside the vehicle. Although the electric vehicle is a new modern vehicle but there has also some parts or sections that can be optimized such as the braking system. There are many braking system control utilized in electric vehicle. Past researchers conducted study on the braking system in term of its control braking system. The objective of the study is to review current method of designing vehicle braking system and propose a new approach of braking system using artificial intelligence integrated with sensor.

2. METHODOLOGY

In this review, various technique adopted by researchers in designing vehicle braking system especially for electric vehicle were compared. The proposed method is used artificial intelligence approach which use a sensor controller such as Adaptive Network based Fuzzy Inference System and sensor can detect distance. The proposed method is to ensure the safety of the vehicle while driver has lost focus while driving and the system make sure the vehicle can break automatically if sense something approach in certain distance.

Ithani et al. [1] discussed about comparison between two braking control methods integrating energy recovery for a two-wheel front driven electric vehicle. They have developed two types of braking control which is controlling the wheel slip ratio and based on ECE R13H constraints for M1 passenger vehicle. The study focuses on stability, potential energy, condition during braking, regulations, constraints, security and maneuverability of the vehicle. The tests are performed in extreme conditions and for different types of road surface. The result get from the paper is that the test is performed on a very good dry asphalt type road with an initial vehicle velocity is 80 km/h. Both controllers follow the specifications of the ECE R13H regulations stating that for an asphalt dry road surface, the stopping time should be lower than 3.82 s and the travelled distance lower than 50.6m. The results of the distance and stopping time is as per below Table 1.

Table 1 ECE R13 Regulation of distance and stopping time for extreme braking conditions on asphalt dry road [1].

Condition	**Stopping Time [s]**	**Traveled Distance [m]**
ECE R13 Constraints Based Controller	2.2389	25.1800
Sliding Mode Controller	1.9086	21.2071

One of the important parts in braking system is the material for the brake pad used. Paper presented by Barros et al. [2] discussed about morphological analysis of pad disc system during braking operations. The researchers have analyze pad disc system during braking operation where the researchers aims to present a method for quantifying the percentage of friction layer covered on the brake disc surface as well as to discuss the relationship between the parameter. The brake pad used for the experiment is non asbestos organic (NAO) and a semi-metallic (SM). The method used to examine the relationship of layer coverage to the coefficient of friction and the morphology of the contact plate existing on the brake pad surface. The results of the experiment as Table 2.

Table 2 Comparison results between NAO and SM [2].

Correlation	NAO	SM
Friction layer coverage vs Coefficient of friction	-0.57	0.76
Area fraction of contact plateaus vs Coefficient of friction	-0.88	0.14
Area fraction of contact plateaus vs Temperature variation	0.95	0.98

Other researchers also study about electric vehicle where their aims to develop a method that provides instantaneous speed information in the form of motor rotation. According to Pan et al. [3], there was lack of information of accurate motor rotational speed signal in regenerative braking system. The researchers used analysis of Hall position sensor digital characteristic of ideal and actual characteristic. They proposed three step of analysis which is firstly; it addressed principles of motor rotational speed measurement in the regenerative braking systems of the electric vehicle. Then, presents ideal and actual Hall position sensor signals conditioning and processing circuit and program for motor rotational speed measurement have been carried out based on measurement error analysis.

The performance of the braking system also needs to be studied to know the performance of the system in the vehicle. Jia et al. [4] discussed about the measurement for performance of automobile braking system with Electronic Vacuum Booster (EVB). The measurement of the performance braking system can be measured using Electronic Vacuum Booster (EVB). The performance parameter used in EVB that needed is relationship between the desired braking acceleration and the braking intension. The results show that the performance parameter of the different cars EVB is not the same so the performance parameters have to be measured respectively. Also, the speed has no influence on the relationship between the desired braking intension and the acceleration. The performance parameters of the both test cars EVB can be separated into two parts as the Eqn (1) and (2).

$$x = \begin{cases} 0, & -0.66m/s^2 < y < 0 \\ -5.5956y + 3.615, & -6.32m/s^2 < y \le -0.66m/s^2 \\ 40, & y \le -6.32m/s^2 \end{cases} \quad (1)$$

$$x = \begin{cases} 0, & -1.60m/s^2 < y < 0 \\ -5.0442y + 2.0526, & -8.70m/s^2 < y \le -1.60m/s^2 \\ 35, & y \le -8.70m/s^2 \end{cases} \quad (2)$$

Some of the braking system has locking system where during braking situation, the locked wheel leads the vehicle to skid and go out of drivers' control. Patra and Datta [5] have done a study about the friction so that the vehicle can be maneuvered by anti-lock braking controller in an emergency braking situation. They used one type of tire which is Dugoff's tire as the model of system dynamics because it has uniform distribution of pressure and has more realistic approach. The simulation is carried out for different road condition for a quarter models. The sliding mode observer is designed and implemented in the BS with non-liner sliding mode controller to deal with non-linearity and uncertainty. The experiment is done in the different variation of road which is dry asphalt, wet asphalt and snow asphalt and shows the good robustness.

3. CONCLUSION

Based on the research works presented by various researchers, braking system in electric vehicle is becoming an important technology in this modern era. It improves the performance of the car in term of stability, potential energy, condition during braking, security, maneuverability, breaking friction, speed information and performance parameter. The method discussed from these researches can be applied into the proposed system. Based on the proposed method, the system is expected can give positive results which is can improve the braking system in the future.

REFERENCES

[1] K. Itani, A. De Bernardinis, Z. Khatir, A. Jammal, "Comparison between two braking control methods integrating energy recovery for a two-wheel front driven electric vehicle," *Energy Conversion and Management*, vol. 122, pp. 330-343, 2016.

[2] L. Y. Barros, P. D. Neis, N. F. Ferreira, R. P. Pavlak, D. Masotti, L. T. Matozo, J. Sukumaran, P. De Baets, M. Ando, "Morphological analysis of pad-disc system during braking operations," *Wear,* vol. 352, pp. 112-121, 2016.

[3] C. Pan, L. Chen, L. Chen, H. Jiang, Z. Li, S. Wang, "Research on motor rotational speed measurement in regenerative braking system of electric vehicle," *Mechanical Systems and Signal Processing,* vol. 66, pp. 829-839, 2016.

[4] B. Jia, G. Lu, Y. Wang, G. Yu, "The measurement for performance parameter of automobile braking system with electronic vacuum booster," *Social and Behavioural Sciences,* vol. 96, pp. 2058-2065, 2013.

[5] N. Patra and K. Datta, "Observer based road tire friction estimation for slip control of braking system," *Procedia Engineering,* vol. 38, pp. 1566-1574, 2012.