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EVALUATING EMISSION CHARACTERISTICS FROM VEHICLES AT ROAD TEST WITH DIFFERENT OPERATING CONDITIONS

تقييم خصائص إنبعاثات المركبات عند عوامل التشغيل المختلفة باستخدام أسلوب إختبار الطريق

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ملخص

فى هذه الدراسة تم تقديم أسلوب جديد لتقييم تأثير بعض عوامل التشغيل المختلفة على إنبعاثات محركات المركبات من خلال إختبارات الطريق. يتميز الأسلوب الجديد باستخدام معامل واحد للتعبير عن التأثير المكافئ لمجموعة الملوثات الصادرة عن المحرك. تعتمد هذه الطريقة على قياس نسب مكونات العادم ثم حساب التأثير المكافئ لها باستخدام التأثير النسبى لكل ملوث بالنسبة لغاز أول أكسيد الكربون. لهذا الغرض تم إجراء مجموعة من القياسات على مركبة معينة لتفادى الاختلاف الحادث من مركبة إلى أخرى حيث جهزت تلك المركبة بجهاز لقياس مكونات العادم (أول أكسيد الكربون والهيدو كربونات الغير محترقة ، وثانى أكسيد الكربون والأكسجين) بالإضافة إلى قياس سرعة كل من دوران المحرك والمركبة. وتم خلال ذلك اختبار تأثير كلاً من سرعة المركبة و ناقل السرعات بالإضافة إلى تأثير توقيت الشرارة وضغط الإطارات على كل من مكونات العادم والتأثير المكافئ للملوثات . وقد وجد من تحليل النتائج أن نسبةأول أكسيد الكربون والهيدو كربونات الغير محترقة تزداد عند السرعات والاحمال العالية وكذلك عند السرعة الخاملة الأمر الذى يظهر فى صورة ارتفاع فى التأثير النسبى المكافئ لمكونات غازات العادم . كذلك وجد أن استخدام ناقل السرعات بالأسلوب الخاطىء سرعة منخفضة مع نقلة عالية (زيادة الحمل) له تأثير كبير على زيادة التأثير النسبى المكافئ لمكونات العادم. تأخير توقيت الشرارة له تأثير طفيف على التأثير النسبى المكافئ لمكونات غازات العادم ولكن ذلك يقابله انخفاض فى أداء المحرك. كما أثبتت الدراسة أن إنخفاض ضغط الإطارات يزيد من التأثير النسبى المكافئ لمكونات غازات العادم.

ABSTRACT

The present work aims to introduce new method to evaluate emission characteristics from vehicles at road test condition. The technique is characterized by introducing unique factor representing the equivalent toxic effect of the exhaust species. The method depends on measuring the concentration of exhaust species and calculating a factor called *'conditioned ton*" taking into consideration the toxic weight of each species relative to CO. For this purpose, a series of measurements were carried out on a specified vehicle to prevent vehicle-to-vehicle variation. The vehicle is equipped with an infrared gas analyzer and a magnetic pickup transducer to measure the concentration of exhaust constituents and engine rotational speed, respectively. The measurements were conducted at different vehicle speed and gearshift. The effect of spark timing and tire pressure was also examined. The results indicate that most of the carbon monoxide and unburned hydrocarbons emission appears at higher load as well as near the idling speed, which appears clearly as increase in the relative conditioned ton. The gearshift gives significant effect on the emission considering its analogy with engine load. Retarding the spark timing offers significant reduction in relative conditioned ton in spite of poor engine performance is experienced. It was found also that, the lower tire pressure exhibits significant increase in relative conditioned ton.

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1 INTRODUCTION

The atmospheric pollution resulting from the vehicles emission is one of the most important problems facing our world today. Its effect on the environment and public health is significant. In spite of numerous techniques used to control the problem, the pollution level is still increasing. Evaluating the emission resulting from the vehicle is essential for designing long-term strategy to face such phenomena. The effect of engine operating parameters like mixture quality, ignition systems were studied intensively [1-4]. Their work aimed to find out the different sources for emission and the formation procedures. Modern technique was proposed to deal with this problem [5]. It is based on developing an expert system analyzing the engine performance to avoid the possible faults in feeding and ignition systems during the operation. The other trend is focused on studying effect of alternative fuels on the resultant engine emission [6-9]. Most of previous work focused on studying the engines performance and emission in labs and not in their real environment. The present work is oriented towards the evaluation of emission from vehicle engines at real road test condition. The measurements were carried out at different vehicle speeds and gearshifts.

The concentrations of CO_2, CO and unburned HC as well as engine speed are recorded for each vehicle speed and gearshift. The results are plotted and discussed .

2 LAY OUT OF THE EXPERIMENTAL WORK

The experimental work is carried out on a specified vehicle to avoid vehicle-to-vehicle variation, which results from different designs, operating conditions and maintenance processes. According to the previously mentioned plan of study, the road-test mode is employed in the present experimental work. The vehicle is equipped with infrared gas analyzer and rpm pickup transducer. The exhaust gas concentration, engine rotational speed and vehicle speed are recorded during the test. The measurements were taken at steady state speeds for specified gearshift. The vehicle speed is varied up to the maximum allowable limit of the engine rotational speed at each gearshift. The effect of ignition timing and tire pressure is examined also at different vehicle speeds. Their effects are examined at top gearshift, which represents the highest load on the engine. Figure 1 represents the layout of the present experimental work.

Standard setting	
Spark gap	*0.8 mm*
Spark timing	*10 BTDC*
Excess air factor	*0.96*
Engine speed	*764 rpm*
Tire pressure	*2.04 bar*

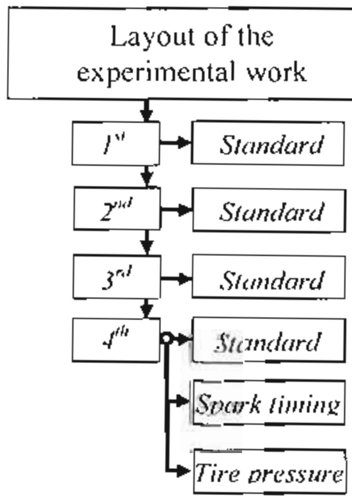

Fig. 1 Layout of the present experimental plan of study

2.1 Vehicle Used in the Present Experimental Work

The vehicle used in the present study is selected to represent statistically the average of manufacturing year for most vehicles in Egypt during the period of study. The vehicle is Peugeot 305 SR model 1983. The engine is spark ignited, four strokes, four in-line cylinders and water cooled. It is transversally mounted with front wheel drive technique. The engine is equipped with manually operated gear box mounted transversally and sharing the same oil sump of the engine. The gear box offers 4 forward speeds and single reversal speed. The technical data for the engine, transmission system and vehicle are represented in tables 1, 2 and 3 respectively.

Table 1 Technical data for the vehicle used in the present experimental study

Vehicle	
Model	Peugeot 305 SR, 1983
Manufacturer	Peugeot
Engine	
No. of cylinders	4 in-line
Type	S-4 SOHC 8 valves total 2 valves per cylinder
Main bearings	5
Bore × stroke	78 mm × 77 mm
Bore / stroke ratio	1.01
Displacement	1472 cc
Compression ratio	9.2
Fuel system	1 Solex carbureotr
Aspiration	Normal
Catalytic Converter	Not installed
Max. output	75 PS (74.0 bhp) (55.2 kW)@6000 rpm
Max. torque	116.0 Nm (86 lb.ft) (11.8 kg.m)@3000 rpm
Coolant	Water
Lubricant capacity	4 litre
Specific output	50.3 bhp/litre
Specific torque	78.8 Nm/litre
Performance	
0-60 mph	13.2 s
Quarter mile	18.5 s
Top speed	156 km/h
Power to weight	78.72 bhp/ton
Chassis	
Engine location	Front
Engine alignment	Transverse
Drive	FWD
Top gear ratio	0.86
Final drive ratio	4.2

Table 2 Technical data for transmission system

Gear selection	Gear ratio
1st forward	0.0737
2nd forward	0.1274
3rd forward	0.1874
4th forward	0.2647
reverse	0.0716

Table 3 Basic dimensions of the vehicle

Item	Dimension
Total length	4240 mm - 166.92 in
width	1630 mm - 64.17 in
height	1400 mm - 55.12 in
Wheel base	2620 mm - 103.14 in
Track front	1380 mm - 54.33 in
Track rear	1330 mm - 52.36 in
Weight	940 kg - 2072.34 lb
Aerodynamics	
Coefficient of drag	0.45

2.2 Instrumentation

Portable version of infrared gas analyzer is used during the experimental work. The gas analyzer is equipped with gas sampling probe to collect the exhaust gas from the muffler. The gas is then filtered and dried before entering the analyzer. Magnetic inductive pickup transducer is used also to measure the engine rotational speed in rpm. It is clipped to any of spark plugs cable in order to capture the spark signal. The sparking rate is then considered as linear proportion to the engine speed. The measuring range and accuracy for measuring instruments are presented also.

3 TEST PROCEDURES

Intensive measurements program were done at different operating condition. The selected vehicle is equipped with previously mentioned measuring instruments. The gas analyzer and its accessories are mounted in the rear seat of the passenger cabinet. Rechargeable power supply and printer are the most important attachment to the analyzer. Gas sampling probe with 3 m long inserted inside the muffler. Its other terminal is connected to the gas analyzer through the window of the rear door. Magnetic inductive transducer is clipped also to the spark plug cable to measure the engine rotational speed. Before starting the measurements, the following precautions are taken into account.

- The engine is warm enough before starting measurements and runs steadily at standard idling configuration.
- All electric accessories like electric fan and radio-cassette are off.
- All the windows of the cabinet are closed except one of the rear windows, which is partially opened (to permit gas sampling connection). This is to keep the drag effect within the standard value.

During the road test, two persons are required to carry out the experimental work. The first is the vehicle driver, which perform the test program with certain sequence. He is responsible to drive the vehidle steadily for enough periods required to obtain steady measurements. The second is the instruments operator, which is responsible to review the test procedure with the driver and observe the output readings. When the signals become steady, the output readings are recorded and next step of vehicle speed is performed. The vehicle speed is increased by step of 10 km/h each time until the maximum allowable speed is obtained for specified gear selection. The higher gear selection is then performed and new range of speed is tested.

4 RESULTS AND DISCUSSION

In the present work, attention is focused on evaluating the emission characteristics at different operating conditions. From the great amount of species exhausted from the engine, CO and unburned HC have an exceptional importance. Beside their great effect on the living organism [10], their presence in the exhaust indicates to poor combustion efficiency. The sources of both HC and CO and the processes lead to their formation were assigned inclusively [11-17]. However, both of the two gasses have different relative toxicities. In order to examine the global effect of exhaust species, an effective mass is calculated called *'conditioned ton'*. This amount represents the total toxic effect of the exhaust gases relative to carbon monoxide. On the other hand, toxic effect of exhaust gases leads to economical losses. The value of economical losses 'L_e'due to air pollution from exhaust gasses of the internal combustion engines can be estimated using the following relation [19].

$$L_e = \phi * R_p * M_p \quad \ldots(1)$$

Where:

ϕ is constant (Takes into consideration the transformation of the equivalent masses of the pollutions into currency).

R_p= factor of the relative danger of pollution over territories of different types. The value of this factor depends on the location and human population in the area under consideration. Its values can be obtained from table in appendix A.

M_p= the annual equivalent mass of pollutants from transportation.

$$M_p = \sum_{i=1}^{n} A_i M_i \quad \ldots(2)$$

n = No. of pollutant gases in the exhaust of the engine.

A_i= coefficient of relative toxicity of each pollutant (condition ton/ton) [19].

M_i= annual mass of each pollutant (ton/year)

The conditioned ton is then used to evaluate the weight of different factors affecting vehicles emission at different operating condition. Relative condition ton (R_t) is then derived to express the relative global toxicity of the exhaust gasses at different operating conditions. It can be represented mathematically as follows.

$$R_t = \frac{\sum A_i M_i}{\left(\sum A_i M_i\right)_{ref}} \quad \ldots(3)$$

Where $M_i = f(CO\%, HC\%)$

In the following, the emission from the vehicle at different road-test condition is presented.

4.1 Road Test at Different Gear Shift

Low traffic density rural road is used to ensure steady state measurements without subjecting any considerable objects. The vehicle is driven steadily at the first gearshift starting 10 km/hr and the measured parameters are then recorded. The speed is increased by step of 10 km/hr up to the maximum allowable engine speed corresponding to the specified gearshift. The vehicle and engine speeds as well as the exhaust species concentration are recorded at each vehicle speed. Transmission is then shifted to the following gearshift and data processing is repeated. The linear relation between the engine and vehicles speed at each gearshift appears clearly in Fig. 2. The slope of each line is function of the final driving ratio between the engine speed and live axle speed. The tire radius is included also in this relation. The linear interpolation representing the vehicle speed with respect to engine speed at each gearshift is given as follows.

$$S_v = C_{gs} \times rpm \quad \text{Km/hr} \qquad \ldots(4)$$

Where C_{gs} is the slope of linear relation. For present vehicle its value varies with gearshift as follows.

gearshift	1st	2nd	3rd	4th
C_{gs}	.00873	.0148	.02075	.0318

The power consumed for driving the vehicle at constant speed depends on rolling resistance and drag force. Neglecting the effect of wind velocity and direction, the driving power is estimated as follows [18].

$$Power(kW) = \left(C_R M_v g + \frac{1}{2}\rho_a C_D A_v S_v^2\right) S_v / 1000 \quad (5)$$

Where C_R is the coefficient of rolling resistance ($0.012 < C_R < 0.015$). The value of this coefficient depends on both road and tire condition. In the present work, C_R=0.014 is considered. The gross weight of the vehicle is 940 kg and the total weight when loading the vehicle by equipments and operators becomes 1155 kg. The second term of the equation represents the drag force applied on the vehicle. It depends on the ambient air density ρ_a, coefficient of drag C_D, the frontal area of the vehicle exposed to the air stream. The vehicle speed S_v is considered as the velocity of relative air stream flowing past the vehicle body. The previous mathematical relation is third order in vehicle speed as plotted in Fig. 3. Equation (5) is applied at each gearshift after replacing the vehicle speed by engine speed [eq. (4)] as shown in Fig. 4. The figure exhibits the graduation in power at each gearshift. The last equation represents the constant load applied on the engine during steady state speeds. The designed rated power generated from the engine is always greater than the maximum driving power. The difference is used to raise the engine performance, specially the ability of engine to accelerate the vehicle speed within short period. In order to estimate the emission level, the exhaust products are measured along the entire range of engine speed and at different gearshift. The results indicate remarkable increase in unburned HC level particularly near the idling condition and at the upper speed limit. This can be attributed to the rich mixture formed at these zones. During the part load, the engine runs at the economical condition, so leaner mixture is employed and the result is lower emission of HC as shown in Fig. 6. Shifting the gear to the upper, leads to increase the unburned HC according to the corresponding increase in engine load as shown in Fig. 4. For a certain vehicle speed, the power required to drive the vehicle becomes constant whatever the gearshift is used, and consequently the rotational speed of driving wheels "N_w" and load torque transmitted by the wheels "T_w" become constants. In this case, the load torque applied on the engine crankshaft depends on the gear ratio for the specified gearshift. The load torque ratio applied on the engine crankshaft at different gearshift is then defined as follows.

$$Load\,torque\,ratio = \frac{(T_e)_{any\,gearshift}}{(T_e)_{first\,gearshift}} = \frac{(N_w N_e)_{any\,gearshift}}{(N_w N_e)_{first\,gearshift}} > 1 \ldots(6)$$

where,

$$\frac{N_w}{N_e} = FGR = Final\ gear\ ratio\ for\ any\ gearshift$$

Using the values of FGR given in table 2, the load torque ratio can be obtained. According to the previous equation, when using higher gearshift, the engine is exposed to higher torque as appears in fig. 5, considering the first gearshift as a base for dimensionless groups. Although the jump from first to final gearshift is practically impossible, a considerable increase in load torque ratio is noticed between the successive gearshifts. From previous analysis one can deduce that the usage of high gearshift at relatively low engine speed leads to increase the load torque on the crankshaft. At this condition, the flame propagation speed becomes high compared to the rate of change of cylinder volume. The result is a remarkable increase in pressure and temperature inside the cylinder. These conditions contribute in increasing the concentration of unburned HC by its different sources. The vaporization and partial burning of lubricant layer is one of these sources. The increase in cylinder pressure leads to a corresponding increase in residual gases inside the clearance volume which in turn helps to increase the unburned HC. The amount of fresh mixture inside the piston crevices increases also with increasing the cylinder pressure. All these factors together prove the relation between the high load applied on the engine and the increase in unburned HC level in the exhaust. The emission of carbon monoxide depends mainly on air to fuel ratio and its distribution inside the cylinder prior and during the combustion. For this reason, it is expected at lean mixture to obtain low concentration of CO considering high degree of homogeneity between the mixture constituents. So the lower concentration of CO is generally accompanied by higher concentration of oxygen as shown in Figs. 7 and 9. The dissociation of carbon dioxide is another source of CO formation. Unlike *UBHC*, this type of reaction is generally very fast, so it approaches the equilibrium state, which is strongly temperature dependent process. This explains the relation between the high load and CO emission. The appearance of CO in exhaust is always on the account of carbon dioxide concentration. The higher the CO the lower the CO_2 as shown in fig. 8. Figure 10 represents the variation of excess air factor along the engine speed at different gearshift. The figure shows that the fuel-air mixture tends to be rich near the top speed, which represents the full load condition. Estimating the relative conditioned ton for the exhaust gases leads to appear great increase in toxic effect at idling speed as well as at high load condition as shown in fig(11). From previous results it is concluded that, the full load condition must be avoided as far as possible. This condition recurrences at the far end of each gearshift speed range.

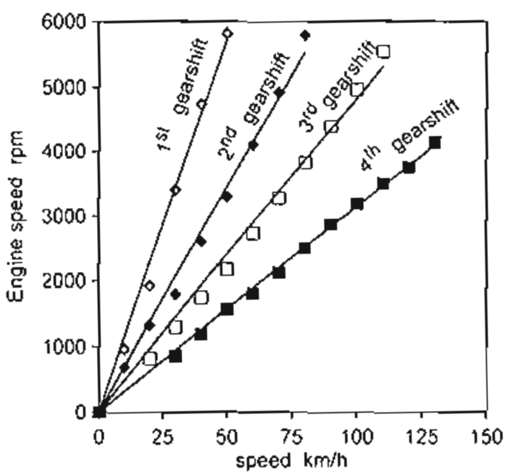

Fig. 2 Relation between engine speed and vehicle speed at different gearshifts.

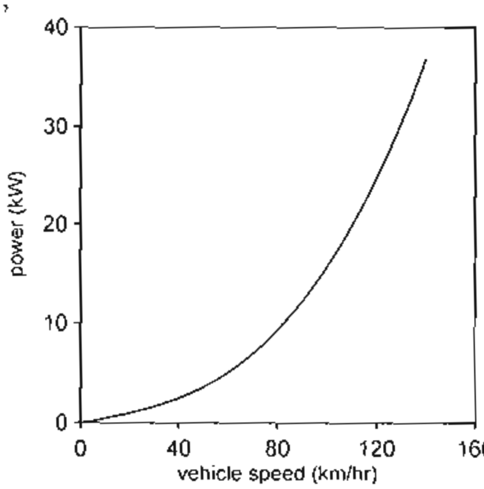

Fig. 3 Road load power vs. vehicle speed

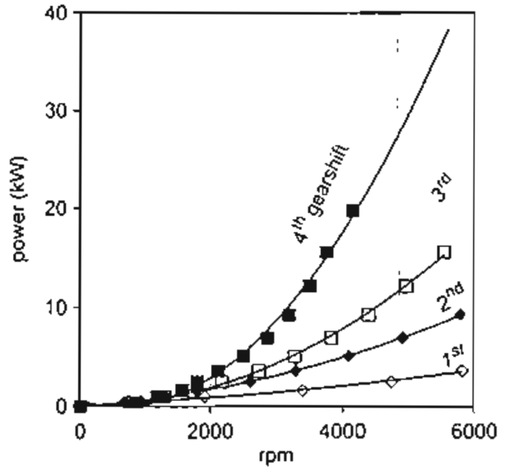

Fig. 4 Road power load vs. engine rpm at different gearshift.

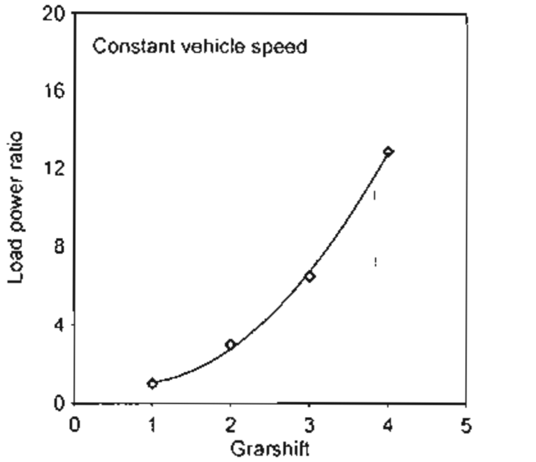

Fig. 5 Load torque ratio with different gearshift at constant vehicle speed.

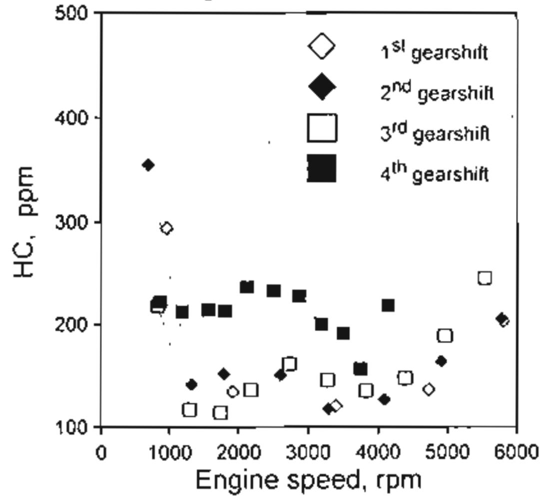

Fig. 6 HC emission along the engine speed range at different gearshift.

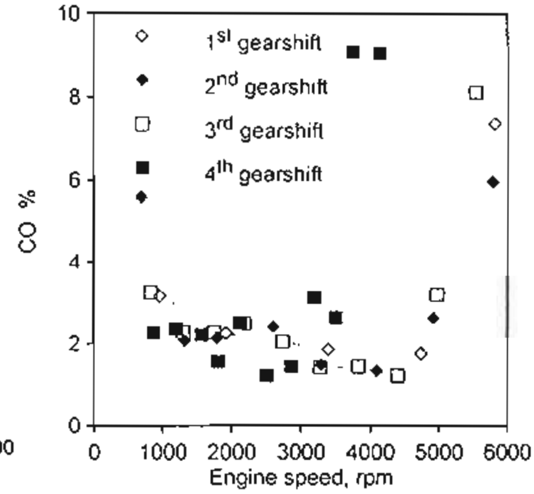

Fig. 7 Emission of carbon monoxide at different engine speed and gearshift.

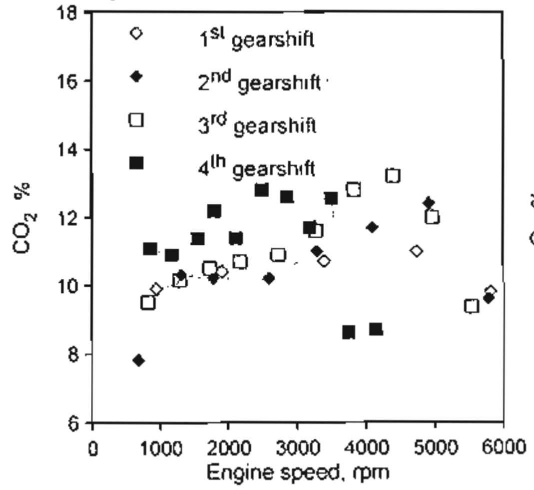

Fig. 8 Emission of carbon monoxide at different engine speed and gearshift.

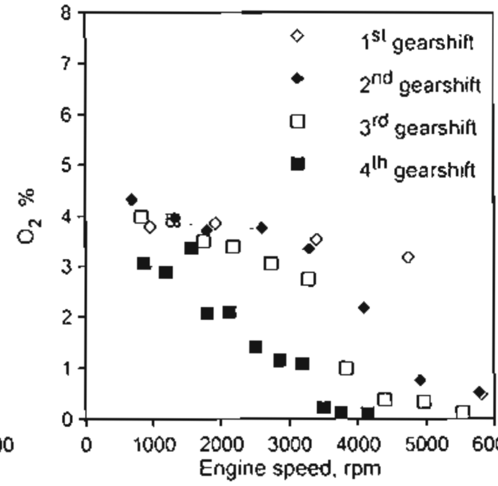

Fig. 9. Oxygen concentration in exhaust at different engine speed and gearshift

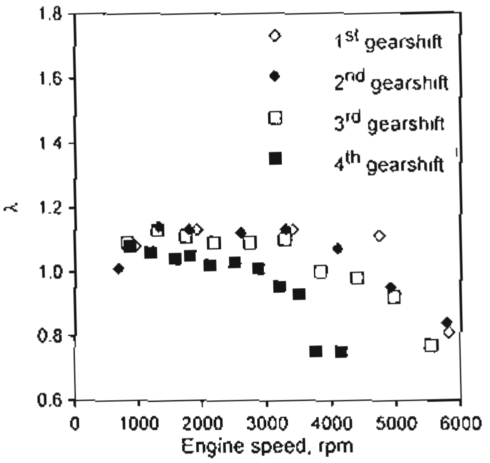

Fig. 10. Variation of excess air factor 'λ' with engine speed and gearshift

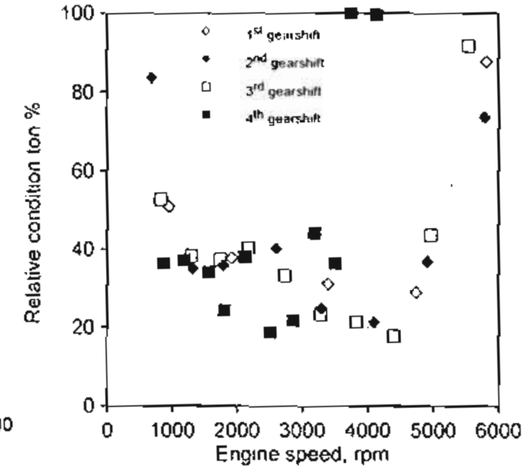

Fig. 11 The effect of engine speed and gearshift on the relative conditioned ton%

4.2 Effect of Spark Timing at Road Test

Spark timing is an important parameter which controls the engine performance and emission. The beginning of combustion as well as pressure and temperature histories during the cycles are affected directly by spark timing and consequently the resultant emissions. The rate of flame propagation depends partly on the thermal condition inside the cylinder at the time of flame initiation. This in turn governs the rate of formation of the product constituents during the primary combustion process. During the acceleration, the spark timing is advanced instantaneously to meet the optimum firing angle corresponding to the engine speed. This timing is divided into two periods as follows.

$$\theta_i = \theta_s + \Delta\theta_d \qquad \ldots(7)$$

Where θ_s is the spark timing at idling condition. Sometimes it is called static ignition timing. Its value is adjusted to optimize the combustion process and to ensure steady state running at idling. The dynamic advancement in firing angle $\Delta\theta_d$ is varied automatically according to the engine speed and load. The sensitivity of dynamic advance to engine speed is due to the centrifugal force issued from two counterparts which rotate with distributor shaft. On the other hand, the effect of engine load appears as a pressure drop in the carburetor throat. Both physical quantities (centrifugal force and pressure drop) are proportional to the second order of engine speed as follows.

$$\Delta\theta_d = f\left(N^2\right) \qquad \ldots(8)$$

or in differential form,

$$d\theta_d = \frac{\partial\theta_d}{\partial N} dN \qquad \ldots(9)$$

Integrating the last equation along the dynamic advancing range, the firing angle is then obtained. The deviation between ignition angle at original setting and that with advanced or retarded setting is derived as follows.

$$\frac{\Delta\theta_i'}{\theta_i} = \frac{\theta_s' - \theta_s}{\theta_s + \left(\dfrac{N^2 - N_i^2}{N_{max}^2 - N_i^2}\right)\Delta\theta_d} \qquad \ldots(10)$$

Where, $\Delta\theta_i'/\theta_i$ is the normalized deviation and θ_s' represents the advanced or retarded ignition angle at idling speed. The graphical representation of the last equation is presented in Fig. 12. It is obvious from previous equation that, the spark timing can be shifted in advance or retard direction by adapting the static ignition timing only, which is adjusted manually. Any misalignment of this angle leads to a corresponding inappropriate timing for the beginning of combustion. In the present work, the effect of spark timing is examined at three different conditions. The first is the original setting which is 10^0 *BTDC*, the second is 9 degree advanced and third condition is 9 degree retarded. Figure 12 shows that at idling speed, the deviation from standard setting is $\pm$ 90 %. As engine speed increases, the deviation decreases until it reaches about $\pm$ 20 % at the maximum speed. This indicates that, great effect is expected at idling speed and the lower range of part load compared with maximum power. This reflects typically on HC emission as shown in Fig. 13. at which the measurements of exhaust concentration exhibits great differences in HC at idling speed. As the engine speed increases, HC becomes less dependent on spark timing. This behavior is interpreted as follows. At retard ignition, the temperature inside the cylinder during the expansion process increases due to the late combustion. This relatively high temperature reduces the depth of quenching zone adjacent to the cylinder wall at which most of HC is formed. At higher speed, the combustion process becomes less sensitive to the ignition timing as mentioned before and the fresh mixture quality becomes the fundamental factor, which govern HC formation. The air to fuel ratio depends mainly on the carburetor configuration and this remain unchanged during the current work. This explains the convergence in HC concentration at the high range of speed and power. The measurements show also the great dependency of CO on the mixture quality rather than the spark timing as represented in Fig. 14. Little disparity in CO concentration is observed when changing the spark timing. It comes from the fact that, for the same engine speed, the throttling valve is not in the same position at different spark timing. In other word, when retarding the spark timing, the engine power is reduced. In order to compensate the drop in power and to return to the same speed, the throttling valve is then partly opened allowing more fuel passing to the engine. Thereby the mixture becomes richer as shown in Fig. 15 and more percentage of CO concentration is formed.

It is important also to indicate that, when retarding the spark timing, the engine experiences high temperature and its ability to accelerate the vehicle drops. This prevents the achievement of high speed range. When applying the previously mentioned analysis of relative conditioned ton to acquire the effect of spark timing on the resultant emission represented by the relative value (present value / reference value), Fig.(16) was obtained. The spark timing at 9 degree advance was taking as reference point

It is clear from Fig.(16) that, the relative toxicity has a little effect with spark timing and this can be referred to lower change of CO with respect to spark timing as mentioned above. It was noticed also that, when retarding the spark timing a remarkable decrease in engine response is experienced and corresponding increase in engine temperature was recorded also.

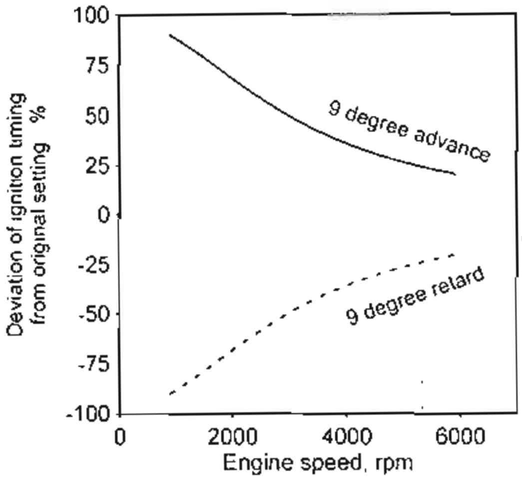

Fig. 12 Deviation of ignition angle.

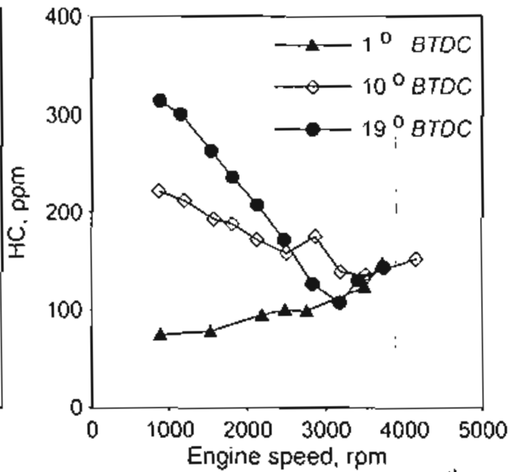

Fig. 13 Effect of spark timing on HC (4th gear)

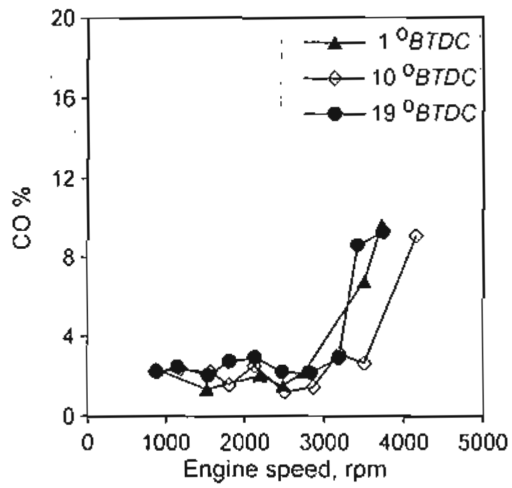

Fig.14. Variation of carbon monoxide with engine speed at different spark timing (4th gear shift).

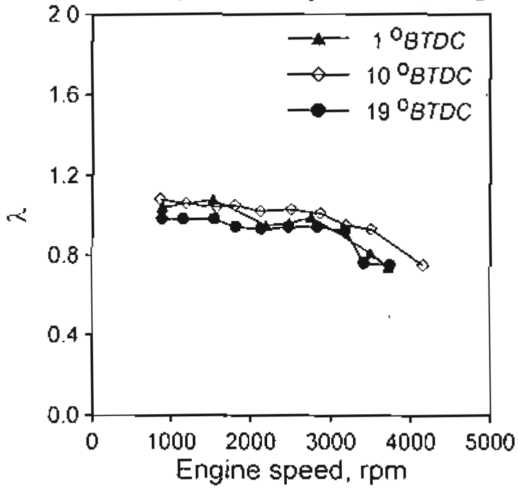

Fig. 15. Effect of spark timing on excess air factor at different spark timing (4th gear shift)

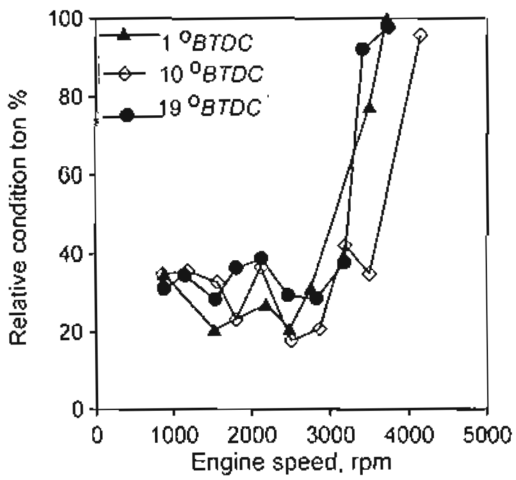

Fig. 16 The effect of spark timing on the relative conditioned ton% (4th gear shift)

4.3 Road Test with Different Tire Pressure

In present work, wide range of tire pressure is used to examine its effect on the vehicle performance and engine emission. Experiments were carried out at selected steady state speeds (60 and 90 km/hr at top gearshift). Low tire pressure is used at the beginning of the test 15 psi (≈ 1 bar). This pressure is considered as a reference point in relative condition ton analysis After each recording cycle, the tire pressure is increased by 5 psi (0.34 bar) up to 35 psi (2.38 bar). When increasing the air pressure inside tires, its effective radius increases also (which is always less than the geometrical radius). On the other hand the contact area with road becomes smaller, a condition that leads to increase the slip between the tire and road. For the same engine speed and gearshift, the vehicle speed depends on the previously mentioned factors as follows.

$$vehicle\ speed \propto \left(\frac{effective\ tire\ radius}{slip\ ratio}\right)$$

It was found that, increasing the air pressure inside tires leads to corresponding increase for the effective tire radius and slip ratio. The resultant effect is insensible at 60 km/hr as shown in Fig. 17. At 90 km/hr, the increase in slip is slightly higher than the effective tire radius. This appears as little increase in engine speed at the higher pressure as shown in the figure. Taking into consideration that the tire pressure is increased above the adjusted values during the motion because of the heat absorbed due to friction between tire and road. On the contrary, the air pressure inside the tires exhibits clear effect on exhaust emission. At low tire pressure, the contact area with road increases and the friction torque increases correspondingly. When the engine speed is almost constant as shown previously, the power consumed to overcome the friction increases according to the following relation.

$$power_f = T_f \times \varpi \qquad ...(11)$$

The increase in engine power as a response of increasing the friction load, comes from burning additional amount of fuel. This in turn lowers the excess air factor at relatively low tire pressure by the same ratio of fuel increased. When increasing the air pressure inside the tire, the rolling resistance and the friction loss decreases and excess air factor increased as shown in Fig. 18. The figure shows narrow band of excess air variation approximately 2% along the range of tire pressure variation. Translation to the higher vehicle speed (90 km/hr) requires wider position for throttling valve. At that position, the engine is in the higher level of part load but not in full load condition. According to the carburetor performance the excess air factor must increased as shown in fig. 18. The variation in excess air factor represents the major factor controlling CO emission. Other minor factor comes from the dissociation of carbon dioxide post the flame. The rate of dissociation depends on the temperature and pressure inside the cylinder. As the friction load increases, the cylinder pressure and temperature increases accordingly. These factors interpret the increase in CO level with reducing the tire pressure as shown in Fig. 19. The results show an average increase in CO concentration by 27 % when reducing the tire pressure by 50%. This significant variation in CO is obtained because the excess air factor lies in the rich zone. For hydrocarbons fuel, the presence of CO in the exhaust leads to reduce the concentration of carbon dioxide. This behavior appears clearly in Fig. 20. The unburned HC issued from multiple sources like flame quenching, oil layer, piston crevices and partial burning or poor combustion quality. The narrow range in excess air factor shows little effect on HC formation as shown in Fig. 21. Examining the effect of tire pressure on the relative condition ton, it was found that, the increase in tire pressure leads to reduce the relative toxicity. This can be attributed to the low variation in HC production with respect to tire pressure. For this reason, the trend of relative toxicity is approximately as CO as shown in Fig [22].

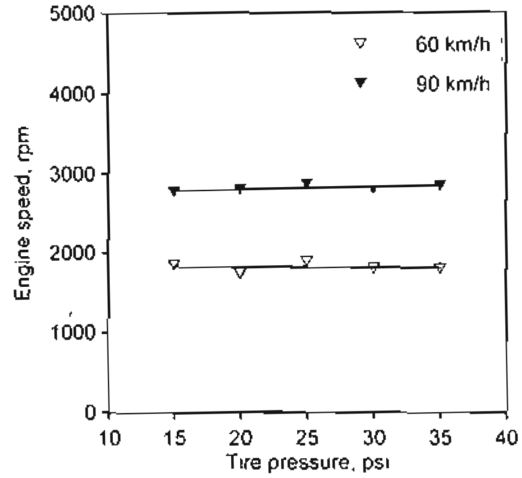

Fig. 17 Variation of engine speed with tire pressure at selected vehicle speeds.

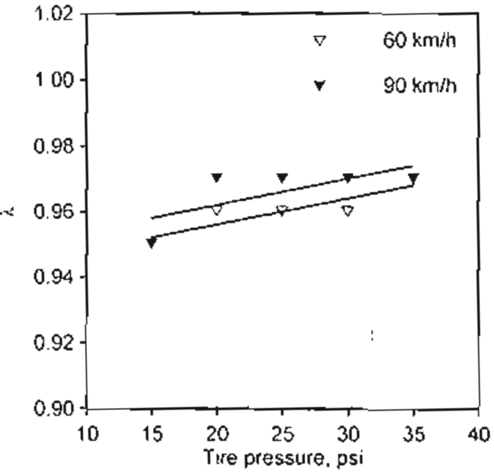

Fig. 18 Effect of tire pressure on excess air factor.

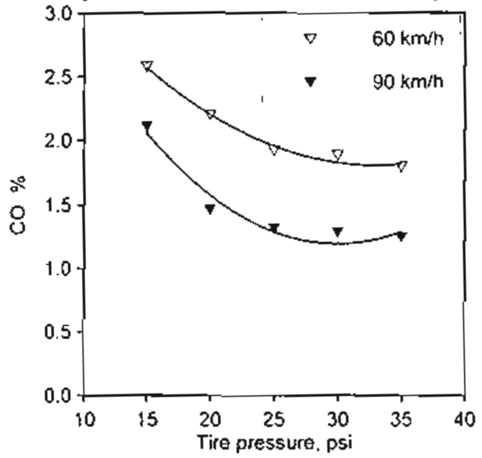

Fig. 19 Variation of carbon monoxide with tire pressure.

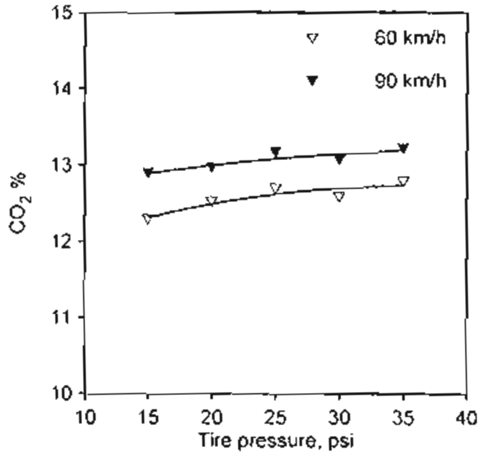

Fig. 20 Variation of carbon dioxide with tire pressure.

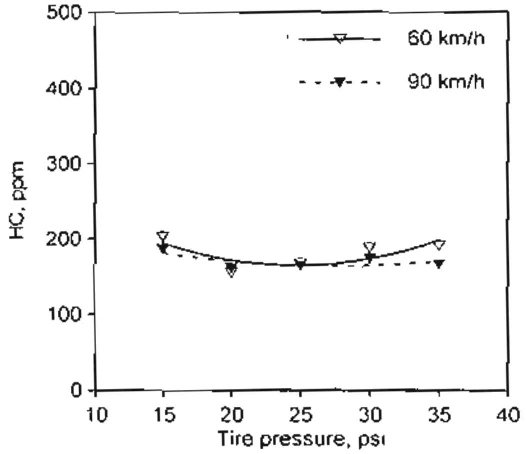

Fig. 21 HC emission at different tire pressure

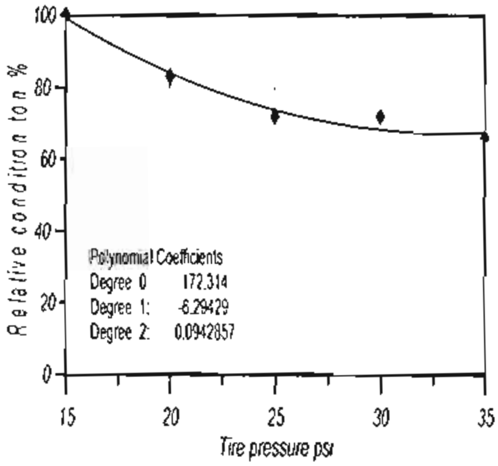

Fig.22 The effect of tire pressure on the relative conditioned ton (60 km/hr)

5 CONCLUSSION

In this work, new manipulation for evaluating emission characteristics from vehicles is presented. The present approach depends on estimating the global toxicity of the exhaust gases. For this purpose, series of experiments were carried out at road test condition. The effect of gearshift, spark timing and tire pressure are examined. The emission level as well as engine performance are recorded and evaluated. The results show clear effect of operating condition on the emission characteristics as follows.

- It is important to avoid running the engine at the upper limit of power as far as possible at which the emission of unburned HC and CO reaches undesired level. This reflects on the increase in relative conditioned ton.
- Using high gearshift at relatively low speeds leads to increase the load torque applied on the engine crankshaft. As an instantaneous response, the pressure inside the cylinder increases and creates the suitable environment to produce toxic emission with high rate represented by increasing relative conditioned ton.
- Retarding spark timing leads to reduce HC emission particularly at low range of power.
- The relative toxicity has a little effect with spark timing and this can be referred to lower change of CO with respect to spark timing as mentioned above. It was noticed also that, when retarding the spark timing a remarkable decrease in engine response is experienced and corresponding increase in engine temperature was recorded also.
- The results show the importance of adjusting the tire pressure. It was found that, when reducing the tire pressure by 50% corresponding increase in relative toxicity level is obtained by 27%, takes the same trend as CO. As HC emission shows little effect.

Nomenclature

symbol	definition	unit
A_v	Frontal area of vehicle	m^2
A/F	Air to fuel ratio by mass	
C_R	Coefficient of rolling resistance	
C_D	Drag coefficient	
g	Acceleration due to gravity	m/s^2
M_v	Mass of vehicle	kg
N_e	Engine rotational speed	rpm
N_w	Driving wheel rotational speed	rpm
R_T	Relative conditioned ton	
S_v	Vehicle speed	Km/hr
T	Torque	N.m
T_e	Load torque applied on engine crankshaft	N.m
T_w	Load torque transmitted by driving wheel axle	N.m

GREEK SYMBOLS

$\Delta\theta_d$	Combustion delay period	degree
θ_{eoc}	Crank angle at end of combustion process	degree
ρ_a	Density of atmospheric air	Kg/m^3
θ_i	Crank angle at the beginning of spark	degree
λ	Excess air factor $(A/F)/(A/F)_{ST}$	
ϖ	Engine angular speed $= 6N$	degree/s
ϕ	Constant (Take into consideration the transformation of the equivalent masses of the pollutions into currency).	

SUBSCRIPT/SUPERSCRIPT

f	Friction
ST	Stoichiometric

ABBRIVIATIONS

BDC	Bottom Dead Center
BTDC	Before Top Dead Center
FWD	Forward Wheel Drive
FGR	Final driving gear ratio
rpm	Revolution per minute
SOHC	Single Overhead Camshaft
TDC	Top Dead Center

UBHC	Unburned Hydrocarbons
R_P	Factor of the relative danger of pollution over territories of different types.
M_P	The annual equivalent mass of pollutants from transportation
A_i	Coefficient of relative toxicity of each pollutant (condition ton/ton)
M_i	Annual mass of each pollutant (ton/year)
R_T	The relative global toxicity of the exhaust gasses

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APPENDIX A

TYPICAL VALUES OF THE COFFICIENT Ai FOR VARIOUS POLLUTANT AND DUST IN THE ATMOSPHERIC AIR [19]

Pollutant, Dust	Condition ton/ton
Carbon monoxide	1
Sulphure gas	16.5
Hdrogen Sulphide	41.1
Sulfuric Acid	49
Nitric Oxides	42.1
Unburned Hydrocarbon	1.5
Solid particulate in exhaust gases of petrol engines using	
Unleaded gasoline	300
Leaded gasoline	500
Solide particulate from diesel engines	200

VALUES OF THE FACTOR R_P [19]

Type of territory	Value of Rp
Health resort, Santeria, preserve, client	10
suburban rest zones, gardens	8
populated area with density of population Np (1000 person/Hectare)	(0.1) NP
industrial enterprises	
forests	
1st group	0.2
2nd group	0.1
3rd group	0.025
Arable land	0.15
fruit gardens	0.5

Note : For irrigated arable land value in table is divided by 2