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Journal of Automobile Engineering and Technology Research

Aims and Scope

The Journal of Automobile Engineering and Research is published three issues yearly by Enriched publications. Journal of Automobile Engineering and Research is peer reviewed journal and monitored by a team of reputed editorial board members. This journal consists of research articles, reviews, and case studies on Subjects. This journal mainly focuses on the latest and most common subjects of its domain.

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Contents

Experimental Study of Two Stroke Engine on Variation of Exhaust Pipe Diameter

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ABSTRACT

The paper deals with two stroke petrol engine. The exhaust pipe diameter is varied to study the performance of the two stroke petrol engine. In this paper the effect that the exhaust pipe geometry has on the characteristic parameters of the scavenging processes in small two stroke engines, by using the experimental method, which gives a reasonable degree of accuracy together with a certain simplicity and execution speed. Better the scavenging the thermal efficiency and brake mean effective pressure will be increase at high speed. The scavenging process depends on the location of transfer port and the engine geometry. The study of the scavenging is beyond the scope of this paper. The variation of exhaust pipe effect on the performance is studied in this paper.

Keywords- Two stroke scavanging, scavanging, exhaust pipe diameter.

Statement: Experimental study of variation of exhaust Pipe diameter.

Objectives:

1. To Study the variation of exhaust pipe on the two stroke petrol Engine.
2. To calculate the air consumption v/s Engine speed.
3. To Calculate the Specific fuel consumption v/s Engine speed.
4. To calculate the fuel consumption v/s Engine speed.

Introduction

The kinetic energy of the gas discharge is the main factor in the scavenging, When the exhaust port opens, there is a pressure release of gas into the ports and pipe. This imparts kinetic energy to the column in front of it. As the slug of pressure reaches the end of the pipe, the energy is transferred to the region of the exhaust port, causing a rarefaction which pulls out the residuals concurrently with arise in port pressure as the kinetic effect decreases. The pressure waves in the out flowing are formed by sound waves and move at the speed of sound. In order to start such a wave, however, a sudden impulse is necessary.

Gas speed: The conventional calculation for gas speed through the value and port is however useful in so far as it gives some indication. This takes piston speed as the primary factor, which is, of course, correct as far as the theoretical exhaust stroke, extending from BDC to TDC, and is concerned. It depends on the cross sectional area of the cylinder and the port.

Column Inertia: The gas Inertia is zero at the top and bottom dead center and it is maximum at about half stroke. At the peak Engine revolutions it is around 200 to 300 feet per sec. it is obvious that any gas speed based on piston speed must be zero at the overlap period. We can hope to preserve some kinetic effect, but a wave moving at 1400 feet per sec or more in the gas is certainly going to have a decisive affect on gas travelling at mean of 200 or so. If the wave is moving away from the port it will help the inertia effect, if it is moving away from the port it will easily overcome the inertia and recharge the cylinder with residual. In using long branch pipe we may have with advantage removed some of the aids to this recharging that exist in conventional manifolds. For the diameter of the pipe most commonly used, the maximum wave pressure is appox. Inversely proportional to the cross sectional area of the pipe. The raise of pressure tend to decrease at a higher engine speed, the exhaust pipe diameter would require to be larger than for one design to operate at more moderate speed. The exhaust pipe is used to evacuate the cylinder of the burnt charges and draw the fresh charge up to into the cylinder. This charge is being assisted by the ramming effect.

Organ Pipe Theory: at TDC of the overlap period. We know that the positive pressure pulse initiated by the valve opening towards BDC of the power stroke emerges from the open end and scatters to atmosphere. A reflected wave of rear faction, i.e. of opposite sign, instantaneously enters the open end and travels up to the valve, where it is again reflected but this time as of the same (negative sign) on again emerging at the open end, a positive travels up to the valve, i.e. open end reflection of the rear faction is accompanied by a flow gas down the pipe. If v= velocity of sound in the gas feet/sec, L is the length of the pipe from valve to outer end in inch, T= Time for pulse to travel from closed to open end and back in sec. In terms of crank angle degrees of the crank shaft rotation in degree. T=(RPM × L)/v. using the value of T equivalent to 120 from the time the exhaust valve opens sufficiently to start an effective wave, this may be assumed for calculation purposes to give favouriable conditions at the port. The speed of the sound is difficult to determine, as will be apparent from later observation. If as suggested, a speed of 1700 ft per sec. is taken as representative, the pipe length to give the above conditions for any fixed Engine speed may be obtained for example rpm for desired maximum torque = 4000, v= 1700 ft per sec, T= 120 crankshaft degrees. Pipe length L in inch = v × T/rpm = 1700 × 120/4000 = 51” = 4 ft 3 “ obeously with any increase or decrease in engine speed, T becomes shorter or longer respectively, and thus L must be shortened or lengthened in proportion. This is exemplified in the table below.

RPM for maximum torque	Pipe Length
2000	8 feet 6”
4000	4 feet 3”
6000	20 feet 10”
8000	2 feet 1.5”

Therefore no doubt that the single cylinder pipe calculated initially on the forgoing lines have shown good results and the formula serve as an admirable basis for starting experiments. There is no guarantee of what it will do at other speeds. For example, a good pipe length under the above conditions might lower the torque at another point in the power curve as to offset any advantages. Also, a fundamental flaw in the principal is seen by the fact that the rpm for maximum torque could be increased to a point at which there is no pipe left at all. A further discrepancy is added by the fact that when the valve is open the cylinder is effect include in the pipe length and, as the piston in moving, so that the resonant length of L is also varying. As regards the value to use T the only accurate method of obtaining this would be direct measurement of pressure.

Pressure and the pipe length: It will be noted that when using a fairly long pipe the maximum pressure at about BDC tend to raise at high speed. The critical length of the pipe, which gives the highest pressure of the first pulse, increases the length beyond this gives no further rise in pressure. The critical length decreases as the engine speed increases. Slow valve opening would call for a long pipe while for a very high speed engine the length would be under 2 feet.

For the diameters of pipe most commonly used, therefore, the maximum wave pressure is approx. inversely proportional to the cross sectional area of the pipe. The table below also gives practical guide to suitably pipe bores.

Area of Bore sq inch	Pressure Lb per sq inch
1.0	12.5
1.5	9
2.0	6.3
2.5	4.7

For 2 and 1.5 liters, 4 cylinder engine the individual cylinder capacity would be respectively 500 and 375 cc. the pipe bores corresponding to the cross sectional area required would also be determined to some extent by the engine port size.

Resonance: At least one designer in the past has attributed a useful torque increase through the theory that the pipe was pulsating in its own period and in time with the exhaust valve, thus producing a low pressure period at the port of overlap TDC. Obviously, this can happen, as seen from some of the curves. A more doubtful claim however, is that such effect will then occur at multiples of the engine speed, i.e. if there is a good boost in torque at 1000 rpm, there will also be one at 2000.

Pressure in Long Pipes: The Magnitude of the rise at higher speeds is such that it cannot be the result of a residual wave, as such a wave would have had its effect earlier and would in any case be small. In high speed engines the gas rushes into the pipe with a considerable velocity head. In doing so it has to accelerate the gas already in the pipe and must therefore translate part of its velocity head into a static head. Further, the supply of this high speed gas is limited, and soon tails off, this causes a decrease of pressure near the engine, and an increase 2 or 3 ft. along the pipe. A pressure wave is thus started back towards the engine.

Now the pipe is increased in diameter by about 50% at this point where the pressure increase takes place, the wave of increased pressured will almost disappear, the pressure drop is much more rapid, and this holds good even when the pipe length is increased. This confirms the above explanation, since such an increase in cross sectional area will reduce the forces necessary to accelerate the gas already in the pipe and eliminate the pressure build-up.

The process of investigating the increasing diameter or stepped pipe has another interesting development, in that the larger diameter length vibrate in its own period, it being possible to combine the wave to give a very low pressure drop after the initial positive pulse.

Exhaust Pipe System: when considering the exhaust system it is most important that the size, shape and length of the exhaust port taken into consideration as it is part of this system can affect results if ignored.

Plain Pipe: The plain pipe of constant diameter cutoff square to the desired diameter cutoff square to the desired length. Thus, length and diameter are the only variables.

Plain Pipe of constant diameter with megaphone of constant angle of taper attached to the end furthest from the exhaust port. Note that the length of plain Pipe could be zero in which case the megaphone would start at the exhaust port. This, in fact gives a system that the same as the first part of type 4 listed below. The variables of the system have increased to diameter of the Pipe, and angle of taper and length of megaphone. It should be noted that the smallest diameter of the megaphone is same as that of the Pipe, and that the larger diameter is dependent on the angle of the taper and the length of the megaphone.

Plain Pipe with an expansion box fitted to the end furthest from the exhaust port. The expansion box may be of any desired shape but usually comprises a shallow tapered divergent megaphone to the end of which is attached a short tapered convergent which reduces the exhaust system again. Attached to this may be a tail Pipe of constant diameter.

With this system the number of variables has further increased to diameter and length of the plain Pipe, angle of taper and length of first megaphone, length and smaller diameter of second megaphone and length and diameter of tail Pipe. It should be noted that normally the larger diameter of the megaphones are the same, and the smaller diameter of the second megaphone and that of the tail Pipe are the same. Usually this diameter is less than that of the exhaust Pipe. The working of the whole system is partially dependent on the total length of the Pipe as well as those of the constitant parts so that the system is much more difficult to experiment with than the first two described.

It can be further shown that a small diameter exhaust Pipe gives a better theoretical air consumption curve than a large one. However, this has to be balanced against the greatly increased frictional losses that arises with the smaller Pipe so that in this case the smaller Pipe has to be a compromise based on the exhaust port. The action of the reverse cone is provide a closed end Pipe effect of a gradual nature so that the initial negative pressure wave effect of gradual nature so that initial negative pressure wave is reflected as positive wave returns to the exhaust port. Thus the cylinder is scavenged by the negative pulse, which also initiates a ramming wave in the transfer passages, and the fresh charge is allowed to spill into the exhaust system but is forced back into the cylinder by the returning pressure wave. As the first wave cycle is the one greatest amplitude, it provides the maximum effect if used correctly as it contains the most energy, subsequent waves contain less energy to attenuation of the wave. Naturally the diameter and to some extent the length of the tail Pipe play a considerable part in this effect as the degree of restriction will affect the pressure building up in the chamber and this in turn affects the speed at which the positive wave returns to the exhaust port. The Engine speed at which maximum torque will be developed is dependent on the length of the tapered and on the length and diameter of the tailpipe. It is also affected by the volume of the chamber. A general guiding rule may be stated that as the Engine speed for maximum torque increases so the length and volume decrease, while the tailpipe length increases and its diameter reduced. As these the tailpipe length increase and its diameter reduces. As these point do interest this can be taken as a general guide. As the length of the tailpipe has an effect on the maximum RPM the Engine will reach, a compromise is often is made. This is done by using a large chamber and bore tailpipe to keep the RPM at while maximum torque occur as low as possible, but to use a long tailpipe to obtain at high maximum speed. The plain pipe is the simplest form of the exhaust system and once the diameter has been chosen. The suitable length may be taken by cutting down in the stages the pipe. A sleeve may be made to slide along the pipe to vary the length, this allows the approx. length to be determined.

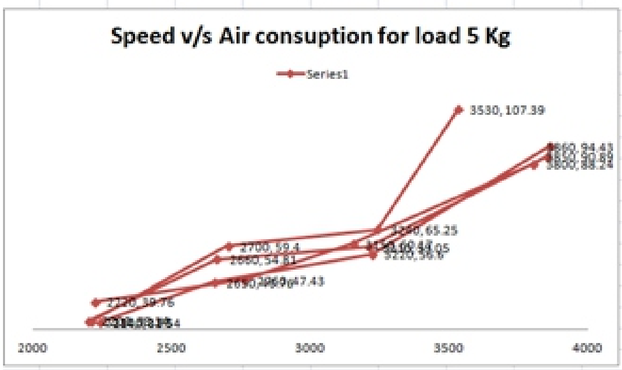
Engine Specification: Bajaj Auto, Bore=57 mm, stroke=57mm, displacement= 145.45 cc, CR=7.4:1, scavenging type= Tangential flow, inlet type= piston controlled port, exhaust open= 53.48 b: BDC. Dynamometer= Rope Brake type, Diameter of the drum= 275 mm, Dia. of Rope= 10 mm, Effective dia= 285 mm; Air Intake measure, Diameter of orifice= 20 mm, coefficient of discharge= 0.6 Manometric fluid= water, fuel measurement= Vol flow.

Other equipments: Four Pipes of dia= 38.5 mm, 43.7 mm, 55 mm, 62 mm, extension chamber for exhaust gases, silencer, Techometer, Stopwatch etc.

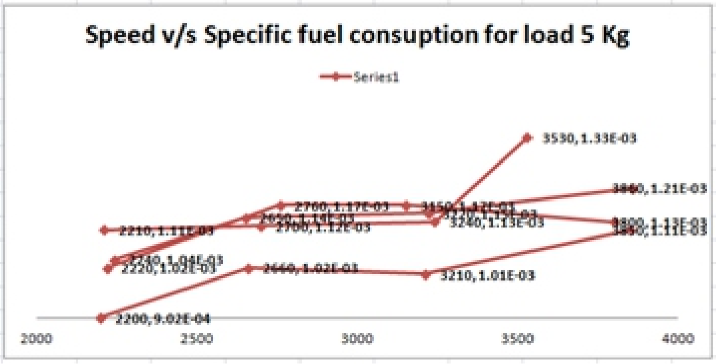
Data Analysis: The data's are as below. The Table(1) for 5 Kg loading.

	A	B	C	D	E	F
1	Load = 5 Kg					
2	No	Speed (RPM)	Fuel Con	Power	SFC	Air Con (cc/cy)
3	Diameter = 38.5 mm					
4	1	2240	0.336	321.16	1.04E-03	32.54
5	2	2650	0.435	380	1.14E-03	46.76
6	3	3220	0.528	461.67	1.15E-03	56.6
7	4	3860	0.672	553.43	1.21E-03	94.43
8	Diameter = 43.7 mm					
9	1	2200	0.284	315.42	9.02E-04	33.14
10	2	2660	0.389	381.38	1.02E-03	54.81
11	3	3210	0.4625	460.23	1.01E-03	59.05
12	4	3850	0.6166	552	1.11E-03	90.89
13						
14	Diameter = 55 mm					
15	1	2220	0.321	315.42	1.02E-03	39.76
16	2	2760	0.435	395.71	1.17E-03	47.43
17	3	3150	0.528	451.63	1.17E-03	60.17
18	4	3800	0.6166	545	1.13E-03	88.24
19	Diameter = 62 mm					
20	1	2210	0.352	316.83	1.11E-03	32.98
21	2	2700	0.435	387.11	1.12E-03	59.4
22	3	3240	0.528	464.53	1.13E-03	65.25
23	4	3530	0.672	506.11	1.33E-03	107.39

Graph (1)



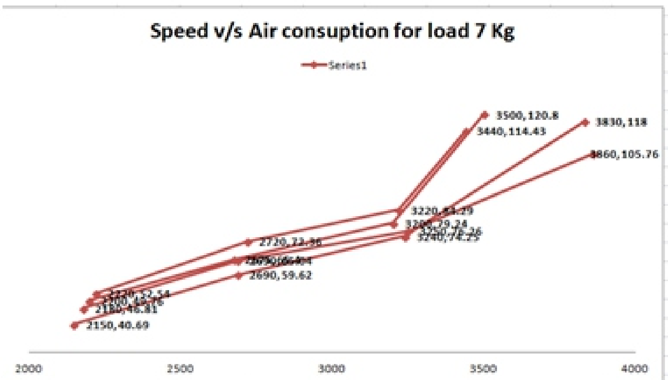
Graph (2)



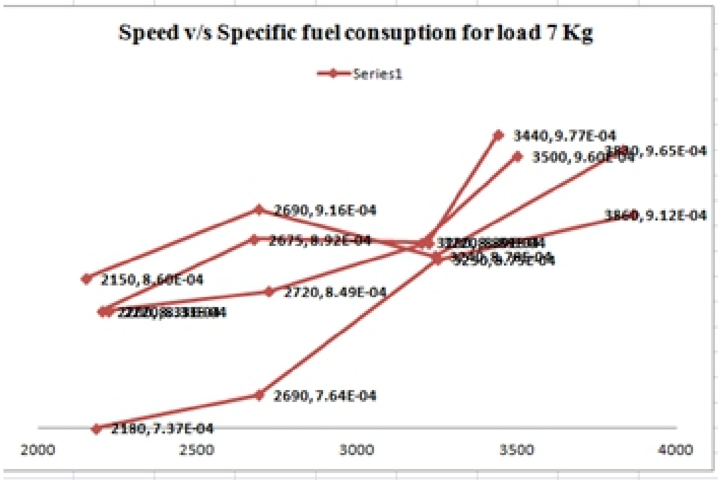
Table(2)

	A	B	C	D	E	F
1	Load = 7 Kg					
2	No	Speed (RI	Fuel Con (	Power	SFC	Air Con (cc/cy)
3	Diameter = 38.5 mm					
4	1	2150	0.37	430.13	8.60E-04	40.69
5	2	2690	0.493	438.16	9.16E-04	59.62
6	3	3240	0.569	648.19	8.78E-04	74.25
7	4	3830	0.74	766.22	9.65E-04	118
8	Diameter = 43.7 mm					
9	1	2180	0.321	436.13	7.37E-04	46.81
10	2	2690	0.411	538.16	7.64E-04	65.04
11	3	3250	0.569	650.19	8.75E-04	76.26
12	4	3860	0.704	772.23	9.12E-04	105.76
13						
14	Diameter = 55 mm					
15	1	2200	0.37	444.13	8.33E-04	49.76
16	2	2675	0.477	535.16	8.92E-04	65.4
17	3	3200	0.569	640.19	8.89E-04	79.24
18	4	3500	0.672	700.21	9.60E-04	120.8
19	Diameter = 62 mm					
20	1	2220	0.37	444.13	8.33E-04	52.54
21	2	2720	0.462	544.16	8.49E-04	72.36
22	3	3220	0.569	640.19	8.89E-04	84.29
23	4	3440	0.672	688.2	9.77E-04	114.43

Graph (3)



Graph (4)



CONCLUSION

Table(1) show the data for 5 Kg load, Table (2) show the data for 7 Kg load. The graph(1) is drawn speed v/s air consumption for 5 kg load, The graph(2) is drawn speed v/s Specific fuel consumption for 5 kg load.

The table(2) show the data for 7 kg load, the graph (3) for speed v/s air consumption for load 7 kg, the graph (4) show speed v/s Specific fuel consumption for load 7 kg.

The graphs show variation of specific fuel consumption with engine speed and variation of air consumption with speed for different exhaust pipe diameter. The specific fuel consumption of the engine reduces at first as the diameter of the exhaust pipe increased and then as the diameter is further increased the SFC raises up again. As the diameter of the exhaust pipe diameter reduces the friction in the exhaust pipe which helps better removal of exhaust gasses. But the increase in diameter of the exhaust pipe affects the scavenging because of higher pressure at the exhaust port during the exhaust which is due to lower kinetic energy of the gases in the exhaust pipe. Maximum possible speed of the engine reduces with increases in diameter of the pipe.

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Vehicle Ride Control Using Human Body Dynamic Simulation

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A B S T R A C T

The aim of this study was Vehicle ride control using human body dynamic simulation. When designing suspension system in ordinary sedans, we cannot only consider comfort and stability criteria. In these occasions, according to the working class of the vehicle, criteria of comfort and stability together and with a certain proportion will intervene in the designing suspension system. Principally, in such occasions, we lose the benefits of mentioned criteria due to relative consideration of the other criterion. What has been investigated in this research is the design of a controller that decreases acceleration caused by vibrations transmitted to the human body and also lowers the displacement of the tire with respect to the road. The proposed model for the body is a model with four degrees of freedom and the proposed model for the vehicle is a model with seven degrees of freedom. The proposed controller is an optimal-fuzzy controller that optimal controller and fuzzy one have been used for the design of state feedback and the change in the weighting functions of optimal controller, respectively. To confirm the strength of the designed controller, the various inputs of the actual road with varying vehicle speed have been applied to the model. The obtained results show that this controller has a major influence on deceleration transferred to the body and also in critical situations results in reduction in the displacement of the tire with respective to the road input.

Keywords: Semi-active suspension, State feedback, Fuzzy logic, Simulink, Bounce acceleration, Biodynamic models of human

Introduction

Work on the design of active suspension systems has been started at the late 1960s. One of the first things performed in this area has been design of anti-roll systems by Meiry and Roseler in 1968 in MIT University [1]. Two issues of ride comfort and lateral stability must be considered in designing inactive suspension system. But the reality is that these two are in conflict with each other in such a way that for improvement of ride comfort the suspension system component must be selected soft but for improvement of the lateral stability the suspension system components must be selected hard. Therefore, in most cases, in the design of inactive suspension system the attempt is that a compromise point is achieved. William in 1997 investigated vehicle behavior in inactive state using 4/1 model. He obtained the vibratory response of the model using the conversion function of 1/4 model by applying road inputs randomly and then compared vibratory behavior of inactive system in various states by changing shock absorber damping coefficient and spring stiffness factor. William in another study in 1997 investigated various types of suspension systems from aspect of separating low and high frequencies exerted by the road and from an industrial aspect he designed an operator for active suspension system using gas or oil springs. In this case, the applied control method has been optimal [2]. Purdy& Bulman in 1997 have exerted 1/4 model for designing active suspension system on the racing vehicles. The presented work has been investigated from both theoretical and practical aspects. The open-circuit response control test has been performed to verify the model. In this research, we tried to work on the suspension system so that it answers properly in both low and high frequencies of the road [3]. It is worthy to note that the dynamics of an vehicle is a rapid one and it is necessary to use methods and control algorithms that can control the system at the least possible time. We can name some various methods of controlling the active suspension systems used in most of the papers in order of time progress that are as follows:

1) Classic controllers
2) Optimal control methods
3) Advanced controllers such as fuzzy and neural networks

Rao and Prahlad in 1995 have used linear 1/4 model for designing active suspension system of the vehicle. They used sky-hook damping model in their study. Fuzzy controller has been used in this work in which location change of suspension system (the difference between the location of center of sprung and unsprung mass) and sprung mass velocity have been considered as fuzzy controller inputs and the output of fuzzy system is changes of the control force. In this paper, bell membership functions for linguistic variables of suspension system displacement and sprung mass velocity and change of control force has been used. The road profile used in this paper has been pseudo-random one with natural frequencies of 1 and 2 Hz. The results of this paper are so that the acceleration of the body in the active suspension system model has been less than that of the reference model. Also, to check the strength of the controller with increased sprung mass by 30% and also decreased stiffness of the suspension system by 30%, the results have not changed. This issue shows capability of fuzzy systems to face with many changes [4]. Yoshimura et al. in 1999 used nonlinear 1.2 model for designing active suspension system. In this model, spring stiffness coefficients and shock absorber damping factors are nonlinear.

The purpose of this paper has been to decrease bounce acceleration and acceleration of pitch acceleration. Because of being overwhelming to design a type of controller for such a nonlinear system, so we have used a combination of a linear controller and fuzzy one as a complementary controller. Control forces are created by means of acting pistons in a hydraulic cylinder. The linear part of this controller is for controlling bounce acceleration that bounce acceleration feedback has been used in front and back suspension location. The road input has been randomly considered in this work [5]. Considering MacPherson suspension system for designing active suspension system Amato and Viassolo in 2000 have applied 1/4 model. The model and the operator used in this paper were linear and nonlinear, respectively. The purpose of doing this work is to decrease bounce acceleration for ride comfort and preventing the collision of suspension system components during control and maintaining life time of suspension system components. The proposed controller has been composed of two rings which inner ring controls hydraulic operator for tracing optional acting force and outer ring is a fuzzy controller that makes the controller update for creating optional operating force [6]. Tzuu-hseng et al. in 2000 use 1/4 model for designing an active suspension system. They used a combination of two fuzzy controllers in their paper. This controlling design includes of a fuzzy feedback controller and a fuzzy feed forward one. Fuzzy feedback controller has been used for a compromise between ride comfort and tire displacement with respect to the body based on speed variations of sprungand unsprung masses and fuzzy feed forward controller has been applied to eliminate the road disturbances. The purpose of proposed controller is to create ride comfort and stability for the vehicle on the rough roads [7]. Following Yoshimura, Fan and Jun in 2001 applied nonlinear 1/2 model for designing active suspension system. Their purpose was to decrease bounce acceleration and pitch acceleration. In this paper, a combination of linear controller and fuzzy one have been used a complementary controller. But the difference between this work and Yoshimura's work is that a PID controller has been used for controlling bounce acceleration instead of using bounce acceleration feedback. The obtained results are resulted from control system improvement with respect to Yoshimura's work [8]. Daniel Fischer and his colleague in 2004 used Mechatronics methods to estimate and control active and semi-active suspension systems and tested the results on experimental models and implemented them on the real vehicle [9]. Els et al. in 2007 investigated compromise determination method between ride comfort and stability on the military vehicles. In this paper three types of military vehicles have been investigated and good results have been obtained for practical designing on suspension system of military vehicles [10]. Guclu and Gulezb in 2009 used one degree of freedom body model on the seven degrees of freedom vehicle model and designed an operator under driver's seat using neural network that greatly decreases bounce acceleration entered to the body [11]. Solomon in 2011 designed a gas damper with variable coefficient on the suspension system of real freight vehicles. He showed in this paper that bounce acceleration can be decreased while stability criteria do not achieve critical situations [12].

As we can see in previously performed works, in most modeling, a vehicle model and in some cases with simple body model has been used to design suspension system. In papers in which body model has been used on the vehicle model, an operator under driver's seat has been embedded.

In direction of doing this research, a combination of seven degrees of freedom vehicle model and four degrees of freedom body model (that is one of the most presented models for the body) have been applied for designing semi-active suspension system , their results have been compared with vehicle model without considering human body model. Also, the operator under driver's seat has been omitted in previously performed papers and the exerted force to the passenger has been provided using available operator in suspension system. The final purpose of this research is to control bounce acceleration of the human body and tire displacement with respect to the road but as we know two issues of ride comfort and tire displacement with respect to the road are in conflict with each other. In this study, depending on vehicle performance situations, we aim to create the optional force in the suspension system. For this purpose, an optimal controller and a fuzzy one have been used to improve ride comfort and tire displacement with respect to the road and to create compromise between these two issues depending on vehicle performance situation, respectively. To control the acceleration of human body and tire displacement with respect to the road, a state feedback designed based on one of the optimal control methods has been applied. Weighting functions used in optimal controller are constant numbers that by selecting them we can improve one of these two issues. Applying a fuzzy controller we can improve the acceleration of the human body or tire displacement with respect to the road in proportion with various vehicle performance situations by changing in weighting functions used in optimal controller. To explain clearly designing optimal and fuzzy controllers, at first, 1/4 vehicle model with one degree of freedom human body have been used and then complete vehicle model and human body were used. The inputs of fuzzy controller are vehicle speed and root mean squares (RMSs) of the acceleration of vehicle body and its output is weighting functions used in state feedback. The road profile used in this thesis is a sinusoidal one, lump, and type B, C, and D real road. In designing semi-active suspension system, ride comfort and stability have a great importance. In this project, the purpose of control is to minimize bounce acceleration of vehicle and human body (from aspect of comfort) and also to minimize tire displacement with respect to the road input (from aspect of stability). This project includes of a state feedback that LQR method has been used to design it. Because of existence of uncertainty in vehicle system and problems related to modeling, the proposed controller in this project is an optimal-fuzzy controller. Firstly, we explain state feedback that is calculated by LQR method. For simplification of explanation of this method, 1/4 vehicle model and also one degree of freedom body model is used and in conclusion section, seven degrees of freedom and four degrees of freedom body model are applied.

Designing state feedback using LQR method [2]:

A 1/4 model with semi-active suspension system with a focused mass representing one degree of freedom body model is shown in figure 1. In this model, a variable damper has been used as the acting force (Fa) that according to various conditions such as type of the road provides necessary force between sprung and unsprung mass based on a control rule. Block diagram of designed controller is shown in figure 1.

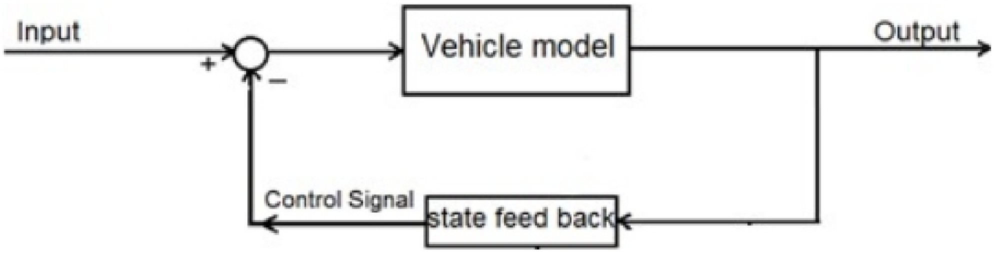

Figure 1-Block diagram of state feedback

The frequency response for different amounts of weighting functions:

A) The state that weighting functions related to the vehicle is against zero and weighting functions related to human body is equal to zero:

$$\rho_1 = 0.1 \; , \; \rho_2 = 0.1 \; , \; \rho_3 = 10 \; , \; \rho_4 = 100 \; , \; \rho_5 = 0 \; , \; \rho_6 = 0$$

B) The state that weighting functions related to the vehicle is similar to state A and weighting functions related to human body is not equal to zero:

$$\rho_1 = 0.1 \; , \; \rho_2 = 0.1 \; , \; \rho_3 = 10 \; , \; \rho_4 = 100 \; , \; \rho_5 = 10 \; , \; \rho_6 = 1000$$

Frequency response of the tire displacement with respect to the road input is given in figure 2.

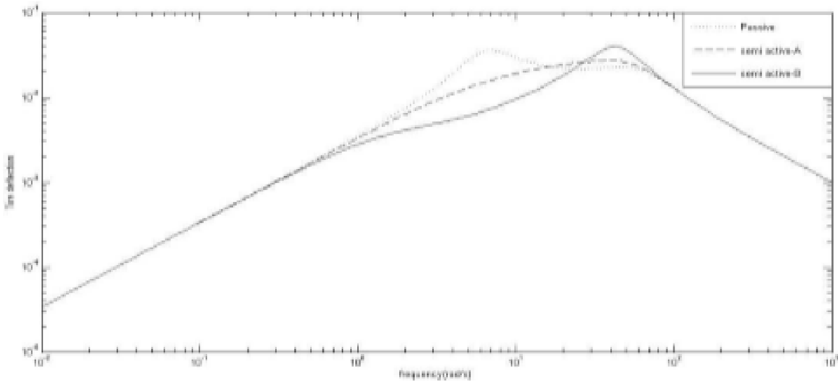

Figure 2-Frequency response of the tire displacement with respect to the road input

Frequency response of the acceleration of vehicle body is presented in figure 3.

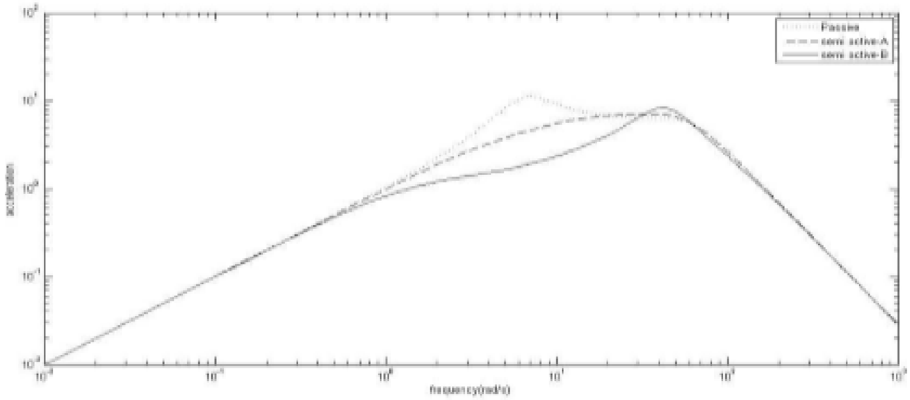

Figure 3-Frequency response of the acceleration of vehicle body

Frequency response of the acceleration of human body is shown in figure 4.

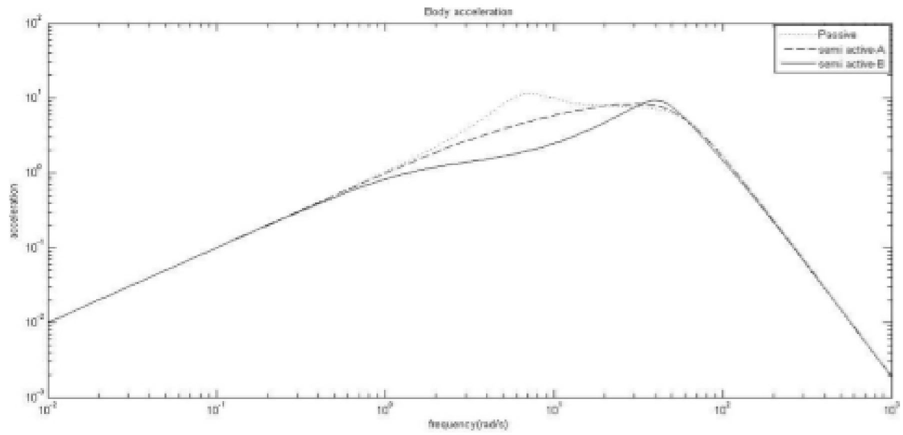

Figure 4-Frequency response of the acceleration of human body

It is must be considered that the behavior of the acceleration of human body and vehicle body are very close to each other at different frequencies. As can be seen, we can decrease the accelerations of the human body and vehicle body at frequencies 1 to 5 Hz using body model and applying weighting functions for it and also we can improve stability but at frequency of 10 Hz, the acceleration and stability become undesirable. To tailor the acceleration and stability at frequency of 10 Hz, we must increase weighting functions related to the tire.

C) We increase weighting functions related to the tire displacement with respect to the road.

$$\rho_1 = 1000000\ ,\ \ \rho_2 = 0.1\ ,\ \ \rho_3 = 10\ ,\ \ \rho_4 = 100\ ,\ \ \rho_5 = 10\ ,\ \ \rho_6 = 1000$$

Frequency response of the tire displacement with respect to the road input is depicted in figure 5.

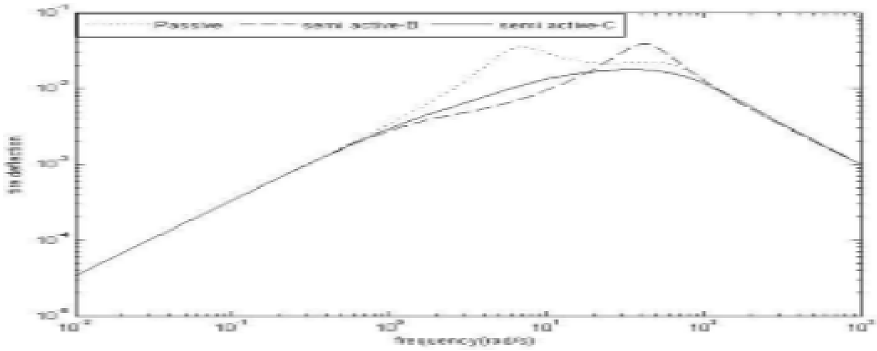

Figure 5-Frequency response of the tire displacement with respect to the road input

Frequency response of the acceleration of vehicle body is presented in figure 6.

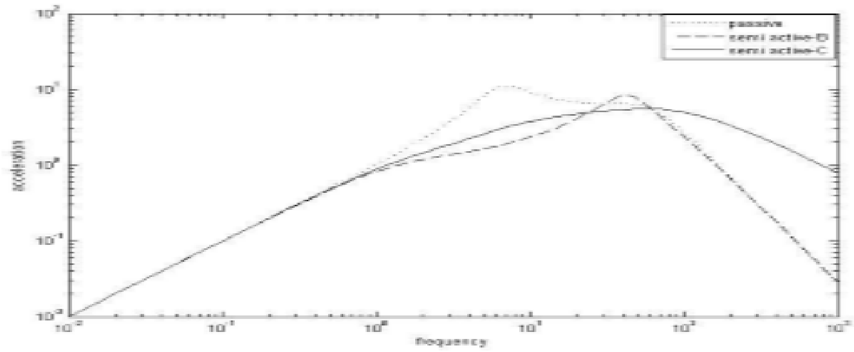

Figure 6-Frequency response of the acceleration of vehicle body

Frequency response of the acceleration of human body is shown in figure 7.

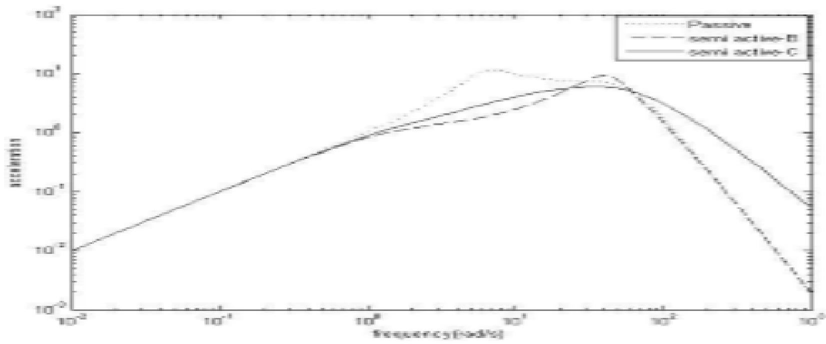

Figure 7-Frequency response of the acceleration of human body

As can be seen in the curves, with enhancement of weighting functions related to the tire, stability and acceleration are improved in frequency of 10 Hz but acceleration has been undesirable at frequencies higher than 10 Hz.

Response to sinusoidal input:

To mentioned contents be clearer, we use a sinusoidal input with frequencies of 1 and 10 Hz.

Sinusoidal input with a frequency of 1 Hz $(z_r = 0.0125\sin(2\pi \times t))$**:**

Tire displacement with respect to the road input, the acceleration of vehicle body and the acceleration of human body are given in figures 8, 9, and 10.

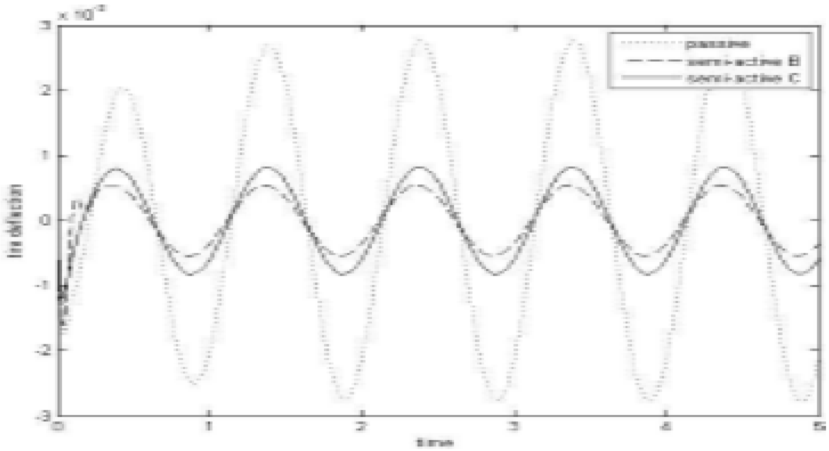

Figure 8- The tire displacement with respect to the road input

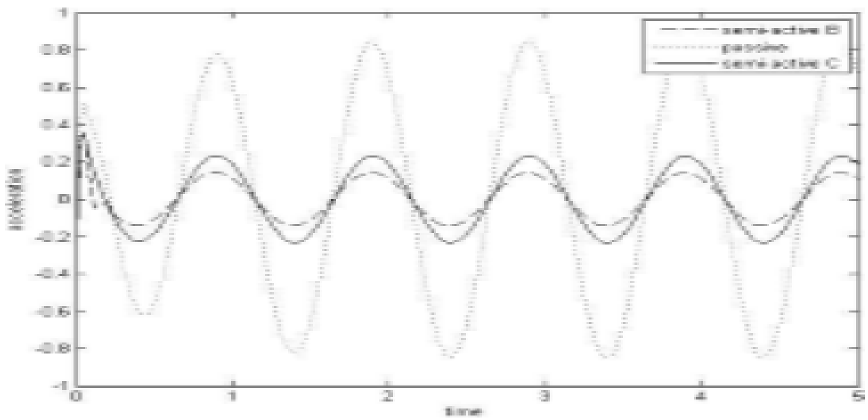

Figure 9- The acceleration of vehicle body

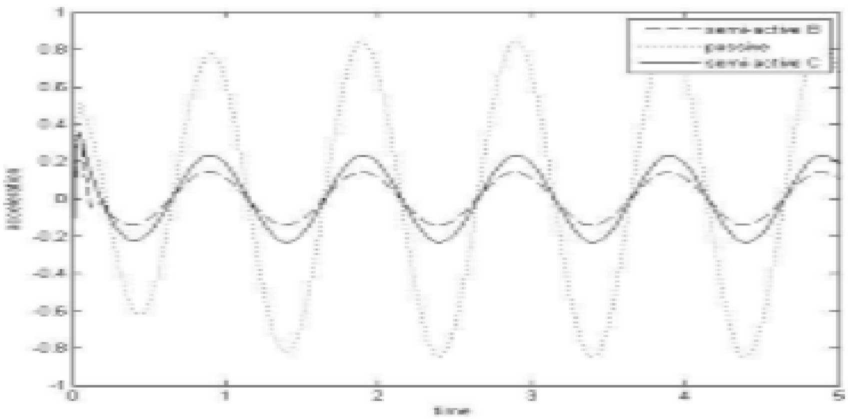

Figure 10- The acceleration of human body

Sinusoidal input with a frequency of 10 Hz $(z_r = 0.0125 \sin(20\pi \times t))$**:**

Tire displacement with respect to the road input, the acceleration of vehicle body and the acceleration of human body are given in figures 11, 12, and 13.

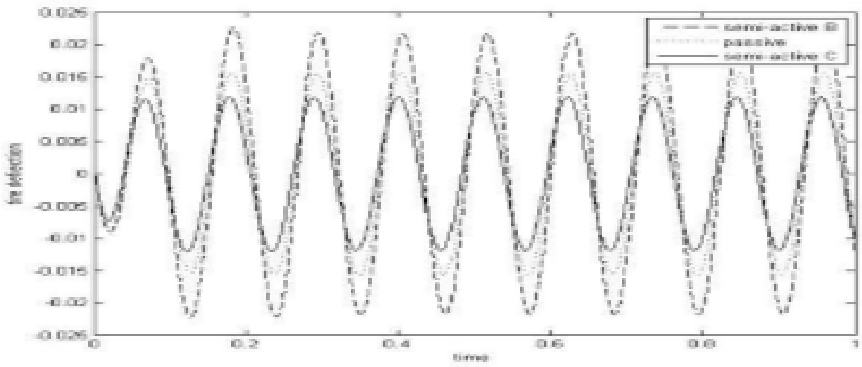

Figure 11- The tire displacement with respect to the road input

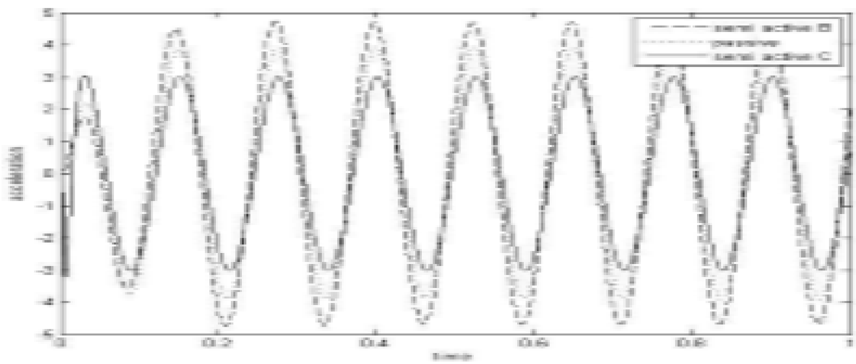

Figure 12- The acceleration of vehicle body

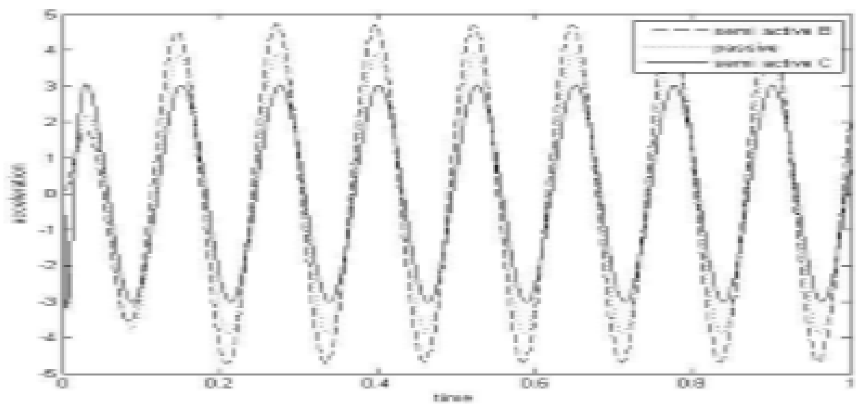

Figure 13- The acceleration of human body

As can be seen in the curves, with enhancement of weighting functions related to the tire, stability and acceleration are improved in frequency range of 10 Hz but acceleration has been undesirable at frequencies higher than 10 Hz.

Response to real road:

We know that real road input is random and without a constant frequency. Therefore, we in this section apply type B road like figure 14 as the input for the model and show its curves. It is necessary to note that vehicle speed has been considered 20 m/s.

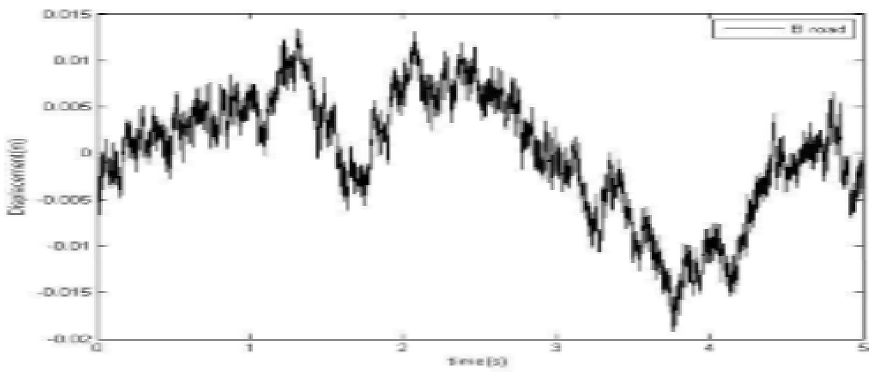

Figure 14-Type B road

Firstly, we apply type B road input to the semi-active suspension system with many weighting functions for acceleration and as we will observe acceleration of semi-active suspension system is better than that of the inactive suspension system but the tire displacement with respect to the road becomes undesirable.

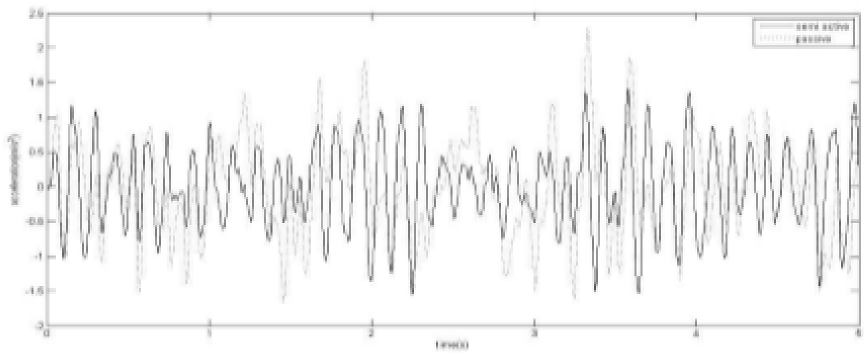

Figure 15- The acceleration of human body

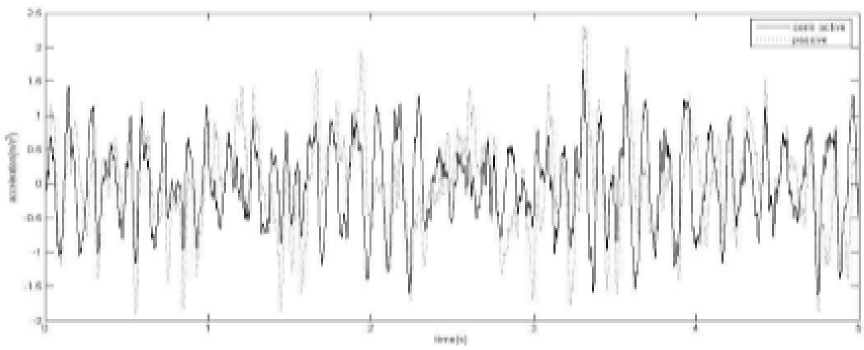

Figure 16- The acceleration of vehicle body

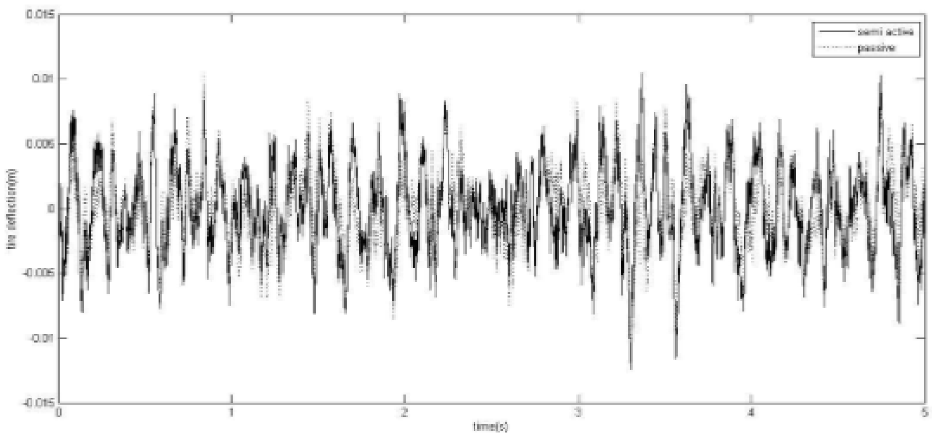

Figure 17- The tire displacement with respect to the road input

Now we apply type B road input to the semi-active suspension system with many weighting functions for tire displacement and as we will see tire displacement with respect to the road finds a desirable situation in semi-active suspension system but the acceleration becomes undesirable.

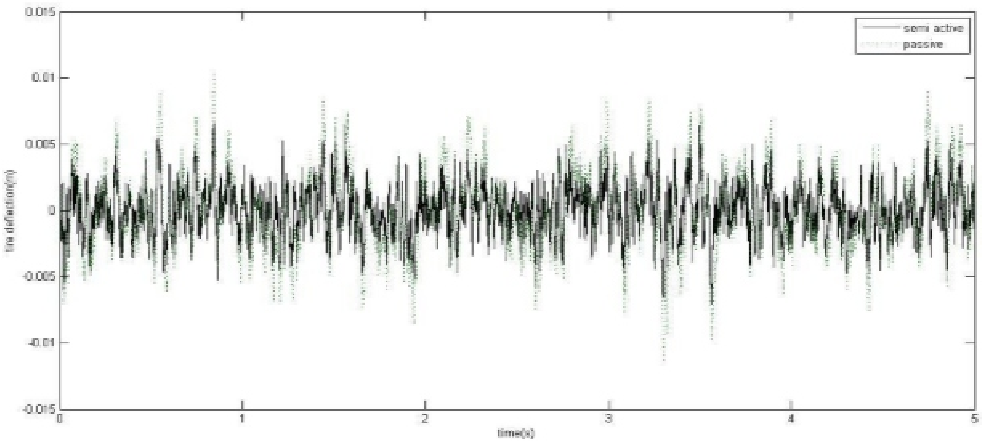

Figure 18- The tire displacement with respect to the road input

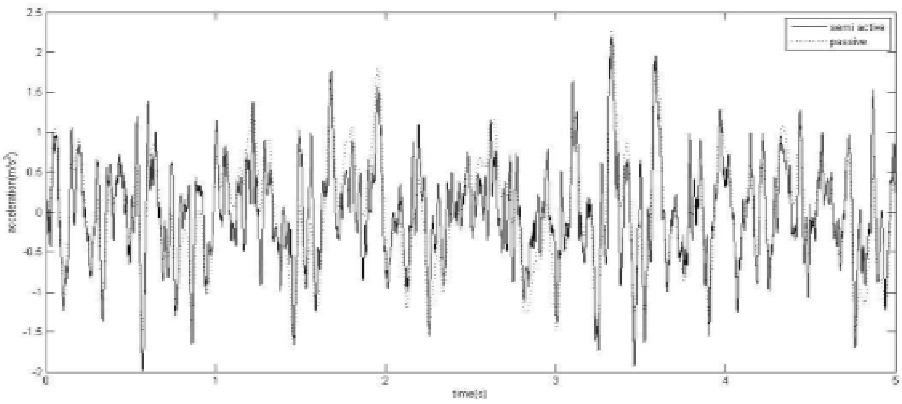

Figure 19- The acceleration of human head

As it is clear from the curves, due to conflict in two important issues of ride comfort and stability, we can only improve one of them; however, with improving one of them, the other one will find an undesirable situation. So, the best solution is to improve ride comfort or tire displacement with respect to the road depending on the condition of vehicle performance. This has been performed by changing optimal controller weighting functions using a fuzzy controller in this thesis. Fuzzy controller input is the summation of root mean squares (RMSs) of vehicle body acceleration and vehicle speed and the output of fuzzy controller of weighting functions used in the controller is optimal.

Seven degrees of freedom vehicle model and body model:

In this section, we give figure 20, equations and definition of state variables for complete model.

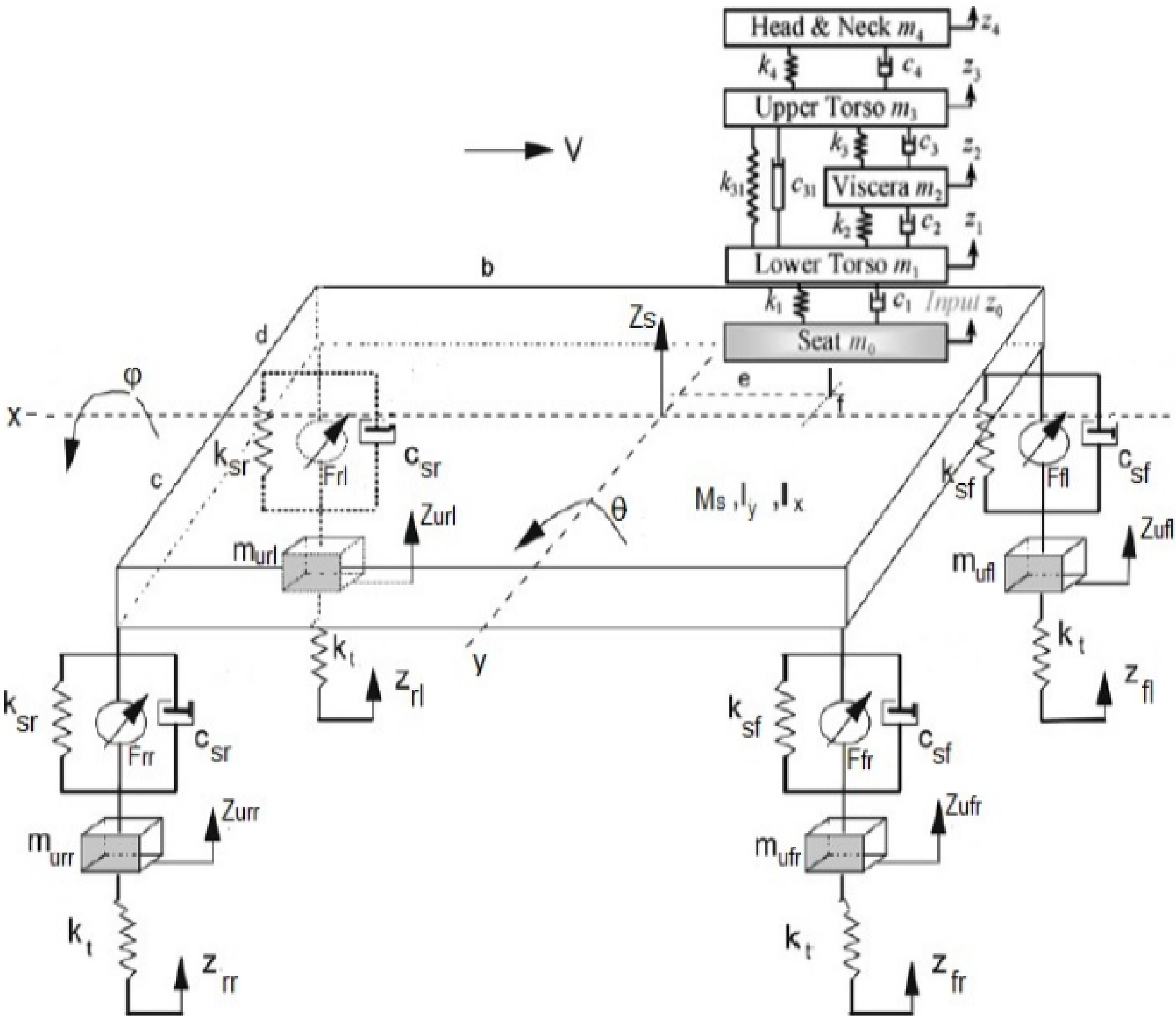

Figure 20- Seven degrees of freedom model and body model

$$m_1\ddot{z}_1 = -(k_1+k_2+k_{31})z_1 - (c_1+c_2+c_{31})\dot{z}_1 + k_2z_2 + c_2\dot{z}_2 + k_{31}z_3 + c_{31}\dot{z}_3 + k_1z_s + c_1\dot{z}_s + ek_1\theta + ec_1\dot{\theta} + fk_1\varphi + fc_1\dot{\varphi}$$

$$m_2\ddot{z}_2 = k_2z_1 + c_2\dot{z}_1 - (k_2+k_3)z_2 - (c_2+c_3)\dot{z}_2 + k_3z_3 + c_3\dot{z}_3$$

$$m_3\ddot{z}_3 = k_{31}z_1 + c_{31}\dot{z}_1 + k_3z_2 + c_3\dot{z}_2 - (k_4+k_3+k_{31})z_3 - (c_4+c_3+c_{31})\dot{z}_3 + k_4z_4 + c_4\dot{z}_4$$

$$m_4\ddot{z}_4 = k_4z_3 + c_4\dot{z}_3 - k_4z_4 - c_4\dot{z}_4$$

$$m_s\ddot{z}_s = k_1z_1 + c_1\dot{z}_1 - (2k_{sf}+2k_{sr}+k_1)z_s - (2c_{sf}+2c_{sr}+c_1)\dot{z}_s + k_{sf}z_{ufl} + c_{sf}\dot{z}_{ufl} + k_{sf}z_{ufr} + c_{sf}\dot{z}_{ufr} + k_{sr}z_{url} + c_{sr}\dot{z}_{url} + k_{sr}z_{urr} + c_{sr}\dot{z}_{urr} - (2ak_{sf}+2bk_{sr}+ek_1)\theta - (2ac_{sf}+2bc_{sr}+ec_1)\dot{\theta} - fk_1\varphi - fc_1\dot{\varphi} + F_{fr} + F_{fl} + F_{rl} + F_{rr}$$

$$I_y\ddot{\theta} = ek_1z_1 + ec_1\dot{z}_1 - (2ak_{sf}-2bk_{sr}+ek_1)z_s - (2ac_{sf}-2bc_{sr}+ec_1)\dot{z}_s + ak_{sf}z_{ufl} + ac_{sf}\dot{z}_{ufl} + ak_{sf}z_{ufr} + ac_{sf}\dot{z}_{ufr} - bk_{sr}z_{url} - bc_{sr}\dot{z}_{url} - bk_{sr}z_{urr} - bc_{sr}\dot{z}_{urr} - (e^2k_1+2a^2k_{sf}-2b^2k_{sr})\theta - (e^2c_1+2a^2c_{sf}-2b^2c_{sr})\dot{\theta} - efk_1\varphi - efc_1\dot{\varphi} + aF_{fr} + aF_{fl} - bF_{rl} - bF_{rr}$$

$$I_x\ddot{\varphi} = fk_1z_1 + fc_1\dot{z}_1 - (fk_1)z_s - (fc_1)\dot{z}_s + ck_{sf}z_{ufl} + cc_{sf}\dot{z}_{ufl} - dk_{sf}z_{ufr} - dc_{sf}\dot{z}_{ufr} + ck_{sr}z_{url} + cc_{sr}\dot{z}_{url} - dk_{sr}z_{urr} - dc_{sr}\dot{z}_{urr} - fek_1\theta - fec_1\dot{\theta} - (f^2k_1+2c^2k_{sf}+2c^2k_{sr})\varphi - (f^2c_1+2c^2c_{sf}-2c^2c_{sr})\dot{\varphi} - dF_{rr} + cF_{fl} + cF_{rl} - dF_{fr}$$

$$m_{ufl}\ddot{z}_{ufl} = k_{sf}z_s + c_{sf}\dot{z}_s - (k_{sf}+k_t)z_{ufl} - c_{sf}\dot{z}_{ufl} + ak_{sf}\theta + ac_{sf}\dot{\theta} + ck_{sf}\varphi + cc_{sf}\dot{\varphi} + k_tz_{fl} - F_{fl}$$

$$m_{url}\ddot{z}_{url} = k_{sr}z_s + c_{sr}\dot{z}_s - (k_{sr}+k_t)z_{url} - c_{sr}\dot{z}_{url} - bk_{sr}\theta - bc_{sr}\dot{\theta} + ck_{sr}\varphi + cc_{sr}\dot{\varphi} + k_tz_{rl} - F_{rl}$$

$$m_{urr}\ddot{z}_{urr} = k_{sr}z_s + c_{sr}\dot{z}_s - (k_{sr}+k_t)z_{urr} - c_{sr}\dot{z}_{urr} - bk_{sr}\theta - bc_{sr}\dot{\theta} - dk_{sr}\varphi - dc_{sr}\dot{\varphi} + k_tz_{rr} - F_{rr}$$

We define state variables as following:

State variables related to vertical motion of the body:

$x_1 = z_1$ و $x_2 = \dot{z}_1$ و $x_4 = \dot{z}_2$ و $x_5 = z_3$ و $x_6 = \dot{z}_3$ و $x_7 = z_4$ و $x_8 = \dot{z}_4$

State variables related to vehicle body (sprung mass):

$x_9 = z_s$ و $x_{10} = \dot{z}_s$ و $x_{11} = \theta$ و $x_{12} = \theta$ و $x_{13} = \varphi$ و $x_{14} = \dot{\varphi}$

State variables related to tires (unsprung mass):

$x_{15} = z_{ufl} - z_{fl}$ و $x_{16} = \dot{z}_{ufl}$ و $x_{17} = z_{ufr} - z_{fr}$ و $x_{18} = \dot{z}_{ufr}$

$x_{19} = z_{url} - z_{rl}$ و $x_{20} = \dot{z}_{url}$ و $x_{21} = z_{urr} - z_{rr}$ و $x_{22} = \dot{z}_{urr}$

We note that the number of state variables is twice the degree of freedom. Similar to 1/4 model and simple body model, we can obtain matrices of state space through defining state variables and by placing state matrices in MATLAB software, we can control seven degrees of freedom model and four degrees of freedom body model similar to 1/4 model and simple body model.

DISCUSSION

The aim of designing semi-active suspension system is to decrease vibrations transferred to human body due to the roughness of the road surface and also stability of vehicle that are appeared by keeping contact of the tire with the road. To show the efficiency of the designed controller, we have applied various types of sinusoidal profiles, lump, and real roads that are considered as inputs for the vehicle to the model in this thesis. The results of applying these inputs are given in following. Then, the response of model with seven degrees of freedom and also body model with four degrees of freedom with respect to various types of inputs will be investigated.

Response to lump input:

The considered lump input is like figure 21.

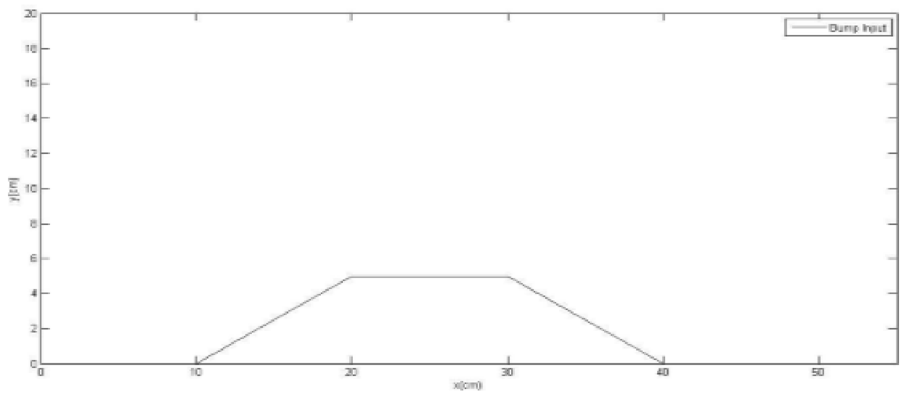

Figure 21-Lump input

Assuming that the vehicle is moving at a speed of 10 m/s, we have presented the tire displacement with respect to the road, the vehicle body acceleration and the human head acceleration in figures 22, 23, and 24.

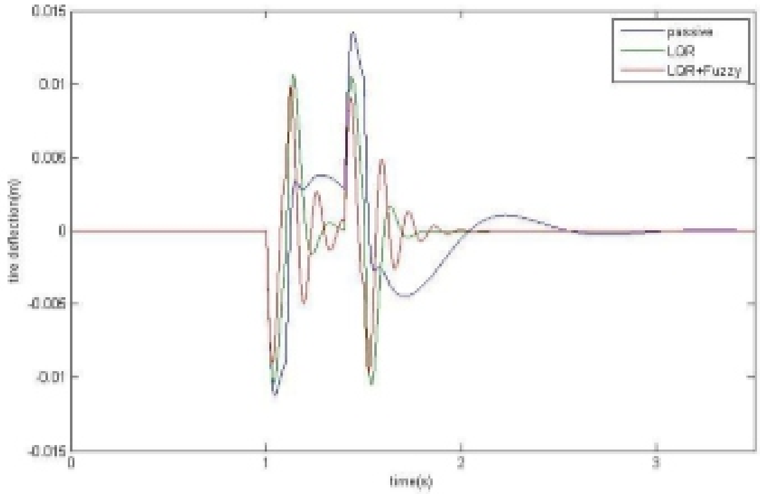

Figure 22- The tire displacement with respect to the road

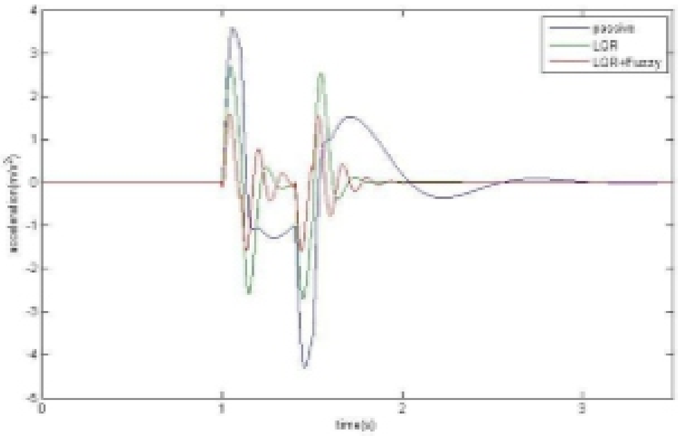

Figure 23- The acceleration of vehicle body

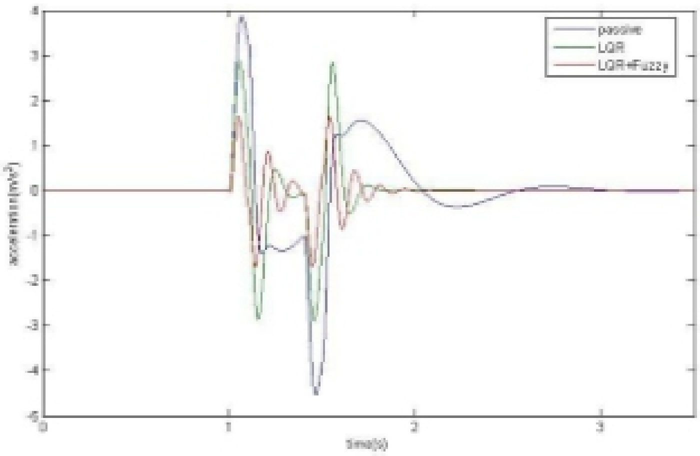

Figure 24- The acceleration of human head

Response to sinusoidal input

Response to sinusoidal input with the frequency of 1 Hz:

Tire displacement with respect to the road, acceleration of vehicle body and human head are given in figures 25, 26, and 27.

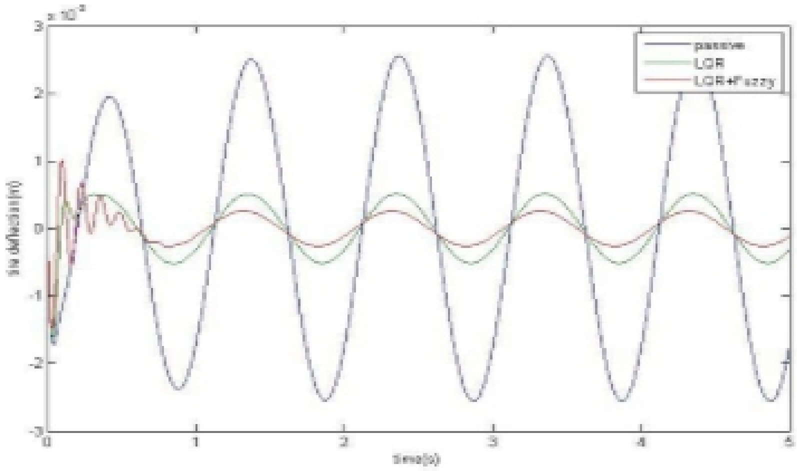

Figure 25- The tire displacement with respect to the road

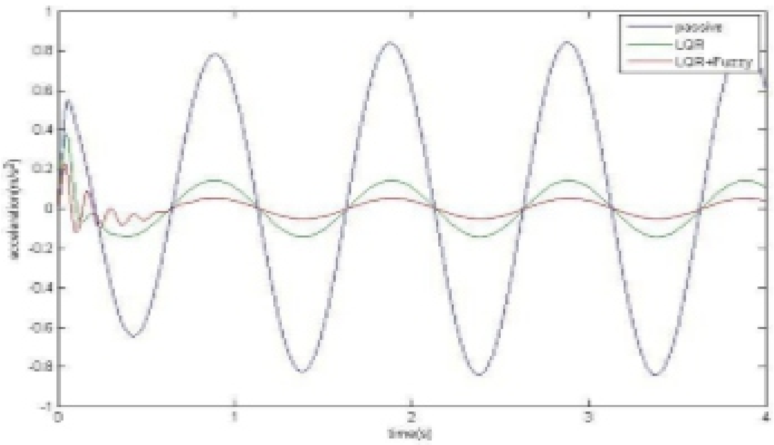

Figure 26- The acceleration of vehicle body

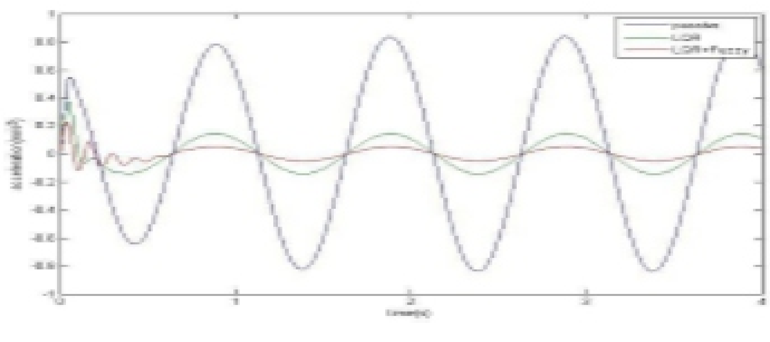

Figure 27- The acceleration of human head

Response to sinusoidal input with the frequency of 10 Hz:

Tire displacement with respect to the road, acceleration of vehicle body and human head are given in figures 28, 29, and 30.

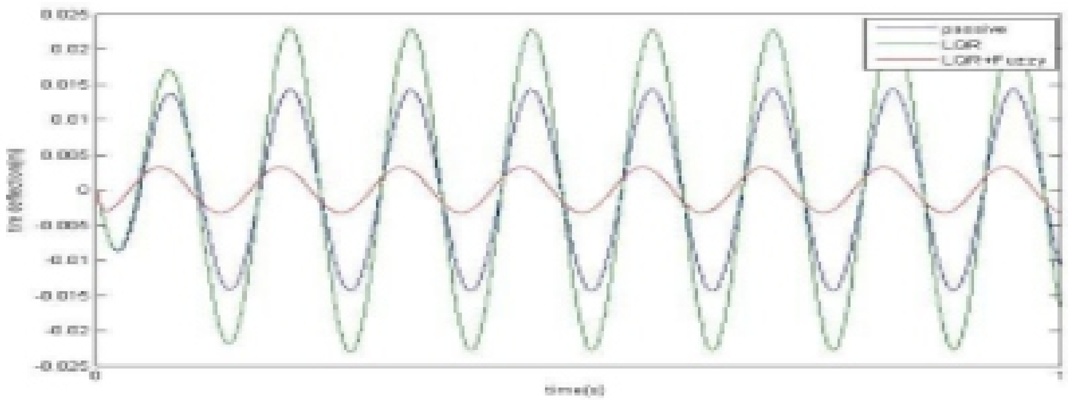

Figure 28- The tire displacement with respect to the road

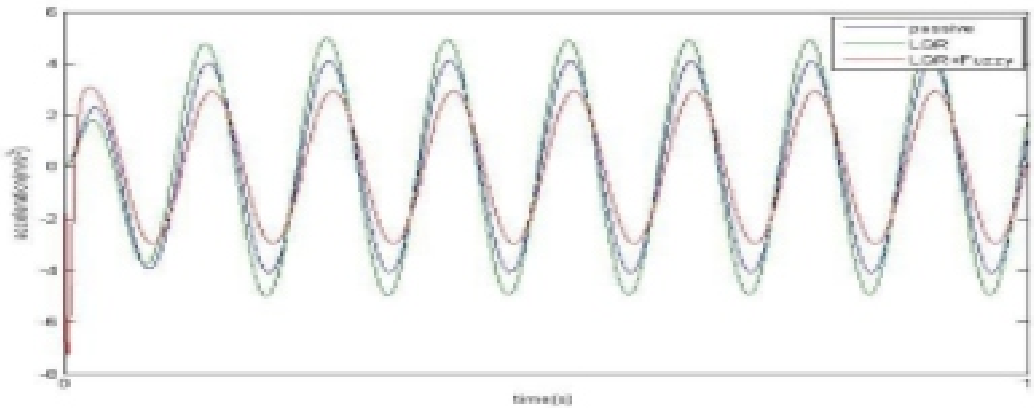

Figure 29- The acceleration of vehicle body

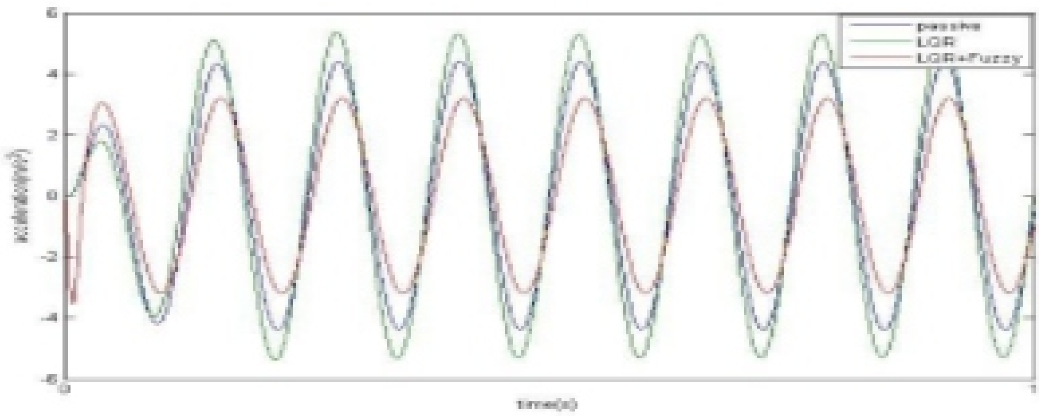

Figure 30- The acceleration of human head

Types of real roads:

Roads are divided into eight types of A to H based on amounts of roughness. In conclusion section, we have used type B, C, and D roads that are of the most common types of the roads. At first, we show fuzzy level for change in amounts of weighting functions in figures 31 and 32.

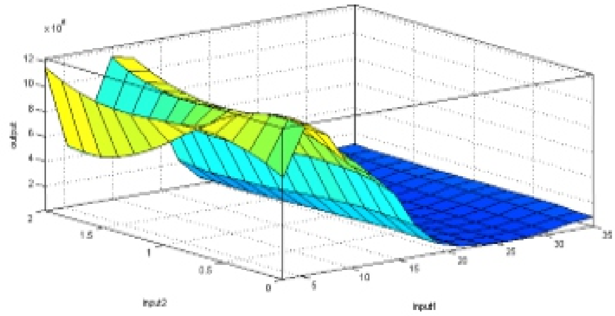

Figure 31- Fuzzy level for weighting function of ride comfort (control of weighting function of acceleration)

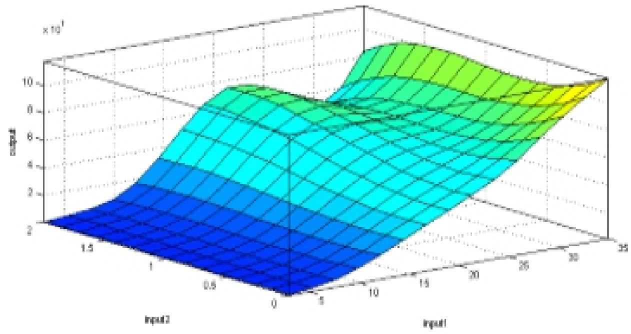

Figure 32- Fuzzy level for stability of weighting function (control of weighting function of tire displacement with respect to road input)

Type B road:

This type of road has a length of 100 m. Vehicle speed on this type of roads has been considered 20 m/s (72 km/h).

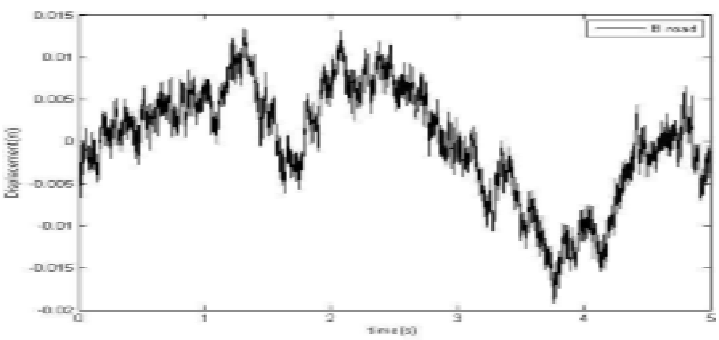

Figure 33- Type B road

We consider trend of variations in vehicle speed on the type B road according to figure 34:

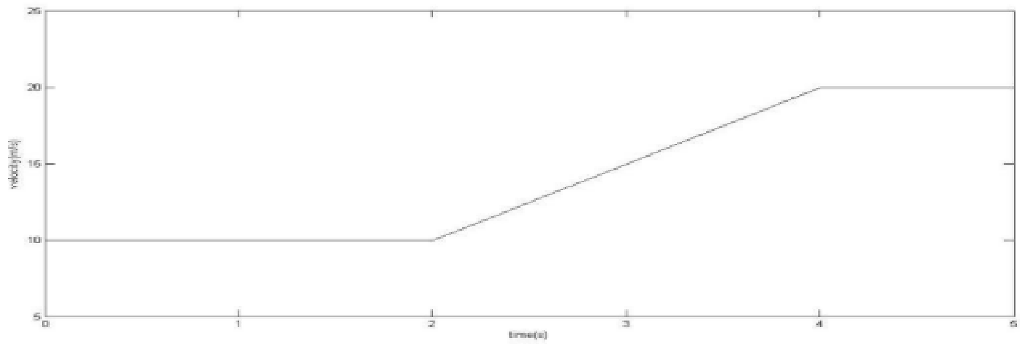

Figure 34- Trend of variations in vehicle speed on the type B road

The weighting function changes related to the tire displacement with respect to the road and the weighting function variations associated with human body acceleration are shown in figures 35 and 36, respectively.

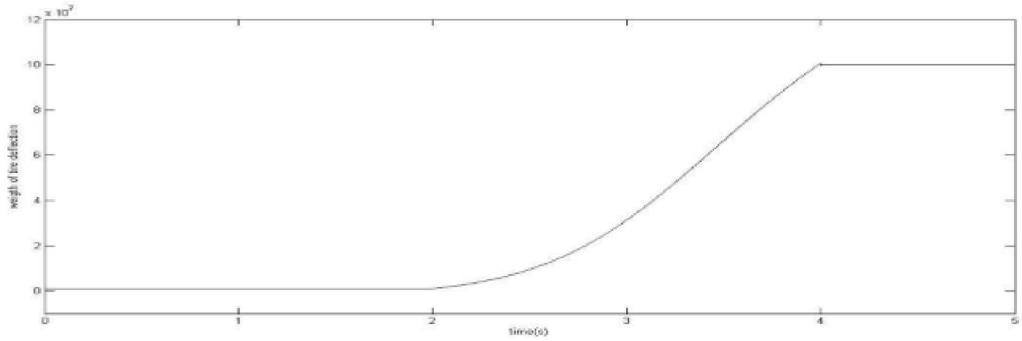

Figure 35- Trend of variations in weighting functions related to tire displacement with respect to the road

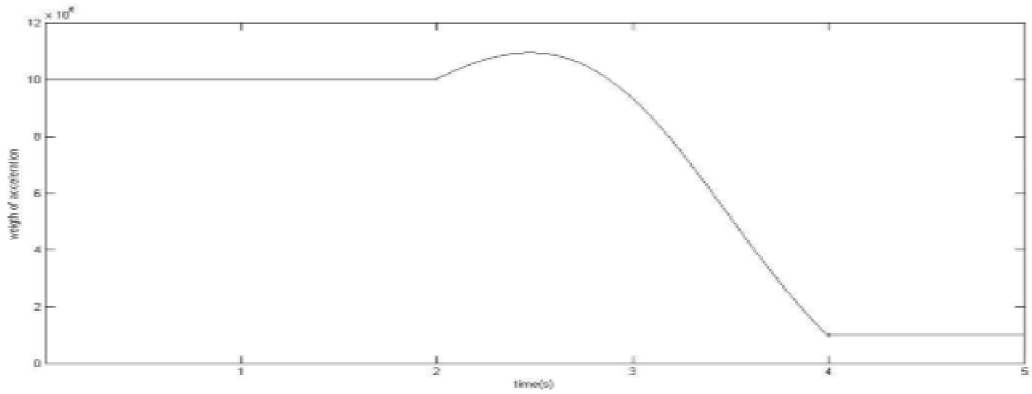

Figure 36- Trend of variations in weighting functions related to human body acceleration

Tire displacement with respect to the road and acceleration of human head are given in figures 37 and 38.

Tire displacement with respect to the road and acceleration of human head are given in figures 37 and 38.

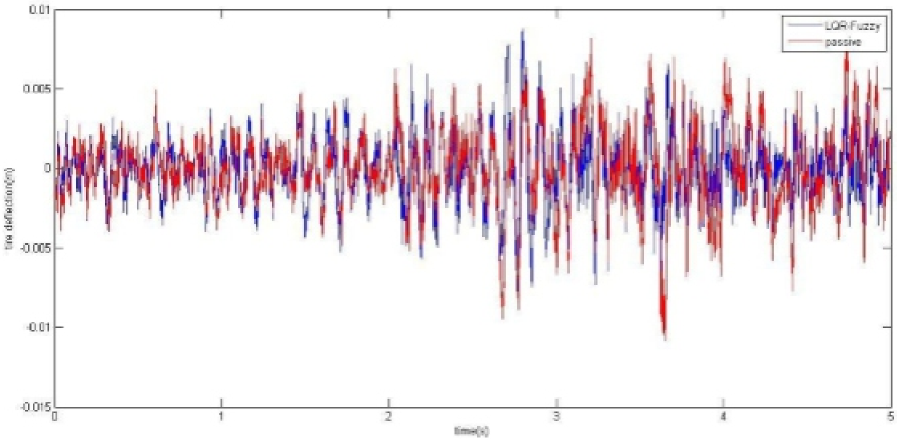

Figure 37- Tire displacement with respect to the road

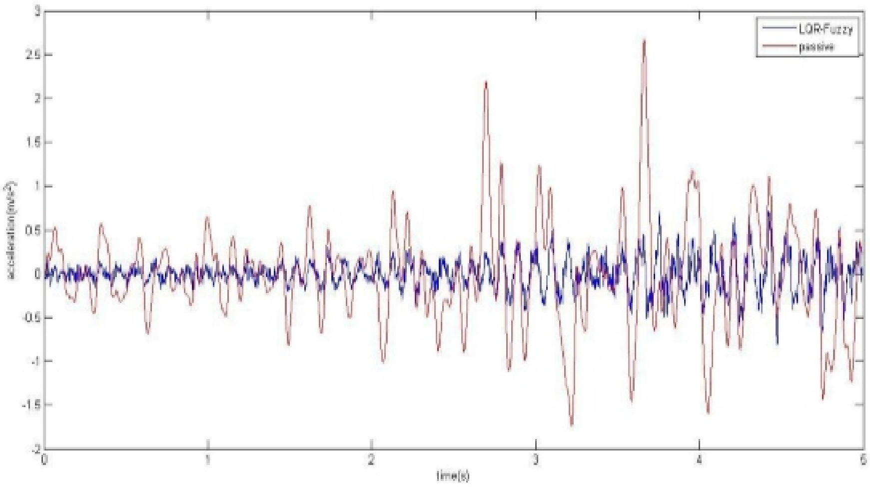

Figure 38- The acceleration of human head

As we can see in the curves, acceleration decreases in lower speeds that are important for our ride comfort and tire displacement with respect to the road has been reduced in higher speeds in which stability is more important to us.

Type C road:

This type of road has a length of 100 m. Vehicle speed on this type of roads has been considered 11 m/s (40 km/h).

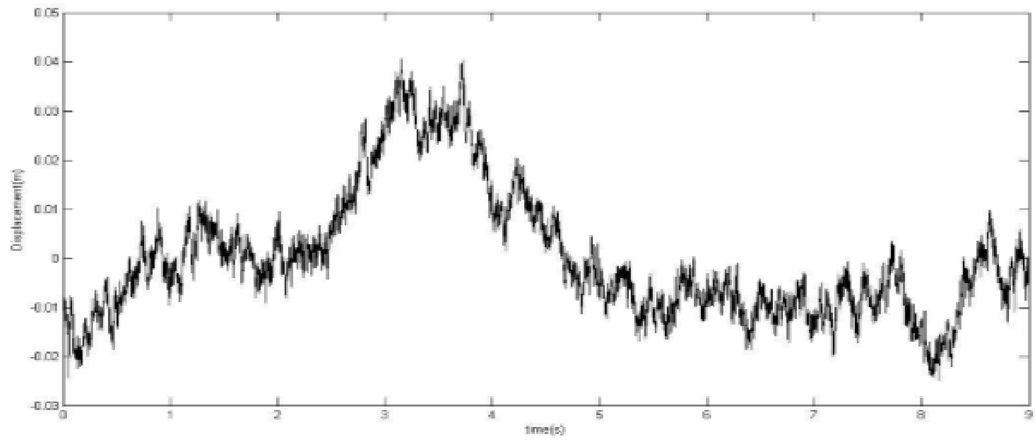

Figure 39- Type C road

We consider trend of variations in vehicle speed on the type C road according to figure 40:

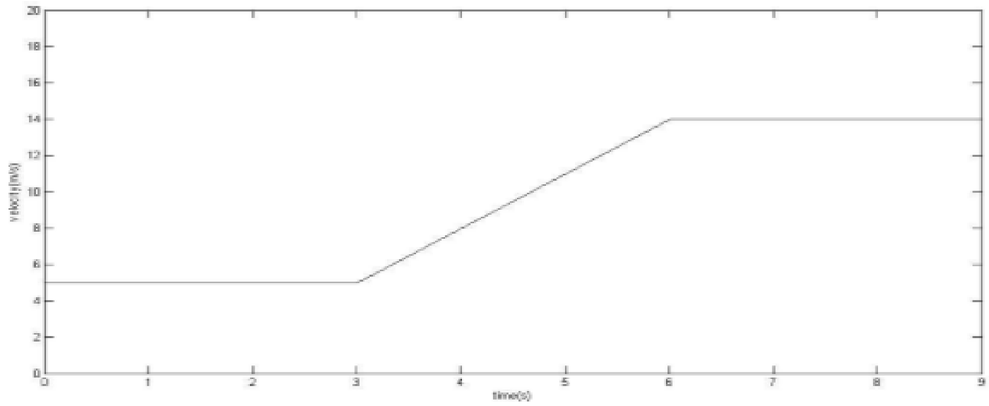

Figure 40- Trend of variations in vehicle speed on the road

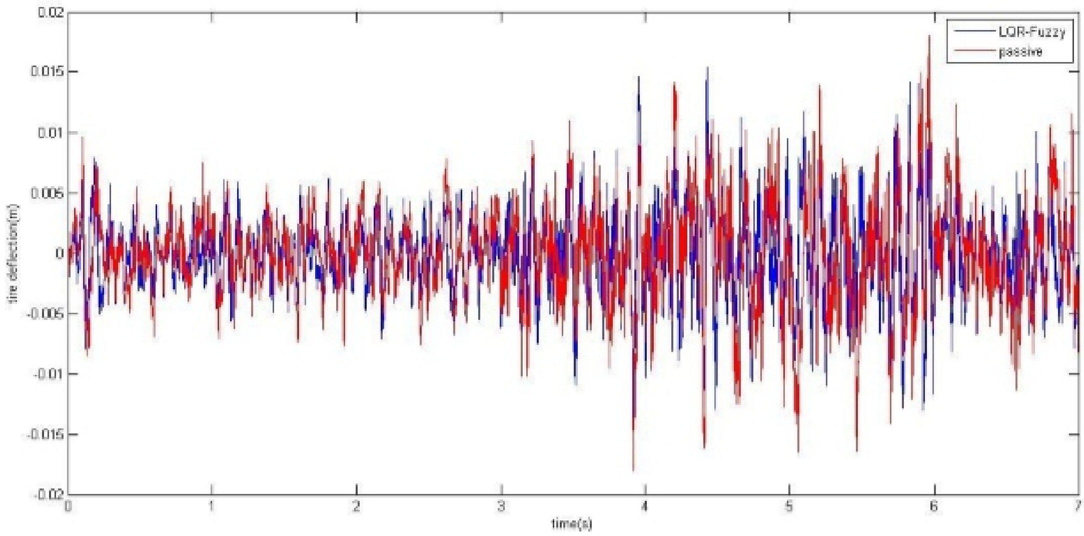

Figure 41- Tire displacement with respect to the road

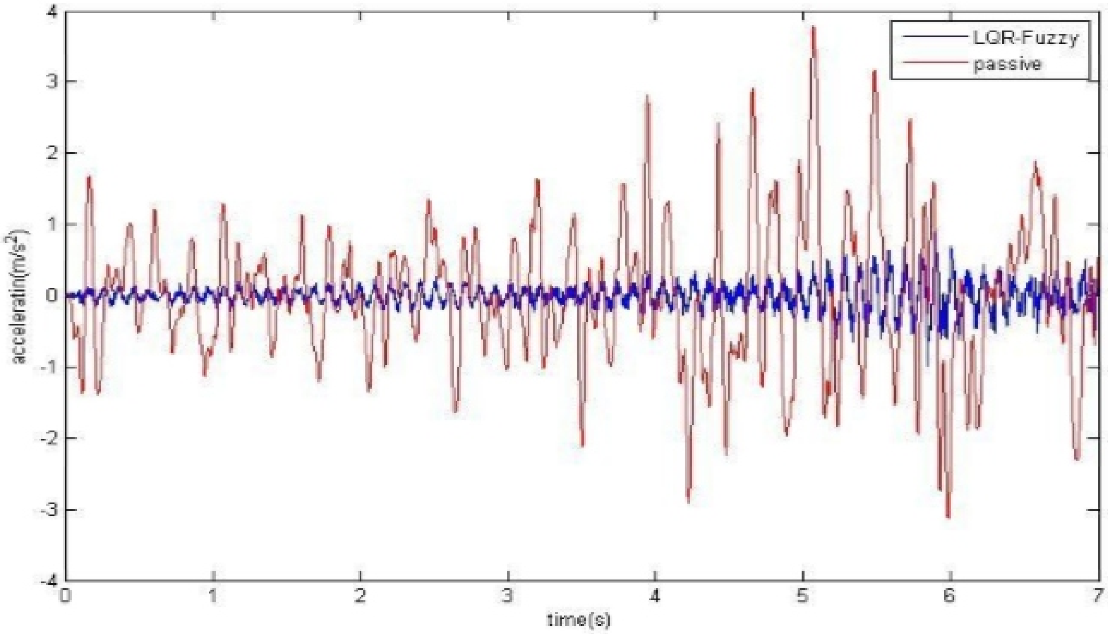

Figure 42- The acceleration of neck and head

Type D road:

This type of road has a length of 100 m and vehicle speed on this type of roads has been considered 5.5 m/s (20 km/h).

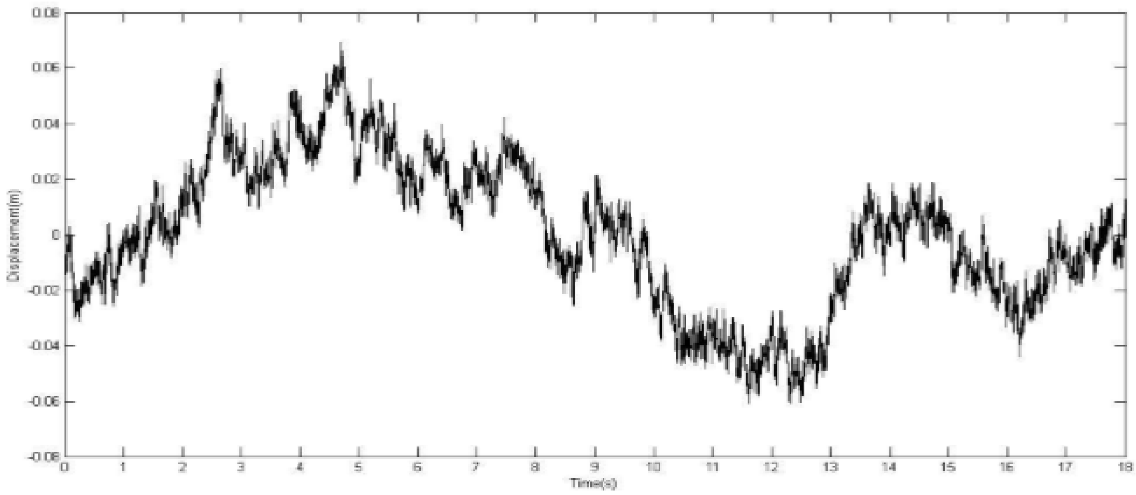

Figure 43- Type D road

We consider trend of variations in vehicle speed on the type D road according to figure 44:

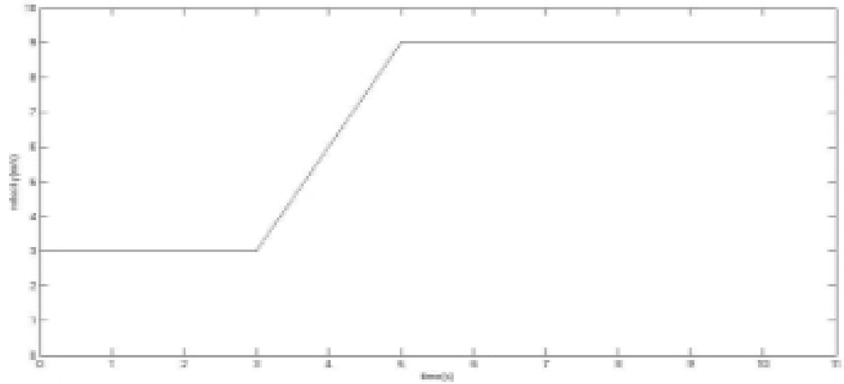

Figure 44- Trend of variations in vehicle speed

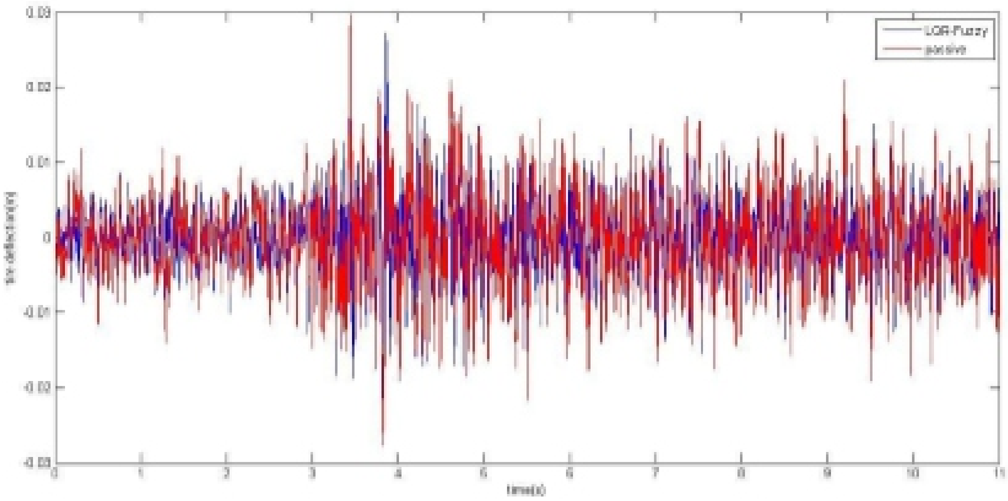

Figure 45- Tire displacement with respect to the road

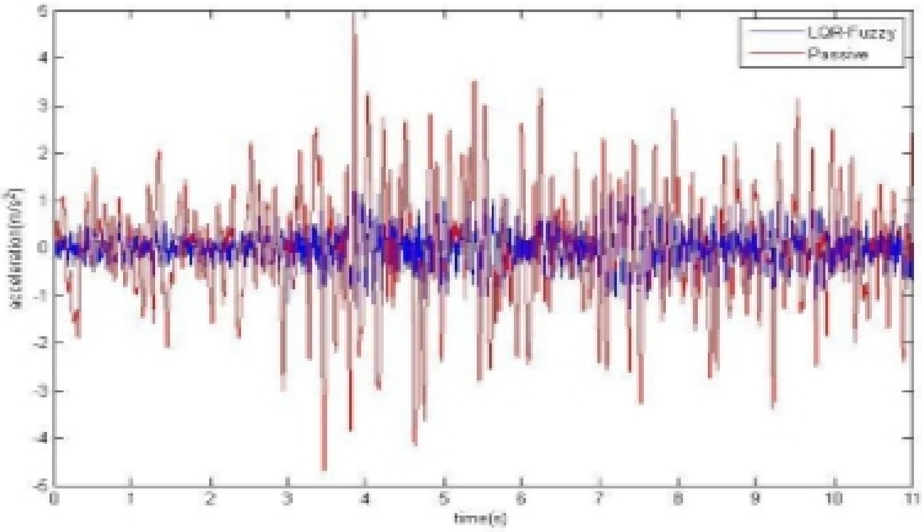

Figure 46- The acceleration of head and neck

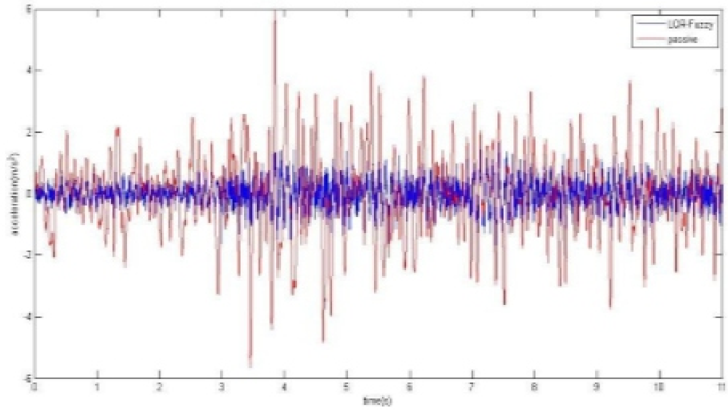

Figure 47- The acceleration of chest, stomach and intestine (viscera)

To be clearer the results obtained from the real road input and the effect of each of controllers, we can use fast Fourier transform (FTT) acceleration curves. As an instance, in this section we show FFT of acceleration of vehicle body and human head caused by type C road input in figures 48 and 49.

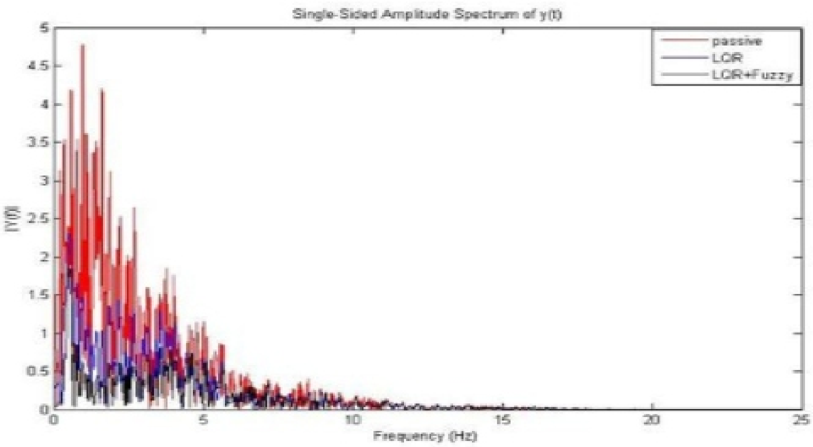

Figure 48- The acceleration of vehicle body

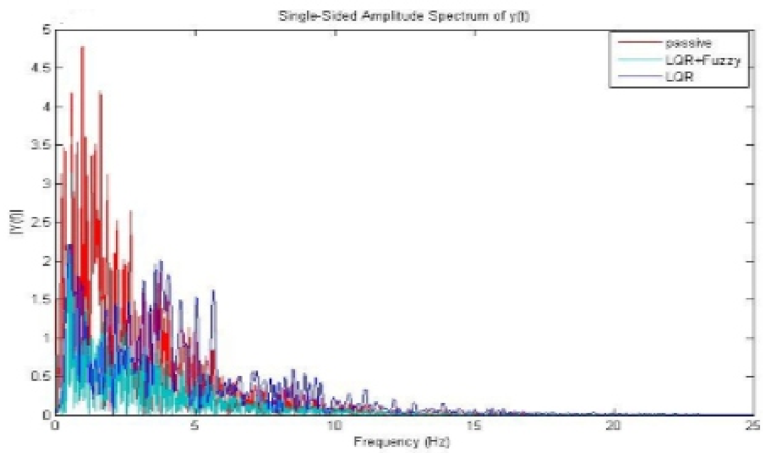

Figure 49- The acceleration of the human head and neck

CONCLUSION

By observing and investigating the results in curves, it is seen that the designed controller in this study greatly decreases accelerations entered to the human body in sensitive frequencies of the body and causes no fatigue for the driver during trip. Also, the occurrence of the vehicle instability in critical situations can be prevented by changing controller weighting functions using fuzzy controller.

SUGGESTIONS

Following suggestions are stated:

1- Considering lateral acceleration as the input of fuzzy controller for determining maneuver state and predicting critical situations.

2- Practical implementation of this study on 1/4 model and the mass that represents one degree freedom model of human body.

3- Using this system for controlling a real vehicle and applying changes in designing the suspension system for embedding the controller.

4- Considering changes of human body model parameters, spring coefficients, and damper used in semi-active suspension system as the input for fuzzy controller to strengthening the system with respect to large changes in system parameters.

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Performance and Emission Analysis of A Diesel Engine Burning Biodiesel From Waste Cooking Oil

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A B S T R A C T

Biodiesel hasavital role in recent years as an alternative fuel for diesel engines. Biodiesel is produced from waste cooking oil by transesterification process. Biodiesel produced from waste cooking oil gives comparable performance and emission characteristics. In this study, a comparative study has been done between blends of biodiesel derived from waste cooking oil and diesel fuels. Diesel- waste cooking oil biodiesel blends of 10 and 20% was prepared. Experimental investigations were tested in a four stroke, single cylinder, diesel engine at a constant speed of 1500 rpm and variable loads. Diesel- biodiesel blends provided significant increase in fuel consumption, specific fuel consumption, exhaust gas temperature, NO_x, CO_2 and oxygen concentrations about diesel fuel. Waste cooking oil biodiesel blends resulted in significant decrease in thermal efficiency, volumetric efficiency, air- fuel ratio, CO and HC emissions about diesel fuel. It is recommended to use waste cooking oil biodiesel up to 20% with diesel fuel without any engine modifications.

Keywords: Waste Cooking Oil, Biodiesel, Diesel Engine, Performance, Exhaust Emissions

Introduction

Waste cooking oil methyl ester was produced from cooking oils using transesterification method. Diesel fuel was blended by waste cooking oil biodiesel with a ratio of 25% on volume basis. Specific fuel consumption for blend B25 increased up to 5.69% compared to diesel fuel. However, HC and CO emission reductions compared to diesel fuel were found to be around 16.24% and 19.81%, respectively. But the amount of increase in NO_x emissions for the same biodiesel fuels reached up to 17.2%. Transesterification method was used to reduce viscosity of waste cooking oil. Thermal efficiency for blend B25 were slightly lower than those of diesel fuel. CO and HC emissions reduced when the blend fuels of B25 in diesel engines. The reductions in these exhaust emissions for waste cooking oil biodiesel were 11.66% and 23.12%, respectively [1]. Waste cooking oil biodiesel fuel produced by transesterification of waste cooking oil had shown very promising chemical and physical properties compared with diesel fuel. Biodiesel of waste cooking oil B50 resulted in a considerable reduction in unburnt hydrocarbons associated with an increase in the CO_2 and NO_x emissions due to advance of fuel injection timing. Results indicated an increase in specific fuel consumption with simultaneous reduction in the engine thermal efficiency compared to conventional diesel fuel due to the oxygen content and the lower calorific value of biodiesel compared to diesel fuel [2]. Waste cooking oil methyl ester and its blends with diesel fuel of 20%, 40%, 60% and 80% had been studied. Biodiesel blends gave a reduction of carbon monoxide, hydrocarbon and increase in nitrogen oxides emissions.

Engine performance reduced with increase

in biodiesel percentage in the blend. The exhaust gas temperature for the blends was higher compared to that of standard diesel fuel. Emission of oxides of nitrogen from the waste cooking oil blend B40 is higher than that of diesel fuel. CO emission of the blend B40 is closer to the standard diesel fuel. CO_2 emission is also lesser at the same conditions. The experimental result also proves that lower and medium percentages of waste cooking oil methyl ester can be substituted for diesel fuel [3].

Five test fuels such as diesel fuel, waste cooking oil biodiesel B5, waste cooking oil biodiesel B20, and waste cooking oil biodiesel B30 were investigated. Experimental results indicate that hydrocarbons emissions reductions by 10.5% to 36.0%, and carbon monoxide by 3.33% to 13.1% as compared diesel fuel. Specific fuel consumption was higher for biodiesel blends because of biodiesel having a lower heating value compared to diesel fuel [4]. Experiments were conducted on direct injection diesel engine using diesel fuel, biodiesel and their blends to investigate the exhaust emissions of the engine under different engine loads at an engine speed of 1800 rpm. Blended fuels containing 19.6%, 39.4%, 59.4% and 79.6% by volume of biodiesel, corresponding to 2%,4%, 6% and 8% by mass of oxygen in the blended fuel, were used. Biodiesel used in this study was converted from waste cooking oil. Specific fuel consumption and the thermal efficiency increase. The HC and CO emissions decrease while NO_x emission increases [5].Waste cooking oil biodiesel blends of B5 and B10 were tested and compared to diesel fuel. Biodiesel blends of B5 and B10 resulted in slightly increment on specific fuel consumption up to 4% and reduction on thermal efficiency up to 2.8%. Biodiesel additions increased NO_x emissions up to 8.7% and decreased of hydrocarbon emissions for the all engine loads. There were no significant changes on CO emissions at the low and medium engine loads, some reductions were observed at the full engine load. CO_2 emissions were slightly increased for the all engine loads. HC emissions showed decreasing trend up to 5% for low and medium engine loads and up to 29% for the high engine load with the addition of the biodiesel fuel [6].

With increase percentage of waste cooking oil biodiesel in diesel-biodiesel blends, higher exhaust gas temperature of waste cooking oil biodiesel which increased with percentage increase of waste cooking oil biodiesel in the blend. A slight increase in NO_x was produced when compared to diesel at rated load.

This was due to higher oxygen content of biodiesel and its blends. CO and HC emissions were found to significantly decrease with biodiesel and its blends due to a more complete combustion caused by higher oxygen content. An increase in specific consumption had been found when using biodiesel blends compared to diesel fuel due to the lower heating value of biodiesel and its blends. There was a decrease in thermal efficiency with increase in percentage of biodiesel in biodiesel blends [7, 8, 9, 10].Waste cooking oil ethyl ester was prepared by transesterification using potassium hydroxide as catalyst and was used in diesel engine. Tests were carried out at a rated speed of 1500 rpm at different loads. Vegetable oils pose operational and durability problems when subjected to long term usages in diesel engines. These problems are attributed to high viscosity and low volatility of vegetable oils. Transesterification was found to be an effective method of reducing vegetable oil viscosity and eliminating operational and durabilityproblems. The exhaust gas temperature increased with increasing biodiesel concentration. Waste cooking oil ethyl ester blends showed performance characteristics close to diesel fuel. Although waste cooking oil biodiesel heating value is lower that of diesel fuel by about 15%. Blend B10 exhibits a heating value about 45.50 MJ/kg is only 2% lower than that of diesel fuel. NO_x emission for waste cooking oil biodiesel is same as that of diesel fuel at lower loads and slightly higher at full loads [8].A diesel engine test using waste cooking biodiesel fuel was run to engine performance and exhaust emissions of diesel engine. By adding 20% of waste vegetable oil methyl ester, the concentration of the CO and HC emissions were significantly decreased when biodiesel was used [11, 12, 13, 14].

2. WASTE COOKING OIL BIODIESEL PRODUCTION

Waste cooking oil has sufficient potential to run diesel engines. It is available in the local market at cheaper rate. Huge quantities of waste cooking oil can be collected from restaurants and food item industry. Viscosity of waste cooking oil is about ten times, greater and its density is about 10% higher than that of diesel fuel. Transesterification process is used to reduce the viscosity of waste cooking oils and converting it to biodiesel. Biodiesel is derived from vegetable oils or animal fats which are basically long chain triglyceride esters with free fatty acids. The long chain triglyceride ester is converted into mono ester by the process called transesterification. In this process, the vegetable oils were reacted with methanol or ethanol in the presence of acid or base catalyst producing fatty acids methyl or ethyl ester. In this study, used cooking oil was transesterified using 1% sodium hydroxide and 20 % methanol at the temperature range of 65-69 C. The reaction time was two hours and conversion efficiency was 92.5%. Properties of waste cooking oil biodiesel such as density, viscosity, heating value and flash point were measured in Egyptian Research Petroleum Institute, Cairo and were indicated in Table 1 [10, 11, 12].

Table 1. Properties of waste cooking oil biodiesel, diesel fuel and their blends.

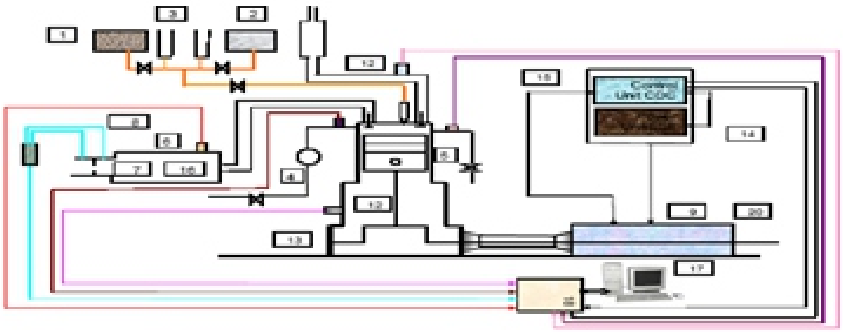
Properties	Diesel	B20	B100
Density (kg/m^3)	830	841	885
Kinematic Viscosity (mm^2/sec) at 40°C	3.05	4.9	12.5
Flash Point°C	65	74	179
Heating Value (MJ/kg)	42	41.25	38.6

3. EXPERIMENTAL SET UP

The present study was carried out to investigate the performance and emission characteristics of biodiesel derived from waste cooking oil blends with diesel fuel in diesel engine and compared to diesel fuel. The test engine is a Kirloskar make, single cylinder, four stroke, water cooled, direct injection, AV1 model diesel engine. Its specifications are given in Table 2. The engine was connected to an eddy current dynamometer to measure the power output and speed. The engine was equipped to measure fuel consumption, engine speed and exhaust gas temperature. The engine receives air through an air box fitted with an orifice for measuring the air consumption. A pressure differential meter is used to measure the difference in pressure between the two sides of the orifice. Fuel consumption rate was determined using a glass burette and stop watch. The engine speed was measured using a digital tachometer. MRU DELTA 1600-V exhaust gas analyzer was used for measuring the exhaust gas emission concentrations of CO, HC, CO_2, O_2 and NO_x. A data acquisition card (National Instrument 6210) was used to acquire data to be fed to personal computer. The schematic diagram of experimental set up and test rig is shown in Fig.1. The engine was warmed up before taking all readings. When the engine reached its stable condition, the experiments were started and measurements recorded. The engine was then operated with blends of diesel and waste cooking oil biodiesel (B10 and B20). For every operating condition, the engine speed was checked and maintained constant at rated speed of 1500 rpm. The performance parameters and exhaust gas emissions investigated were fuel consumption, specific fuel consumption, thermal efficiency, exhaust gas temperature, volumetric efficiency, air- fuel ratio, carbon dioxide (CO_2), carbon monoxide (CO), nitrogen oxides (NO_x), unburned hydrocarbons (HC) and oxygen (O_2) concentrations.

Table 2. Test Engine Specifications.

Engine parameters	Specifications
Type	Kirloskar
Number of cylinders	Single
Cycle	Four stroke
Cooling	Water
Cylinder diameter (mm)	85
Piston stroke (mm)	110
Compression ratio	17.5:1
Governing speed	1500 rpm
Rated power (HP)	6.5



1- Diesel Tank	2- Biodiesel Tank
3- Burette	4- Inlet water temperature sensor
5- Outlet water temperature sensor	6- Intake air temperature sensor
7- Orifice	8- pressure differential sensor
9- Dynomemeter	10- Power unit
11- Exhaust temperature sensor	12- Oil temperature sensor
13- Diesel engine	14- power unit
15- Control unit	16- Air tank
17- Personal computer	18- Data acquisition card

Fig.1. Schematic Diagram of a diesel Test Engine fueled with Diesel and Biodiesel Blends.

4. RESULTS AND DISCUSSION

Fuel Consumption

The variation of fuel consumption with brake power is shown in Fig.2. It is observed that as the load increased, fuel consumption increased for all the fuels. This variation increased from lower loads to higher loads due to the increase in injected fuel with the increase in output power.

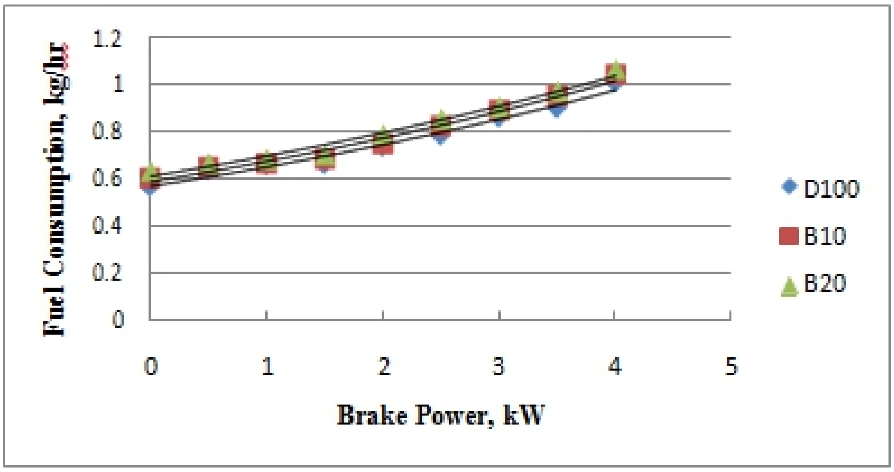


Fig.2. Variation of Fuel Consumption with Brake Power for biodiesel blends

As percentage of biodiesel increases, the fuel consumption tends to increase because of lower heating value of biodiesel. At full load, the maximum increase in fuel consumption for blend B10 and B20 in comparison with diesel fuel was about 1.4 and 2.6 %, respectively. Above results are confirmed with these references [12, 13].

Specific Fuel Consumption

Variation of specific fuel consumption with brake power for diesel and diesel- biodiesel blends was shown in Fig.3. Specific fuel consumption decreased with the increase in load for all fuels due to increase of fuel consumption with load. Specific fuel consumption for biodiesel blends was higher than diesel fuel. This may be due to higher fuel density, higher viscosity and lower heating value of biodiesel compared to diesel fuel and this leads to lower heat content. At full load, the highest value of specific fuel consumption for blend B10 and B20 in comparison with diesel fuel was about 1.3 and 2.2 %, respectively. These results are confirmed by references [13, 14, 15, 16, 17, 18, 19].

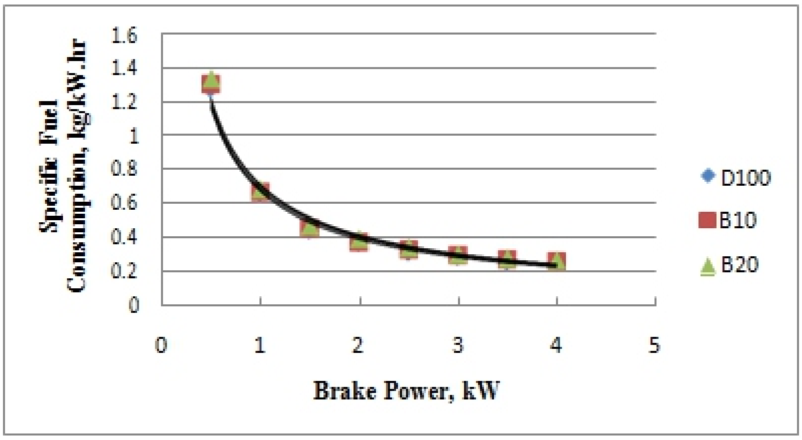


Fig.3 Variation of Specific Fuel Consumption with Brake Power for biodiesel blends.

Thermal Efficiency

The variation of thermal efficiency with brake power for diesel and diesel- biodiesel blends was shown in Fig.4. Thermal efficiency was having tendency to increase with increase in engine load. This was due to the reduction in heat loss and increase in power developed with increase in engine load. Thermal efficiency for biodiesel blends was lower than diesel fuel. This may be due to poor atomization, higher viscosity and reduction in heat loss of biodiesel blends compared to diesel fuel. Decrease of thermal efficiencies of biodiesel blends compared to diesel fuel was due to higher fuel consumption and lower heating value of biodiesel. At full load, the maximum decrease in thermal efficiency for B10 and B20 about diesel fuel was 2.5 and 3.5%, respectively. The above results are in agreement with the results reported by other researchers [14, 15, 16, 17, 18, 19, 20].

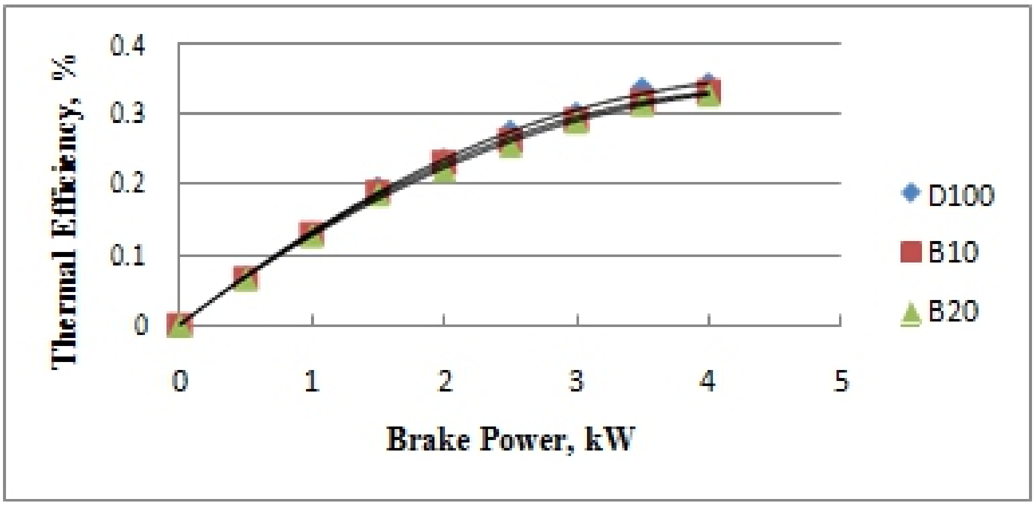


Fig.4.Variation of Thermal Efficiency with Brake Power for biodiesel blends.

Exhaust Gas Temperature

Figure 5 showed the variation of exhaust gas temperature with brake power for diesel and diesel-biodiesel blends. As load increased, exhaust gas temperature increased for all fuels. At full load, the chemically correct ratio of air and fuel was used and high heat was generated. Exhaust gas temperatures for all biodiesel blends were higher than diesel fuel due to the higher viscosity of biodiesel and the oxygen content. For biodiesel blends, the combustion was delayed due to higher physical delay period.

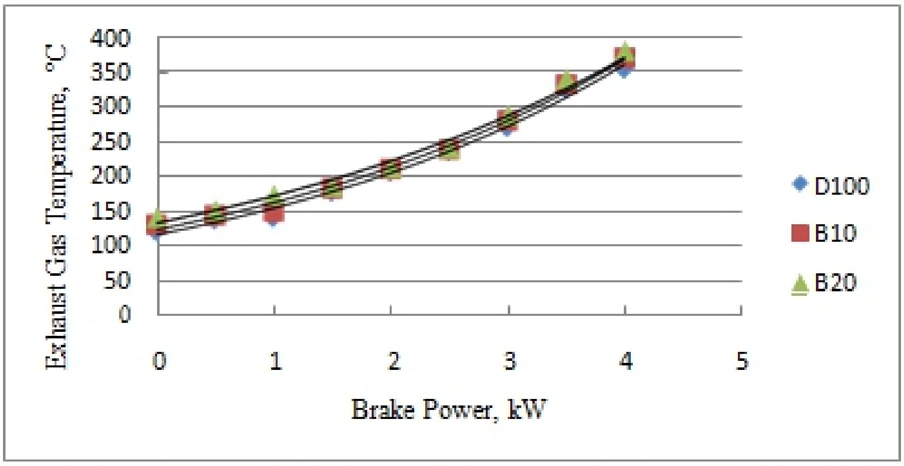


Fig.5 Variation of Exhaust Gas Temperature with Brake Power for biodiesel blends

As the combustion was delayed, injected biodiesel fuel molecules may not get enough time to burn completely before top dead centre, hence some fuel- air mixtures tended to burn during the early part of expansion, consequently after burning occurred and the exhaust gas temperature increased. At full load, values of exhaust gas temperatures for diesel, B10 and B20 fuels were 355, 370 and 380°C. These results are confirmed with these references [8, 11, 16].

Volumetric Efficiency

In Fig.6, variation of volumetric efficiency with engine load was shown. The volumetric efficiency increased with the increase in engine load. This was due to the less flow restrictions in the air filter and intake manifold. This led to increase in the amount of air enters the cylinder. It was seen that volumetric efficiencies were lower for diesel- biodiesel blends compared to diesel fuel. Volumetric efficiency decreased with the increase in biodiesel percentage in biodiesel blends because biodiesel fuel contains oxygen which decreased the amount of air needed for complete combustion. At full load, the maximum decrease in volumetric efficiency for B10 and B20 about diesel fuel is 3.5 and 8%, respectively.

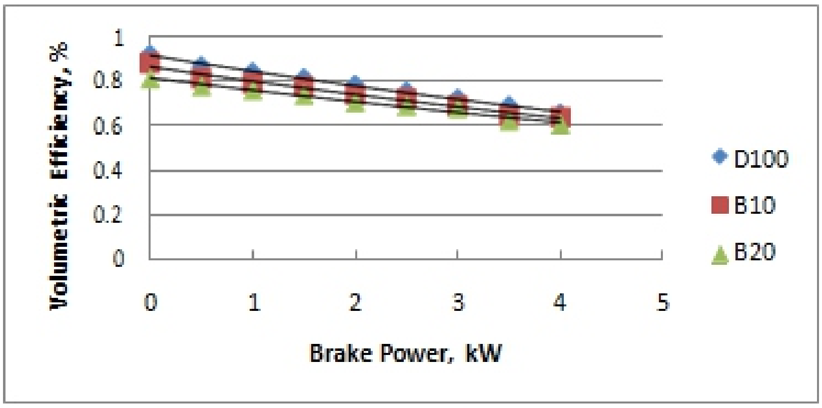


Fig.6. Variation of Volumetric Efficiency with Brake Power for Biodiesel Blends.

Air- Fuel Ratio

The variation air-fuel ratio with load for diesel and biodiesel blends was shown in Fig.7. A richer mixture was needed at higher loads. Air- fuel ratio decreased with the increase in load due to the increase in mass of fuel and the compensation of load with increasing the amount of fuel injected. Fuel consumptions were higher for biodiesel blends compared to diesel fuel hence air-fuel ratio decreased. Air- fuel mixing process was affected by the problems appear in atomization of biodiesel due to its higher viscosity. At full load, the maximum decrease in air- fuel ratio for B10 and B20 were about diesel fuel is 4.5 and 11 %.

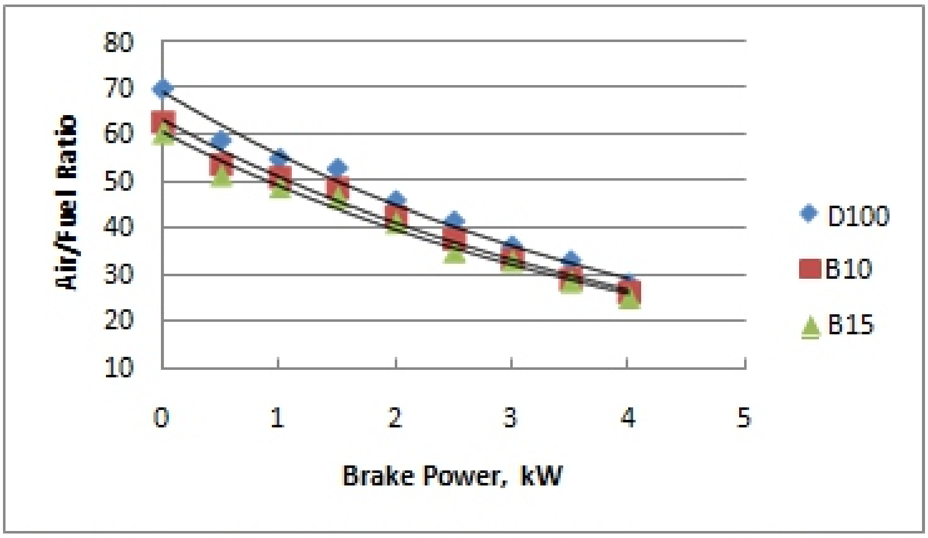


Fig.7 Variation of Air- Fuel Ratio with Brake Power for Biodiesel Blends.

5. ENGINE EXHAUST EMISSIONS

Carbon Dioxide Emission

It can be observed from Fig.8 that the amount of CO_2 emission increased with the increase of engine load due to higher fuel consumption at higher loads. Lower percentages of CO_2 emissions were produced when diesel engine fueled with biodiesel blends compared to diesel fuel. This was due to the lower carbon to hydrogen ratio in biodiesel blends compared to diesel fuel. Diesel fuel has 85% carbon atoms while biodiesel has about 76%.Biodiesel had oxygen content which improved combustion. B20 and B10 produced very low levels of CO_2 emissions. Using higher concentrations of biodiesel, CO_2 emission levels were lower than that of diesel fuel. At full load, the values of CO_2 emissions for diesel, B10 and B20 fuels were 6.5, 6.3 and 6.2%, respectively. The above results are in agreement with other researchers [13,17,19, 20, 22].

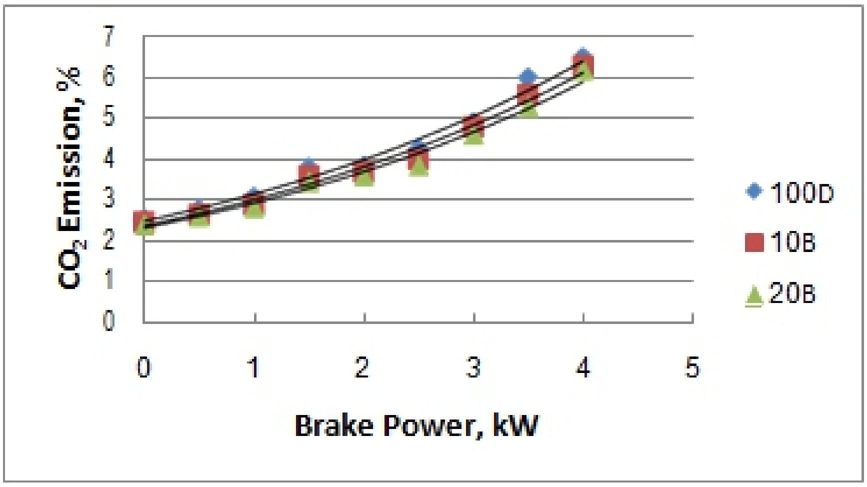


Fig.8. Variation of CO_2 Emission with Brake Power for biodiesel blends

Carbon Monoxide Emission

Carbon monoxide emissions of diesel and waste cooking oil biodiesel blends are shown in Fig. 9. CO emissions increased with the increase in load due to decrease of air- fuel ratio at higher loads. The higher combustion temperature at lower engine loads contributed to the general decreasing trend of CO emission. The decrease in carbon monoxide emission for biodiesel blends was due to more oxygen molecule present in the fuel, improved atomization and better vaporization of biodiesel resulting in complete combustion as compared to diesel fuel. The higher amount of oxygen in biodiesel will promote further oxidation of CO emission. At full load, CO emission for B10 and B20 blends were about 6 and 18% lower than that of diesel fuel. The above results are closer to the results reported by other researchers [13, 14, 16, 19, 20, 21 and 22].

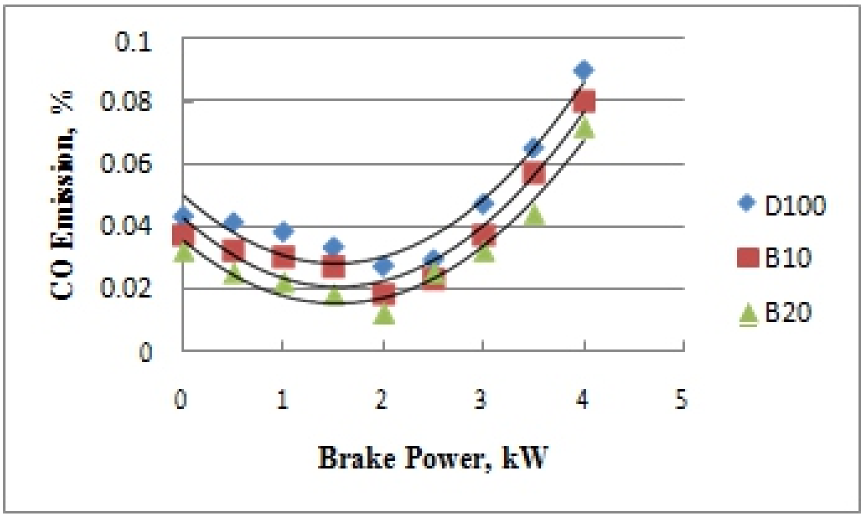


Fig.9.Variation of CO Emission with Brake Power for biodiesel blends.

Nitrogen Oxide Emission

Concentrations of NO_x emissions variation with brake power are shown in Fig.10. NO_x emission increased as the load increased due to higher combustion chamber temperature and higher fuel consumption. NO_x emissions of biodiesel blends increased with the increase of biodiesel percentage. NO_x emission is a function of oxygen inside the combustion chamber, combustion flame temperature and reaction time. Oxygen concentration in biodiesel blends might have caused the formation of NO_x emissions. Furthermore, the increase of NO_x emission was due to the higher cetane number of biodiesel which will reduce the ignition delay and higher combustion chamber temperature. Premix combustion of biodiesel blends led to higher emissions of NO_x. From diesel to B20, NO_x emission increased. The peak concentrations at full load were625 ppm, 630 ppm and 650 ppm for diesel, B10and B20, respectively. The above results are in agreement with other researchers [13, 14, 16, 18, 19, 20 and 21].

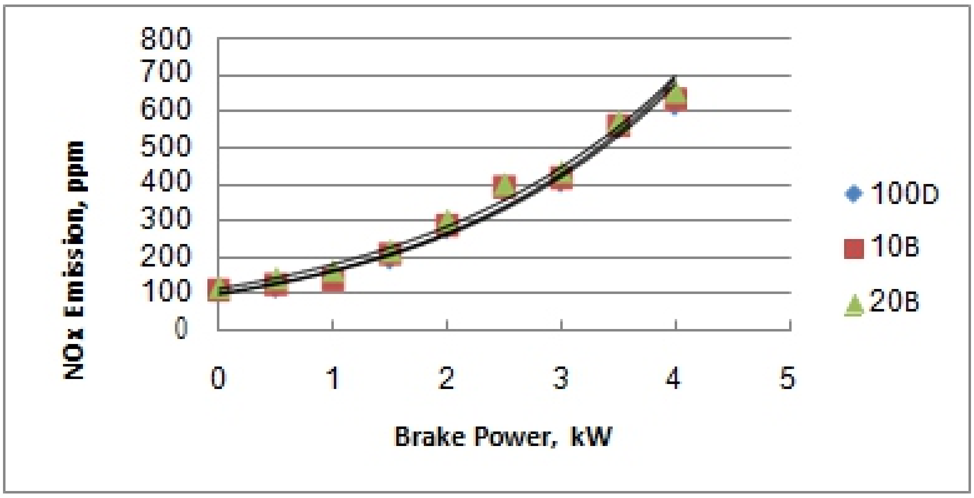


Fig.10.Variation of NOx Emission with Brake Power for biodiesel blends.

Hydrocarbons Emissions

Variations of HC emissions for diesel and biodiesel blends are shown in Fig. 11. HC emissions increased as the engine load increased due to the increase of fuel consumption at higher loads. Higher cetane number of waste cooking oil biodiesel resulted decrease in HC emission due to shorter ignition delay. Lower HC emissions of biodiesel blends were due to the presence of fuel bound oxygen and warmed up conditions. The maximum concentrations of HC are 32 ppm, 27 ppm and 23 ppm for diesel, B10 and B20 fuels, respectively.. The above results are closer to the results reported by other researchers [13, 14, 17, 19, 20, 21, 22].

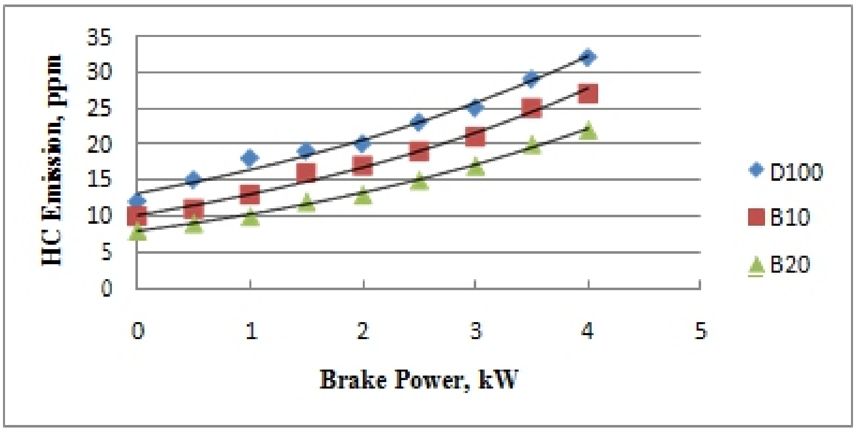


Fig.11. Variation of HC Emission with Brake Power for biodiesel blends.

Oxygen Concentration

The variation of oxygen concentration with brake power was shown in Fig.12. There is around 10% oxygen in the chemical structure of waste cooking oil biodiesel. A high amount of oxygen in biodiesel improved combustion and thus provided complete combustion. The observed decrease in oxygen content in the exhaust with increase in load may be due to richer mixture being burnt in the engine cylinder. The higher cylinder temperature led to produce a larger portion of oxygen which reacted with nitrogen and carbon to form CO, NO_x and CO_2 at higher loads. Hence less oxygen was released to the atmosphere. It can also be observed that the oxygen concentrations increased with increase in biodiesel percentage in diesel- biodiesel blends. The increase in oxygen concentration with increase in blend proportion may be due to the inherent oxygen present in biodiesel. It can be noted that the percentage of oxygen in the exhaust was maximum for biodiesel blends and it decreased for other biodiesel blends in the order B20, B10 and diesel fuel. The maximum increase of oxygen concentration for B20 and B10 in comparison with diesel fuel was about 3 and 1%, respectively. These results are confirmed with these references [1, 9].

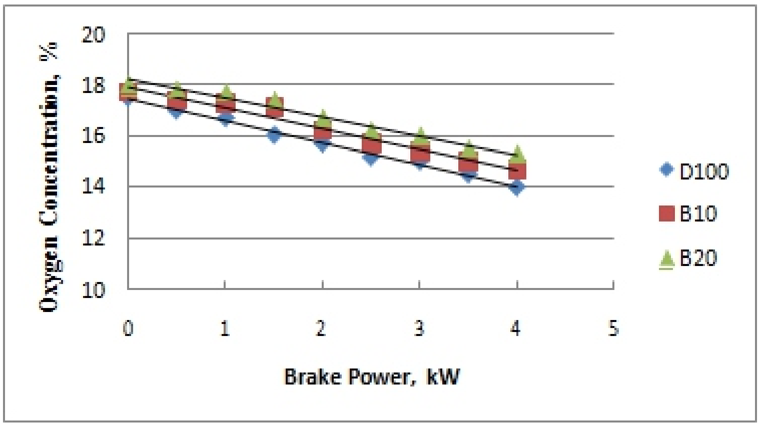


Fig.12.Variation of Oxygen Concentration with Brake Power for biodiesel blends.

6. CONCLUSIONS

Waste cooking oil biodiesel can be used as alternative fuel for diesel engines. Waste cooking oil biodiesel blends of 10 and 20% were tested in a four stroke, single cylinder, diesel engine at a constant speed of 1500 rpm and variable loads. Experimental results of biodiesel blends compared with diesel fuel showed that**:**

- Diesel- biodiesel blends showed increase in fuel consumption due to the lower heating value of the biodiesel. B20 and B10 showed an increase of 2.6 and 1.4 %, respectively in fuel consumption compared to diesel fuel. Biodiesel blends B20 and B10 showed increase in specific fuel consumption about 2.2 and 1.3%, respectively in comparison with diesel fuel.
- Biodiesel blends B20 and B10 showed decrease in engine thermal efficiency about
- 3.5 and 2.5%, respectively in comparison with diesel fuel.
- Volumetric efficiency for B10 and B20 achieved reductions of 3.5 and 8%, respectively and air-fuel ratio for B20 and B10 had reductions of 4.5 and 11%, respectively compared to diesel fuel.
- The exhaust gas temperature increased with the operation of biodiesel blends about diesel fuel. Biodiesel blends B20 and B10 and diesel fuels recorded about 380, 370and 355°C of exhaust gas temperatures.
- At full load, the values of CO_2 emissions for diesel, B10 and B20 fuels were 6.5, 6.3 and 6.2%, NO_x emissions values were 625, 630 and 650 ppm, respectively.
- At full load, the maximum values of HC emission for diesel, B10 and B20 fuels were about 32, 27 and 23 ppm, respectively.
- The maximum increase of oxygen concentration for B20 and B10 in comparison with diesel fuel was about 1 and 3%.
- Using neat waste cooking oil biodiesel in conventional diesel engine is not recommended. Waste cooking oil biodiesel blends can be used up to 20% with diesel fuel without any engine modifications. Performance and emissions of a diesel engine using biodiesel blends up to 20% with diesel fuel were closer to diesel fuel.
- Waste cooking oils are very suitable as low cost feed stocks for biodiesel production. The environment will be cleaned by collecting and recycling these waste oils, human health will be protected and reducing the dependency on fossil fuel resources.

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Characterization of Motor Lubricating Oils

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ABSTRACT

Very important component for safe and undisturbed functioning of any internal combustion engine is a sound lubrication of vital parts with engine (motor) oil. Simultaneously with engines development, new requirements were also set for engine oils in order to get more power from the engines to withstand higher working temperatures, wear and load and to last longer. At first mineral engine oils were used, but synthetic engine oils that could fulfil the highest requirements are more and more implemented. Nowadays, technically and commercially, engine oils have gained more than 60% of market share on the global market of lubricants. For proper functioning of the engine, engine oil must fulfil requirements: minimizing wear, assisting in cooling, keeping the compression ratio, reducing corrosion and friction and controlling the deposits. Lubricant characteristics and performances are managed by standard or industrial organization as API, ACEA, and SAE through specific norms. Each norm defines technical requirements as physical properties, engine tests results and other various criteria. This paper describes the importance of grade of engine oil as per SAE and compare various properties of engine oil as per their grade and brand of engine oil. It gives idea about the selection of engine oil depends on the climate condition, load, speed, driving condition and local availability etc.

Key words: lubrication, lubricants, engine oil, properties, additives

1. Preliminary Observation

Introduction

The principle of supporting a sliding load on a friction reducing film is known as lubrication. The substance of which the film is composed is a lubricant, and to apply it is to lubricate. A lubricant (sometimes referred to as "lube") is a substance (often a liquid) introduced between two moving surfaces to reduce the friction between them, improving efficiency and reducing wear [1]. Lubricants are comprised of a base fluid, usually of petroleum origin, combined with added chemicals that enhance performance. Base fluids are collected from two main sources.

Refined crude oil (the crude oil is refined into gasoline, diesel, kerosene, LPG, naphtha and base stocks (Lube) or a mixture of chemical compounds that perform the same task. Typically lubricants contain 90% base oil (most often petroleum fractions, called mineral oils) and less than 10% additives [2]. Lubricants play a vital role in every industry including: Electronic, Automotive, Aerospace, Forestry, Naval and numerous others. Lubrication failure may result in thousands of dollars of production losses including downtime and equipment failure. In this research paper only four wheeler gasoline engine oil is included for review purpose. It includes comparison between kinematic viscosity at 40˚C and 100˚C for different brands of engine oil, characterization of properties for different grade engine oil and takes a case study about maruti gasoline passenger car.

Properties of Lubricating Oil and its Additives [3]

The quality of a lubricating oil is tested for the following various properties like viscosity, Flash point, Pour point, Total Base Number (TBN) and Viscosity Index (VI) etc to evaluate its suitability and merits for certain service conditions.

Additives [4] are chemical compounds added to lubricating oils to impart specific properties to the finished oils. Some additives impart new and useful properties to the lubricant; some enhance properties already present, while some act to reduce the rate at which undesirable changes take place in the product during its service life. Table 1 shows the different additives, its purpose and typical compounds contained by it.

Table 1: Classification of additives

Additives	Purpose	Typical Compounds
Viscosity Index Improvers(VI)	Reduce the rate at which oil viscosity decreases with increasing temperature	Polyisobutylene, methacrylate polymers, olefin copolymers
Pour Point Depressants	Modify wax crystal formation to reduce interlocking	Alkylated naphthalene and phenolic polymers, polymethacrylates, maleate/fumerate copolymer esters
Emulsifiers	Promote formation of stable mixture (emulsion) of water & oil by changing interfacial tension	Soaps of fatty acids, sulfonic & napthenic acids, certain animal & vegetable oils
Friction Modifiers	Reduce or modify friction	Long chained polar compounds (amides, phosphates, phosphites, acids, etc)
Dispersants	Keep oil degradation by products, and/or combustion related by products in small suspended state within the bulk oil by preventing agglomeration	Alkylsuccinimides, alkylsuccinic esters, and mannich reaction products
Detergents	Keep surfaces free of deposits	Metallo-organic compounds of sodium, calcium and magnesium phenolates and phosphonates
Corrosion and Rust Inhibitor	Prevent corrosion and rusting of metal parts in contact with the lubricant	Zinc dithiophosphates, metal phenolates, basic metal sulfonates, fatty acids and amines
Anti-Wear (AW) additives	Reduce friction and wear and prevent scoring and seizure	Zinc dithiophosphates, organic phosphates, acid phosphates, organic sulfur and chlorine compounds, sulfurized fats,

2. Lubricants Market in India[5]

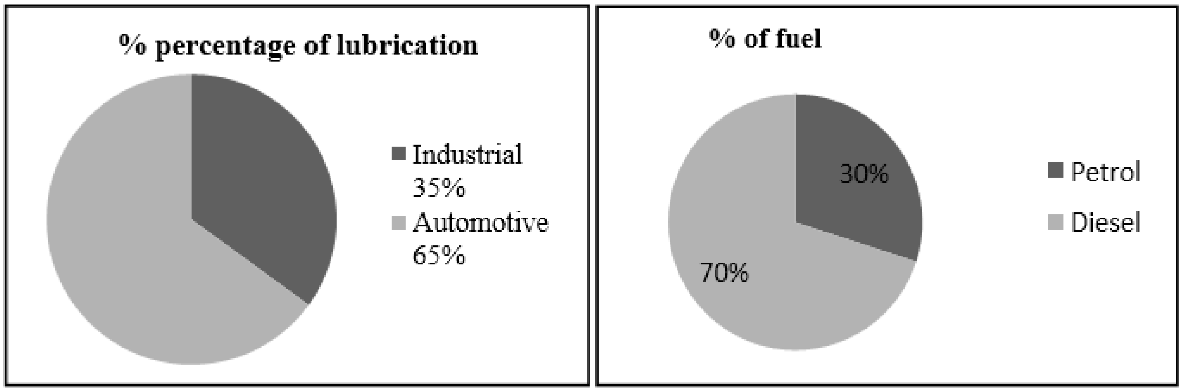

Figure 1: Indian Automotive lubricants market

India is the fifth largest lubricant market globally in volume terms behind the US, China, Russia and Japan. India is a net base oil deficit market and many additives used in lubricants are mostly imported. Volume consumption of lubricants in India has consistently declined over past few years as a result of improving lubricant and engine quality. Following figure 1 shows the market condition of the Indian Automotive Lubricants

Lubricant companies in India [6]

The India auto lubricant manufacturer produces classified into two types:

Table 2: Public sector units of engine oil

Company Name	Brand Name	Major Product
Indian Oil Corporation Ltd.	SERVO	Servo super mg, Maruti genuine oil, Servo superior xee
Bhara Petroleum Corporation Ltd.	MAK	MAK classic, MAK supreme, MAK ultima
Hidustan Petroleum Corporation Ltd.	HP	HP extra super motor oil, HP SGX, HP cruise classic

Table 3: Private sector units of engine oil

Company Name	Brand Name	Major Product
Castrol India Ltd.	Castrol	Castrol GTX , Castrol EDGE, Castrol Magnatec
Gulf Oil Corporation Ltd.	Gulf	Gulf Formula GX, Gulf MAX Supreme, Gulf Multi GTS
Shell oil corporation Ltd	Shell	Shell helix HX3, Shell helix HX5 , Shell helix HX7

The oil PSUs (IOC, HPCL and BPCL) along with Castrol control ~80% of the market, with 15 other players competing for the remaining pie. IOC is the market leader in the overall lubricants industry in PSU.

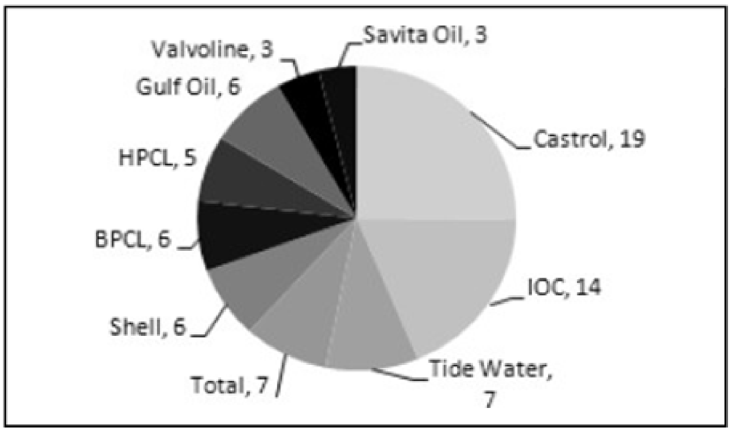

Figure 2: % of consumption in Indian market

Engine oil designation and standards [7]

Lubricant characteristics and performances are managed by standard or industrial organization as American Petroleum Institute (API), Association des Constructeurs Européensd' Automobiles (ACEA), Japanese Automotive Standards Organization (JASO) and International Lubricant Standardization and Approval Committee (ILSAC). Each standard defines technical requirements as physical properties, engine tests results and other various criteria. The Society of Automotive Engineers (SAE) established a viscosity grading system for engine oils. According to the SAE viscosity grading system all engine oils are divided into two classes: monograde and multigrade [8]:

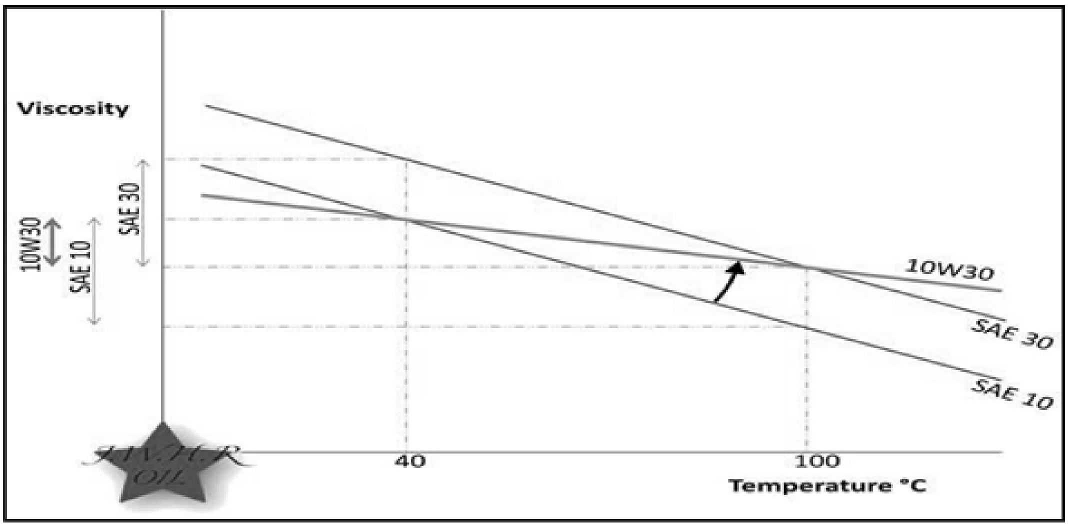

Figure 3: Monograde V/S multigrade oil

3. Characterisation of different engine oil

There are number of manufacturers with number of brands available in the market with specific properties. The selection of the lubricants for the particular engine is dependent on the number of factors [13] [15].

3.1 Compare kinematic viscosity at 40˚C and 100˚C for different brand

Kinematic viscosity (cSt)Figure 4 is showing the relationship between different viscosities. It is observed that at the low temperature test (at 40˚C), the viscosity variation is almost 60% but as the oil warms up, they get very close to each other. Notice that the lines do not cross even though there are only two data points, the viscosity of oil has a linear behaviour so the oil that you select is going to be important from that standpoint.

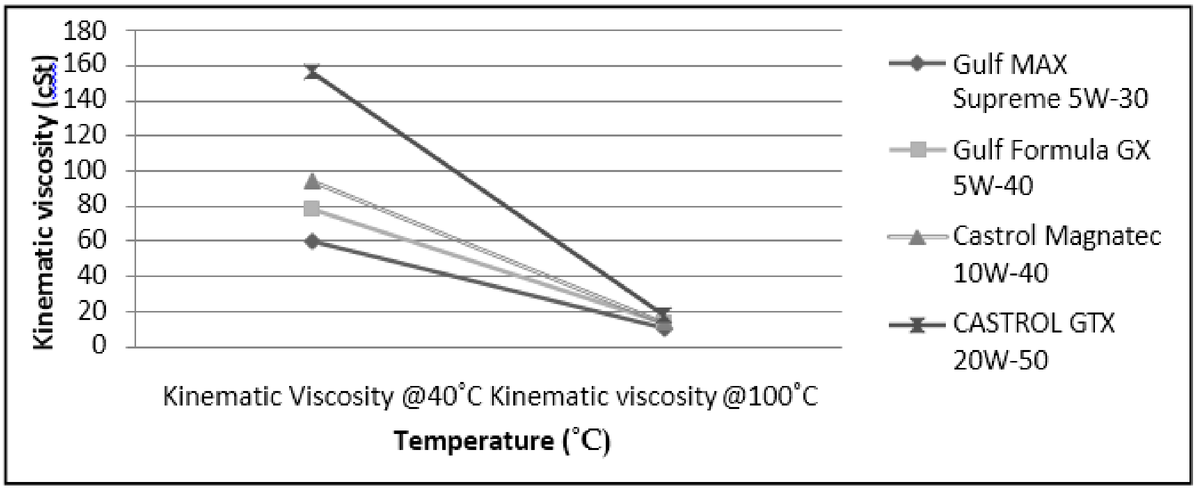

Figure 4: Oil viscosity v/s Temperature for different brand

3.2 Compare various properties of 20W-50 with 10W-30 engine oil

Kinematic Viscosity (D 445)

The kinematic viscosity determines the value at which the fluid can flow. With the higher grade of engine oils (20W-50), the value of kinematic viscosity is 62% to 77% higher because the engine oil is thick hence more internal deformation and shear is present. This graph shows that there is a little difference in kinematic viscosity for all types of products [9].

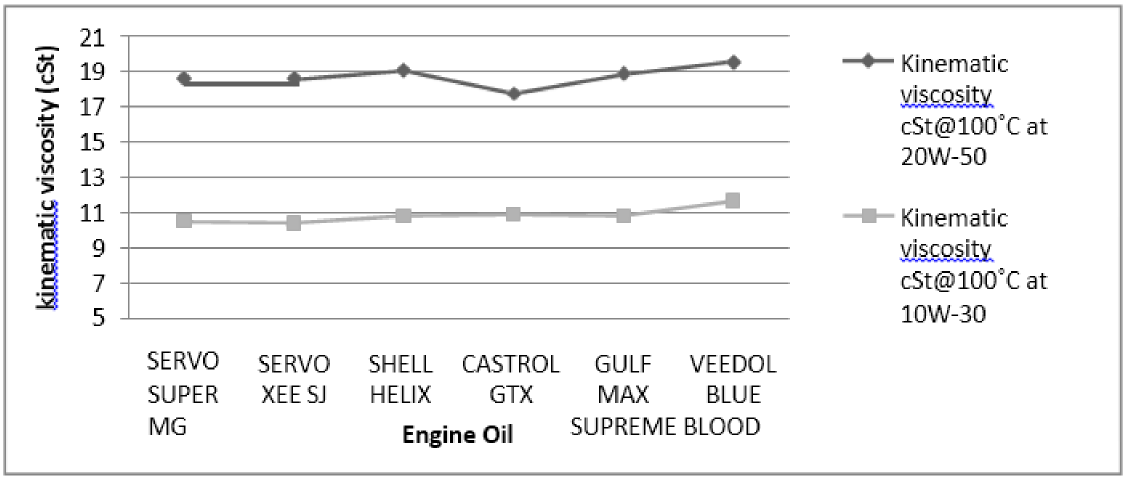

Figure 5: Kinematic Viscosity v/s Engine oil for different grade

Flash point (D 92)

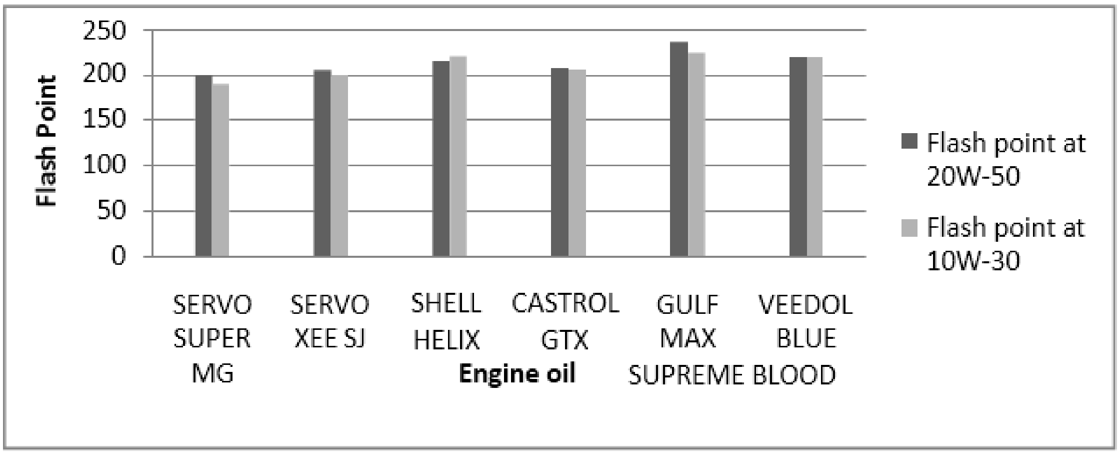

Figure 6: Flash Point v/s Engine oil for different grade

Figure 6 shows that there is only 0% to 5% change in value of flash point due to change the grade of engine oil. A higher FP is better for engine oil. Servo, Castrol and Shell have almost similar values while Gulf and Veedol have a higher flash point at the same grade of engine oil. Gulf has the highest flash point value because it is Saudi Arabian product, so the engine oil prefers more value of flash point for the particular grade.

Pour Point (ASTM D-97)

Pour point From the figure 7, it is observed that the value of the pour point for a higher grade of engine oil (20W50) is 12% to 55% lower as compare to the low grade of engine oil (10W30). The lower a lubricant's pour point better protection it provides in low temperature service.

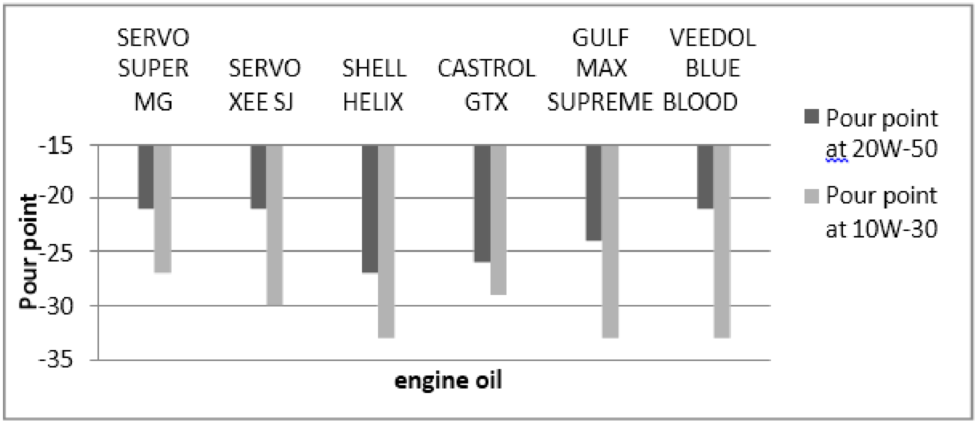

Figure 7: Pour point v/s Engine oil for different grade

Viscosity Index (VI) (D 2270)

It is observed from figure 8 that lower grade of engine oils have 2% to 12% higher value of viscosity index compare to higher grade of engine oils because lower grade of engine oil is suitable for cold countries where the temperature difference is quite more. A low VI means a relatively large viscosity change with temperature and a high VI denotes a smaller change of viscosity with temperature [10].

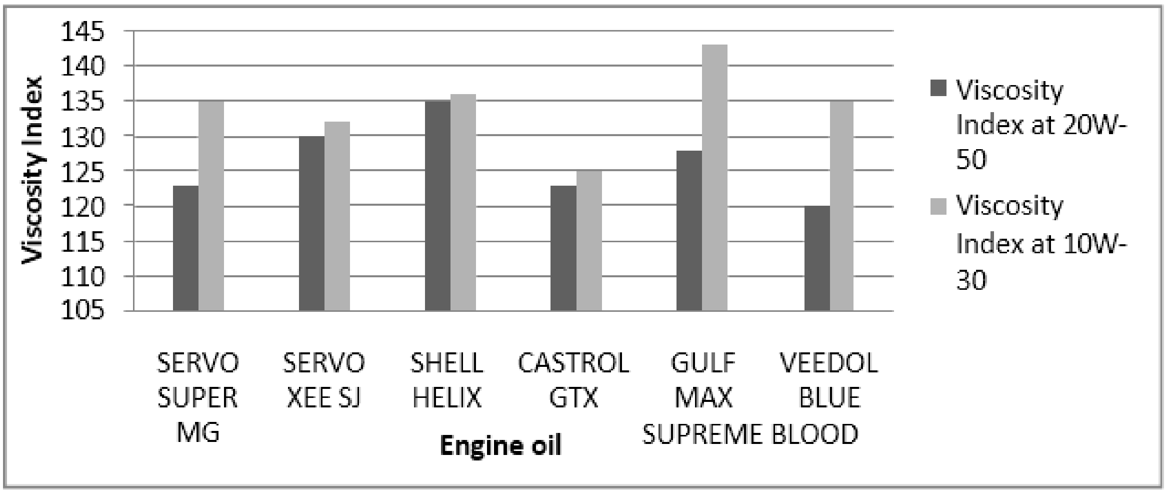

Figure 8: Viscosity Index v/s Engine oil for different grade

Total Base Number (ASTM D-2896)

Total Base Number Total Base Number (TBN) is the measurement of a lubricant's reserve alkalinity, which aids in the control of acids formed during the combustion process. Generally engine oils have a minimum TBN of about 6. Dangerously low would be 2. Oils rated for diesel have a higher TBN, around 10, because they contain more detergents.

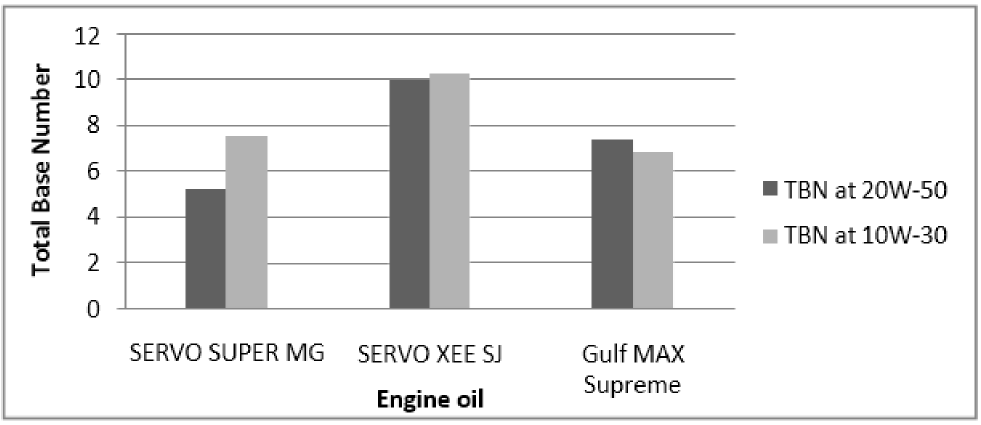

Figure 9: Total base numbers v/s Engine oil for different grade

The high TBN of Motor Oil allows it to effectively combat wear-causing contaminants and acids, providing superior protection and performance over extended drain intervals.

CASE STUDY

Engine oil 5W-30 (used in modern car) and 20W-40 (used in before 2008 model) used in maruti gasoline passenger car.

As per maruti user's manual, newer vehicles will specify lower viscosity oils such as 5W-30 while older vehicles will specify higher viscosity oils such as 20W-50. This is because today's engines are built with tighter bearing clearances to take advantage of the fuel economy benefits of lower viscosity oils. It is not really a good idea to use thicker oil in one of these engines because it will disrupt the oil flow characteristics of the engine and may create excessively high oil pressure. In an older engine that was designed with larger bearing clearances, it is appropriate and recommended to use a thicker oil to maintain proper oil pressure and provide adequate bearing film thickness.

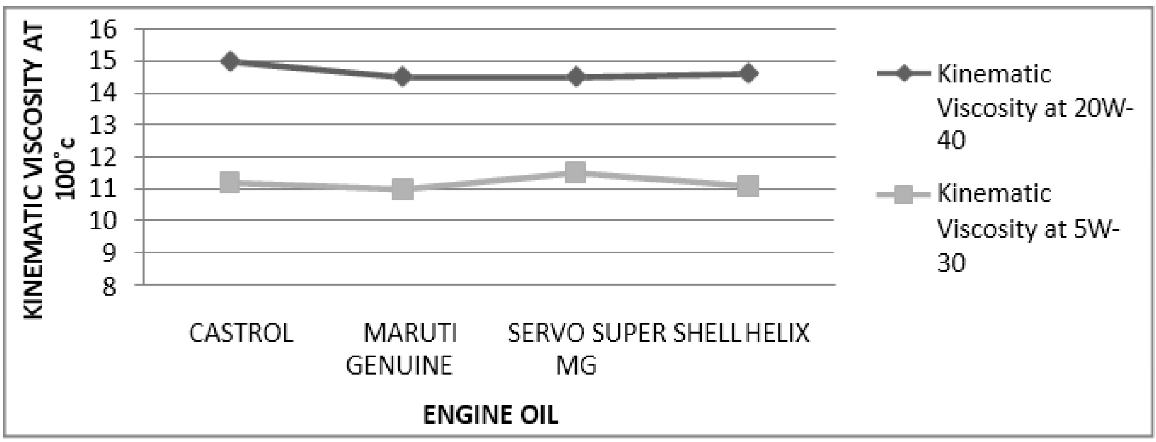

Figure 10: kinematic viscosity v/s Engine oil for 20W-40 and 5W-30

Figure 10 shows the comparison of kinematic viscosity at 20W-40 and 5W-30 grade engine oil. For both the grades of engine oil, it will follow the same trend and there is 3.5% to 4.5% variation in values which is found from the graph it indicates that we can use any brand of engine oil.

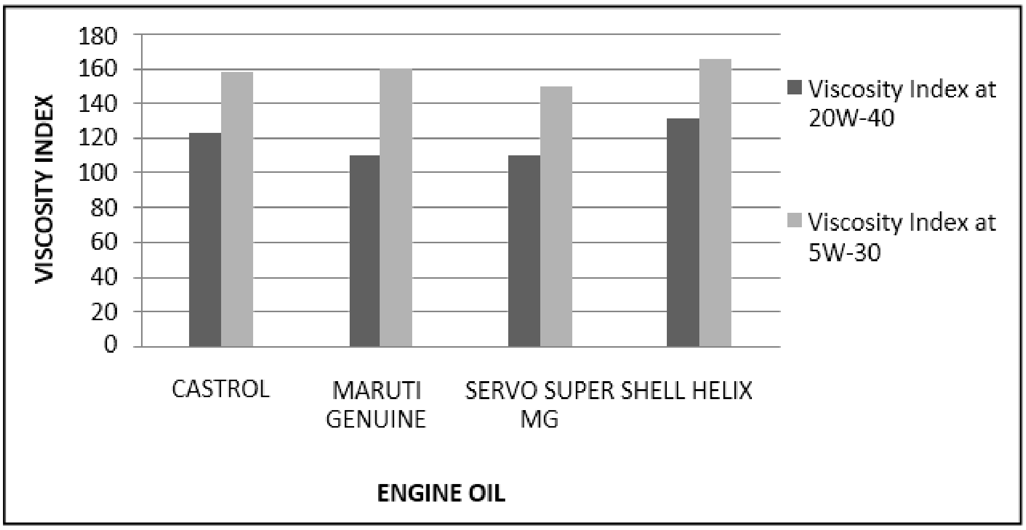

Figure 11: Viscosity Index v/s Engine oil for 20W-40 and 5W-30

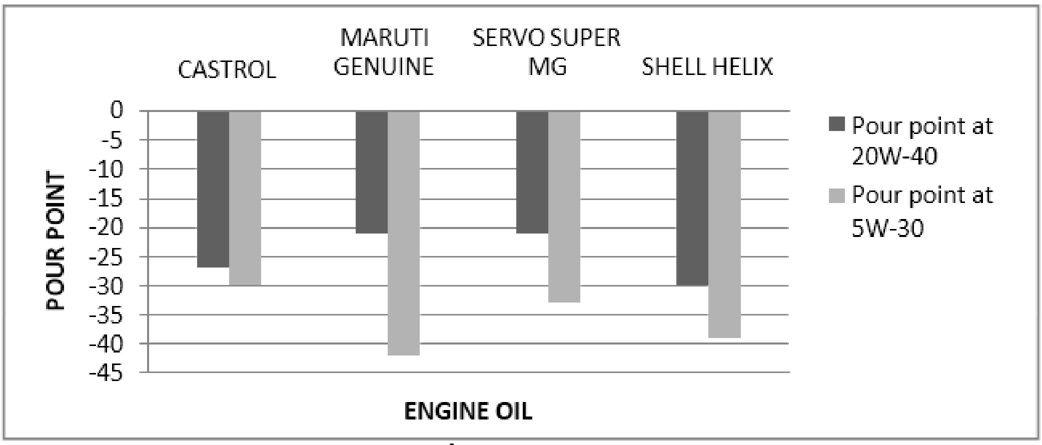

Figure 12: Pour Point v/s Engine oil for 20W-40 and 5W-30

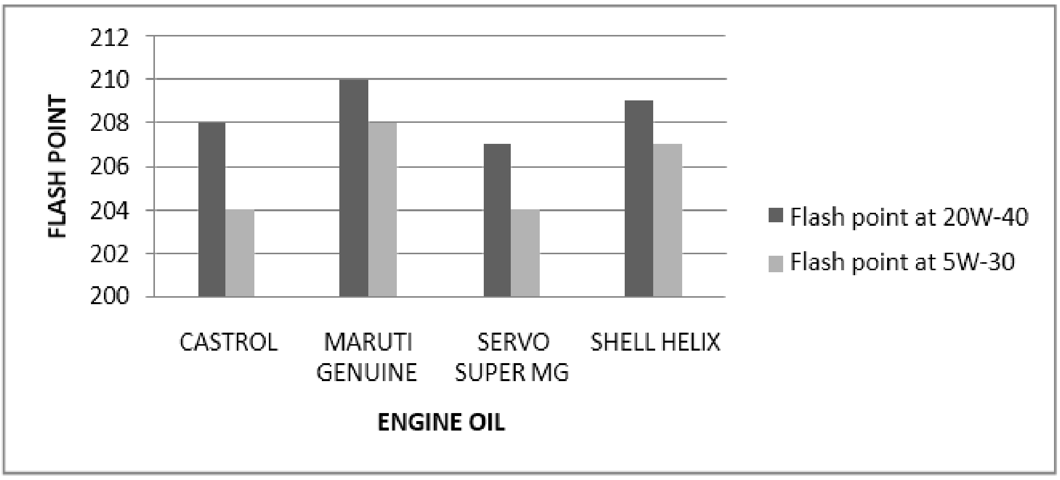

Figure 13: Flash Point v/s engine oil for 20W-40 and 5W-30

Synthetics engine oil like 5W-30 grade can be used in wide temperature extreme, not just cold weather but heat as well. So due to wide temperature range

- High viscosity index (like 25% to 45%) improver is required to take a small change in kinematic viscosity of engine oil.
- Lower value of pour point is required to take the benefits of cold condition in winter hence the variation is 5% to 20%. Maruti genuine engine oil with servo brand has the lowest value of pour point.
- The figure 13 shows that there is only 1.5% to 2% variation in flash point and it is due to change in grade only.

What type of motor oil to use, it is actually recommended by the owner's manual of the vehicle and it depends on the climate condition and clearance between the components of the engine. Maruti engine generally prefers IOC Company with servo brand. Servo has provided special name Maruti Genuine Engine Oil with 20W-40 and 5W-30 to avoid confusion with different brands and grades of engine oil. They also prefer Castrol and Shell lubrications as per the availability of engine oil with same grade.

CONCLUSION

Using the correct oil keeps your engine running smoothly and gives better performance. The selection of appropriate engine oil depends upon load, speed and driving condition of a car. However, their applicability strongly depends on the local availability and atmospheric condition.
Some of the concluding remarks based on the comparisons are given below:

- Thinner oils have a water-like consistency and pour more easily at low temperature hence it can be used for lower starting and/or operating temperature, lighter the load and faster the operating speed of an engine.
- Thick engine oil is better for maintaining film strength and oil pressure at high temperatures and loads hence it can be used where higher the starting or operating temperature, higher load and slower the operating speed of an engine.
- With the higher grade of engine oils like 20W-50, the value of kinematic viscosity 62% to 77% higher as compare to 10W-30 because the engine oil is thick hence more internal deformation and shear is present.
- It is observed that lower grade of engine oils like 10W-30 have 2% to 12% higher value of viscosity index as compare to 20W-50 because they are suitable for cold countries where wide temperature difference is there[9].

- A higher FP is better for engine oil. About 400ºF (204˚C) is the lowest acceptable FP for new oil. With the change of grade, the variation in flash point is only 0 % to 5% because it is the temperature which effect at the operating condition of an engine.
- The lower a lubricant's pour point, the better protection it provides in low temperature service. With the change of grade, the variation is quite more (depends on the grade) because it depends on the lowest temperature of an engine which depends on the climate condition of the particular region.
- The higher a motor oil's TBN, the more effective it is in suspending wear-causing contaminants and reducing the corrosive effects of acids over an extended period of time. Generally engine oils have a minimum TBN of about 6. Dangerously low would be 2.

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