AN INFLUENCE OF THE INTERNAL COMBUSTION ENGINE CHARACTERISTICS UPON THE TRAFFIC SAFETY IN THE REGIMES OF OVERTAKING

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1. INTRODUCTION

The intensive progress of traffic, as a logical consequence of the technological progress of the society in general, produces series of undesired effects upon the environment: the increase in the number of traffic accidents with fatal consequences, the pollution of the environment as a result of the exhaust gases of engines, the noise, non – recycled materials of parts of transport means, complete with a transformation of natural surfaces into asphalt and concrete surfaces of traffic roads, parking space and the like. As regards the traffic safety, a particular influence is shown by: the increase in the number of transport means (vehicles), the improvement of their performances (maximum speed, acceleration), and also, by the inadequate development of the road network based on the traffic capacity and the way of traffic regulation. Considering the fact that the influential factors in this area are connected to the man – vehicle – environment system, what is going to be analyzed in this paper is the influences of vehicles, that is, of their drive units, although the other two factors cannot be completely excluded because of the evident interactions.

A driver performs the overtaking manoeuvre on the basis of his/her own evaluation of the position and speed of not only his/her own vehicle but of the other vehicles passing by as well, incessantly endeavouring to drive the vehicle along the optimal distance with the maximum acceleration, so that he could perform the operation itself in the shortest amount of time. The vehicle acceleration is defined through the tractive balance equation and it represents the most significant parameter of the vehicle longitudinal dynamics [4]:

$$a = \frac{1}{m\delta}\left(F_O - \sum R\right)$$

where: m represents a vehicle mass, δ – the coefficient of a participation of a vehicle's rotating masses, F_o – a driving (propelling) force, $\sum R$ – the sum of the tractive resistance.

A driving force of a vehicle's driving wheels is defined through the characteristics of the drive unit (T_e) and a vehicle transmission ratio ($i_m i_o$) according to the following equation [4]:

$$F_O = \frac{T_e i_m i_o \eta}{r_d}$$

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The following symbols are used in the previously given equation : T_e – the torque, i_m and i_o – the transmission ratio and the rear axle ratio, η – the transmission efficiency and r_d –the rolling radius. The foregoing equations practically define the influence of the drive unit's characteristics (in the shape of the torque's curve) upon the vehicle's acceleration values, and therefore, upon the overtaking vehicle's manoeuvre as well.

2. GENERAL CHARACTERISTICS OF OVERTAKING

Overtaking practically represents going round a vehicle along the same carriageway at a certain speed. Considering the fact that this kind of a manoeuvre entails an alternation of the carriageway, the risk of the possible crash with other vehicles moving along the carriageway where this overtaking is taking place – is significantly being increased. Frontal or back impacts usually occur in those situations and their consequences are the most serious ones.

Characteristics of the overtaking process differ depending on whether it is being done along the roads with one – way or two – way traffic. These two cases are schematically presented in the Figure 1.

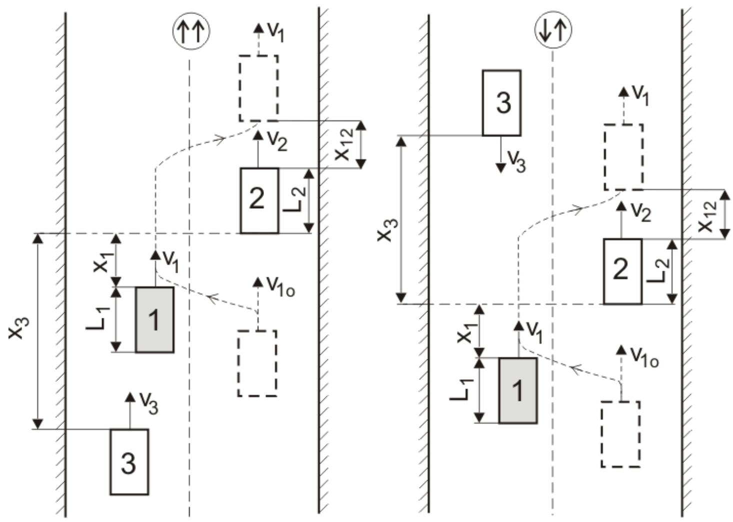

Figure 1.

According to the chart given in the Figure 1, the vehicle 1 is overtaking the vehicle 2, and during this process it is supposed to avoid a possible crash into the vehicle 3 moving in the same direction, in the first case, and in the second case, it is moving in the opposite direction. The vehicles' speed values are the following: v_1, v_2 i v_3, and the distances between the foregoing vehicles are defined by the following values: of the x_1 and x_3 frontal areas of the vehicles 1 and 3 of the rear side of the vehicle 2. The length values of the vehicles 1 and 2 are: L_1 and L_2. The overtaking process is being terminated at the moment when the rear of the vehicle 1 occurs in front of the front part of the vehicle 2.

Monitoring the positions of particular vehicles during their motion can be realized by displaying their distances which have been illustrated through the diagrams presented in the Figures 2 and 3. Because of the assumption relating to vehicles going at a constant speed at the initial moment, the law referring to the alternation of a distance is linear. The characteristic position presented in the diagram is a point *b* which represents the moment (time t_b), when the frontal areas of the vehicles 2 and 3 are found going in the same direction (S_2=S_3=S_b). It is evident that the vehicle 1 is supposed to terminate the overtaking process before that particular moment. Variants in relation to the initial speed of a vehicle that can have the following values: $v_{1o}= v_2$ i $v_{1o}> v_2$ – are also presented in these diagrams. In the first case, overtaking is performed after a vehicle has previously been going in a weaving lane and it requires a vehicle to be accelerated. In the second case, if there is enough difference between the vehicles' speeds: $v_1 - v_2$, the overtaking process can be performed even at a constant speed v_1; on the contrary the vehicle has to be accelerated. The alternation of a distance of the vehicle 1 during the acceleration has been displayed in the diagrams by using a curve S_1*(accel.)*. If the overtaking process is to be performed in a safe manner, what is necessary is to realize a vehicle's acceleration values according to which the distance of the vehicle 1 at the moment t_b is: $S_1(t_b) \geq S_b$, which represents the basic criterion taken into account within the analysis of factors influencing the overtakaking process itself.

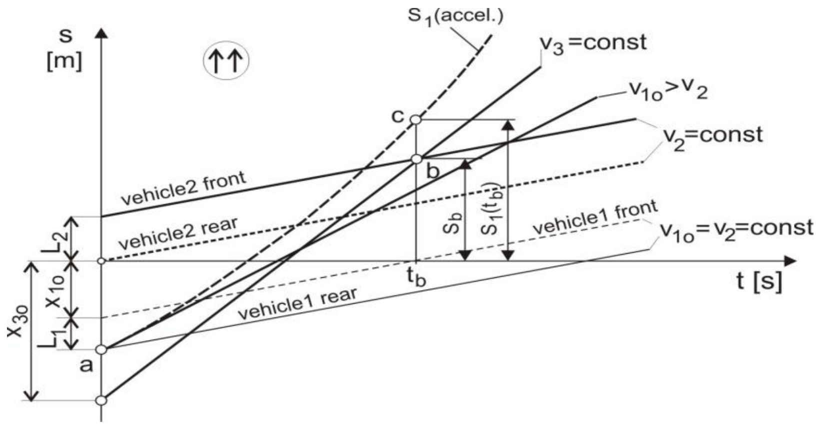

Figure 2.

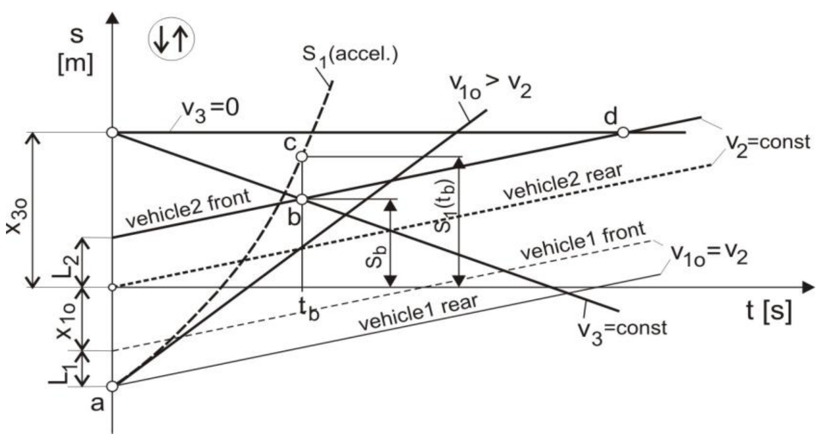

Figure 3.

While defining the simulation model the following assumptions have been made:

- what is being neglected is the carriageway alternation manoeuvre of the vehicle 1 – at the initial moment the vehicle 1 is going along the carriageway used for overtaking,
- it is not predicted to use the braking system with the purpose of changing one's mind as regards overtaking the vehicle 1 or the speed alternation of the vehicles 2 and 3, the values of which are always constant,
- what is used in this model is a minimum distance x_{1o} which is equal to the safe following distance and its value according to [4] is: $x_{1o}=(v_1^{\,2}- v_2^{\,2})/2g\cdot 3.6^2(\varphi+f)$, where: g represents the acceleration of gravity, φ – the coefficient of adhesion, f – the rolling resistance force coefficient. The following value: $x_{1o}=10\ m$ is taken in the case when: $v_1 = v_2$.

3. AN ENGINE MODEL

Engines as drive units of motor vehicles operate in extremely varying modes of operation. Existing (drive) engine characteristics are defined and determined in the steady (steady–state) modes of operation and the same characteristics cannot be used for an examination of vehicles' overtaking manoeuvre which is a typical non–steady process. Therefore, what is required is to form an engine model which is to generate the fundamental drive values (power and torque) in the rapidly varying modes of operation, such as a vehicle acceleration during the overtaking process. While establishing such kinds of models, it is possible to use the following two approaches:

- a model based on the calculation of the engine cycle in varying modes of operation,
- a model established due to the use of empirical and semiempirical formulae.

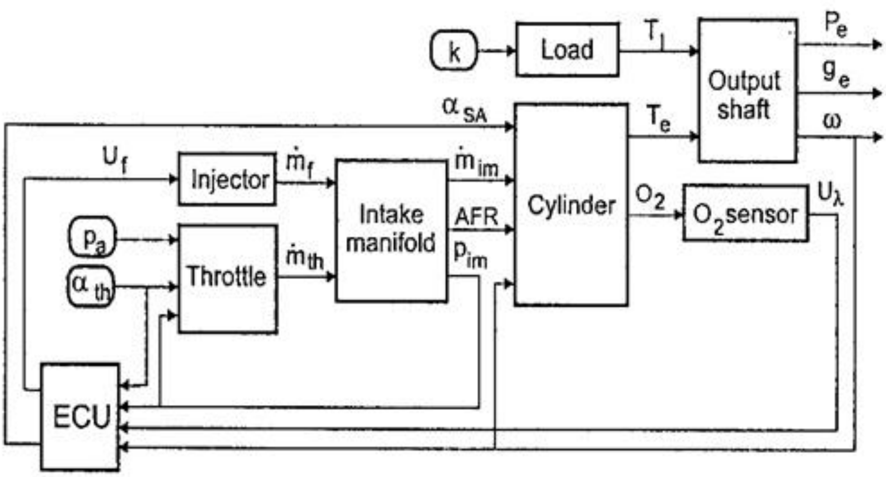

Figure 4.

The first method entails a calculation of an actual work cycle according to the block diagram presented in the Figure 4, [2], taking into consideration actual processes of an alternation of working substance and fuel combustion. The fundamental control variables are: a position of the throttle α_{th} and the load intensity k. The results of a simulation performed while using such a model are presented in the Figure 5, [2].

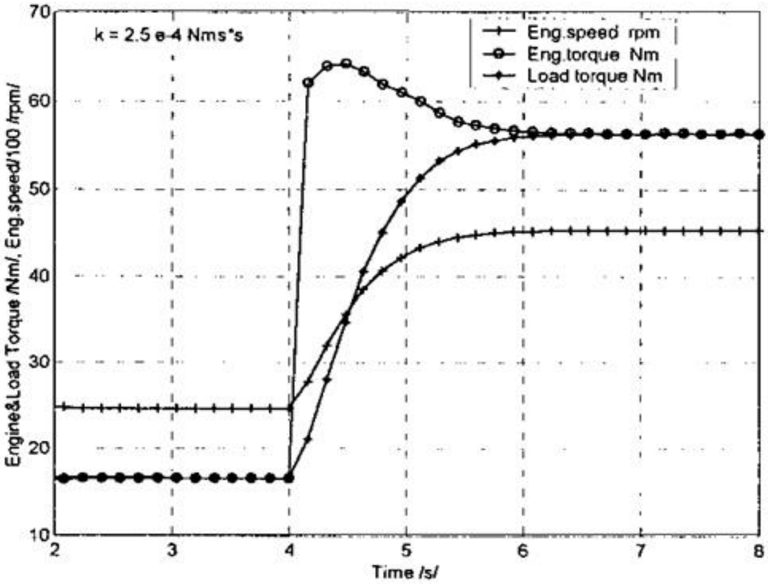

Figure 5 [2.]

The alternations of output values: the torque, load and a number of revolutions have been obtained during a sudden (step-function) opening of the throttle from 20% to 60% and the load factor: $k=2.5e\text{-}4\ Nms^2$.

The second method is based on the use of empirical power and torque formulae. One of the often used formulae is given through the third degree polynomial (1) showing power dependance P_e on the angular speed of the crankshaft ω. By using the well–known connection between the engine power and torque T_e ,(2), an appropriate torque formula is obtained (3).

The values of the coefficients *a, b, c* and *d* given in these formulae are being defined based on the conditions which are valid for the characteristic points on the engine performance and torque curves presented in the Figure 6: the maximal engine torque (n_T ,T_{emax}); the maximal engine power (n_P ,P_{emax}); the moment value if the number of revolutions is n_P, ($T_e(n_P)$). These conditions are given within the following formulae: (4), (5) and (6).

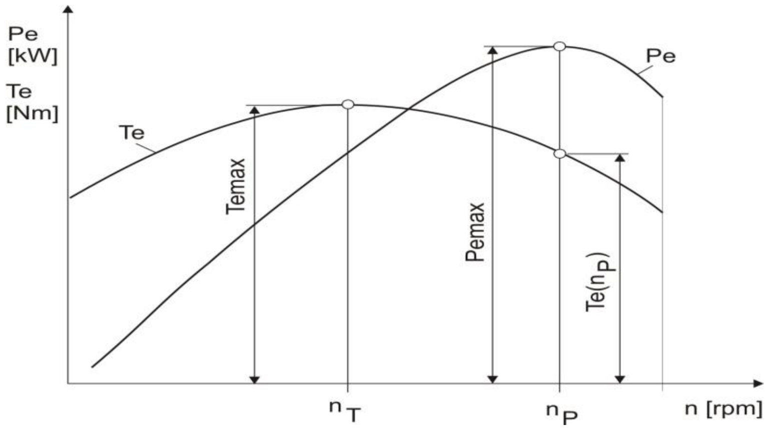

Figure 6.

$$P_e = a\omega^3 + b\omega^2 + c\omega + d \tag{1}$$

$$T_e = P_e / \omega \tag{2}$$

$$T_e = a\omega^2 + b\omega + c \tag{3}$$

$$\frac{dP_e}{d\omega} = \omega \frac{dT_e}{d\omega} + T_e \tag{4}$$

$$\frac{dP_e}{d\omega} = 0 \quad \Rightarrow \quad P_e = P_{e\max} \tag{5}$$

$$\frac{dT_e}{d\omega} = 0 \quad \Rightarrow \quad T_e = T_{e\max} \tag{6}$$

$$e_n = \frac{n_T}{n_P} \tag{7}$$

$$e_T = \frac{T_{e\max}}{T_e\left(n_P\right)} \tag{8}$$

$$d = P_{eo} \geq 0 \quad \Rightarrow \quad e_T \geq 1.5 - 0.5 e_n \qquad\qquad c = T_{eo} \geq 0 \quad \Rightarrow \quad e_n \leq 0.75$$

The shape of the curves presented in the Figure 6 is often expressed through the compliance coefficient: according to the number of revolutions e_n, (7) and the moment e_T, (8); therefore, it is convenient to express the coefficients given in the empirical power and torque formulae in the function of these values.

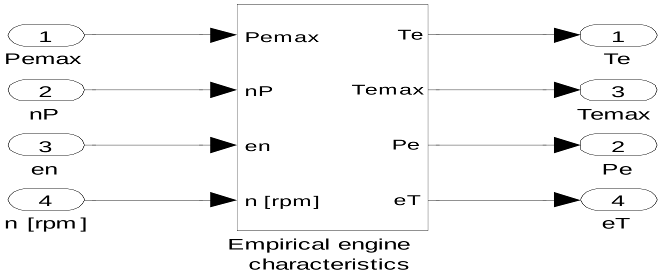

Figure 7.

Commencing with the conditions given by the foregoing formulae, the values of the coefficients *a, b, c* and *d* are defined and also, by using the *Simulink* program from the *MathLab* program package, the process of modeling the power and torque empirical formulae is performed. The model is depicted by the block presented in the Figure 7, in which it is shown how, based on the given inputs: P_{emax} ,n_P ,e_n and n, the output values: $T_e = f(n)$, T_{emax} , $P_e = f(n)$ and e_T are being defined, where n represents the number of revolutions of the crankschaft which has been adopted for the value of the independently variable.

What has been done with the purpose of checking the empirical model is a comparison of the characteristics obtained by its use (the full line drawn in the Figure 8) and the experimental results obtained after running inspection based on the engine of *1.4 l* – for the *Florida* vehicle (the dotted line). The appropriate curves are obtained during the full throttle. Deviations are minimum, as it can be seen from the diagram depicted in the Figure 8.

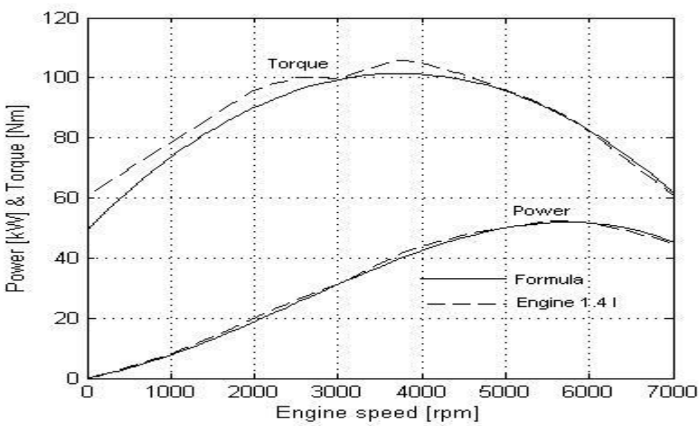

Figure 8.

By using the same model it has been possible to evaluate an influence of the compliance coefficient, according to the number of revolutions, upon the shape of the engine performance and torque curves. The curves presented in the Figure 9 are obtained for the engine with the following characteristics: $P_{emax}=60$ *kW*, $n_P=6000$ *o/min* and $e_n = 0.1 - 0.75$, with the maximum (100%) full throttle.

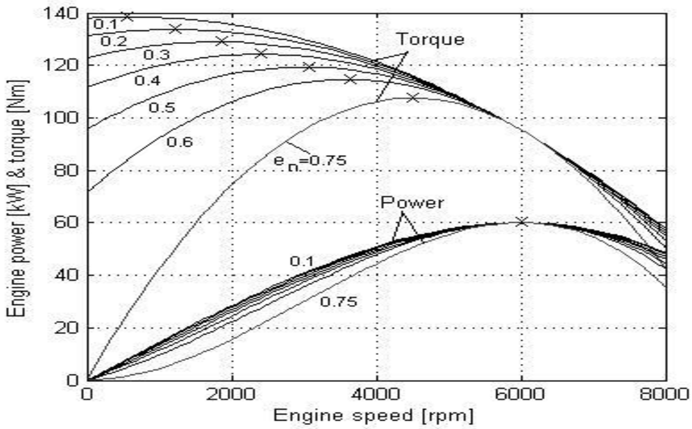

Figure 9.

The compliance coefficient's alternation e_n within the given boundaries influences the position and value of the maximum moment, complete with the shape of the engine performance and torque curves, as it is shown in the diagram presented in the Figure 9.

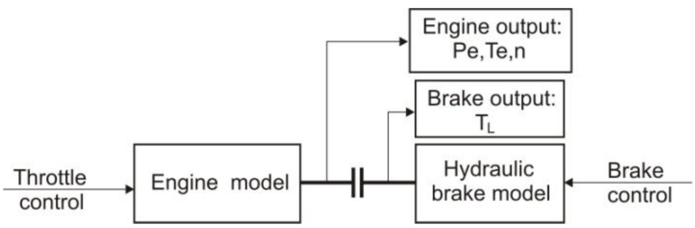

Figure 10.

Modeling of the engine non–steady characteristics is performed, in this case, by introducing a characteristic of the throttle position K_{th}, the value of which ranges within the boundaries: $0 - 1$, which corresponds to the angle of the throttle position: $\alpha_{th} = 0 - 90^o$, that is, to the throttle openness: $0 - 100\%$:

$$[T_e(n)]_{unsteady} = K_{th}\,[T_e(n)]_{steady}$$

The control of the output characterisrics of this model has required the same model to be realized within the *Simulink* program according to the block diagram shown in the Figure10. Except for the engine model (the empirical one), the hydraulic brake model has also been installed with the purpose of the load torque simulation. The simulation results are displayed in the Figure 11, in which steady characteristics are also displayed, the ones corresponding to the throttle openness: *20*, *40*, *60*, *80* and *100%*, complete with the non–steady characteristic occuring during an alternation of the throttle position according to the step function from *20* to *60%*.

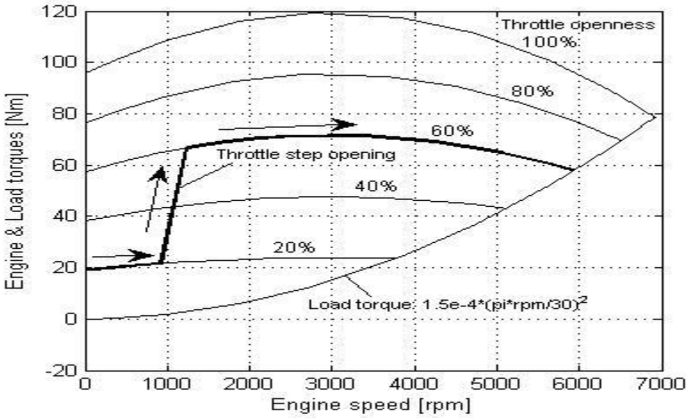

Figure 11.

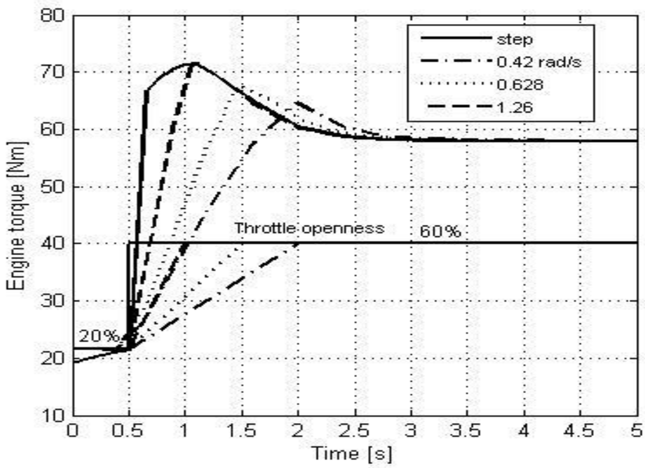

Figure 12.

Non–steady engine characteristics for various speeds of the throttle opening (intensities of effects upon the accelerator pedal) have been simulated by using the same model, commencing with the step function. The results are displayed by means of a diagram given in the Figure 12, and they are considerably in accordance with the results obtained by the use of the model displayed in the Figure 4, which are presented in the Figure 5.

4. VEHICLE MODEL

A model has been formed, by using the *SimDriveline* program, for the purpose of simulating the output values of vehicles (a vehicle distance *S*, vehicle speed *v* and vehicle acceleration *a*) in the regimes of overtaking, and its block diagram is displayed in the Figure 13.

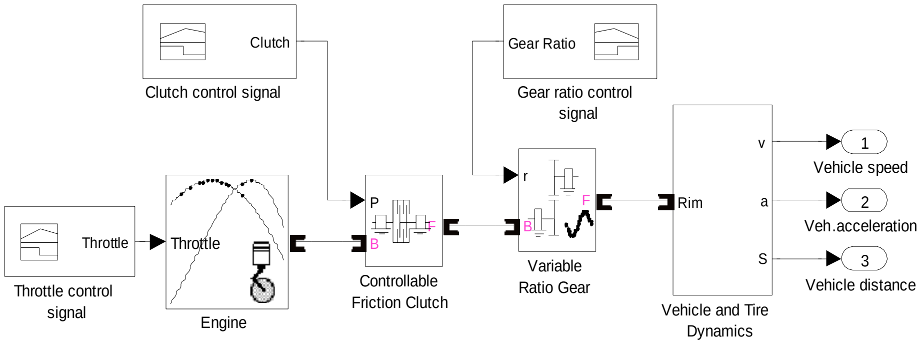

Figure 13.

The model's structure consists of the following: the engine model (which refers to the previously depicted empirical model) with the throttle control signal, the friction clutch with a control signal for its connection and disconnection, the steplessly variable ratio gear with the gear ratio control signal with the purpose of defining the law of the ratio gear alternation and a model of pneumatic tyres for vehicle simulations.

For the purpose of a verification of the output values, the model has been used for a calculation of performances of a *Florida 1.4 l* vehicle, with the input values characterized as it follows: the full throttle (100%), the clutch which is always connected (the time required for the ratio gear alternation is being neglected), the ratio gear alternation is being done according to the diagrams given in the Figure 14.

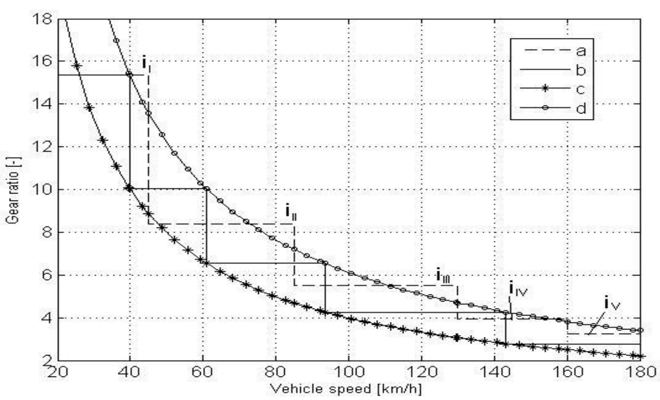

Figure 14.

Diagrams shown in the Figure 14 present the law of the ratio gear alternation of various types of a gear (a gearbox). The line *a* (the dotted one) corresponds to the step– shaped gearbox of the *Florida 1.4 l* vehicle, the line *b* corresponds to the step–shaped gearbox in which the ratio gear alternation is being done with T_{emax} and P_{emax}, the line *c* corresponds to the variable ratio gear which enables the engine running in the T_{emax} regime and the line *d* corresponds to the variable ratio gear which enables the engine running in the P_{emax} regime.

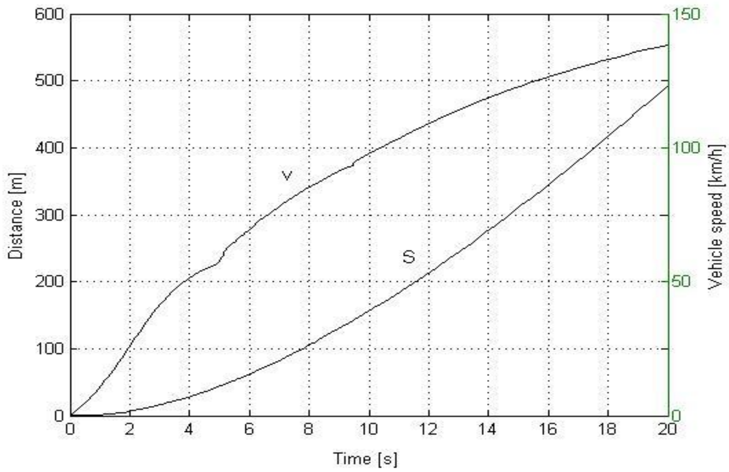

Figure 15.

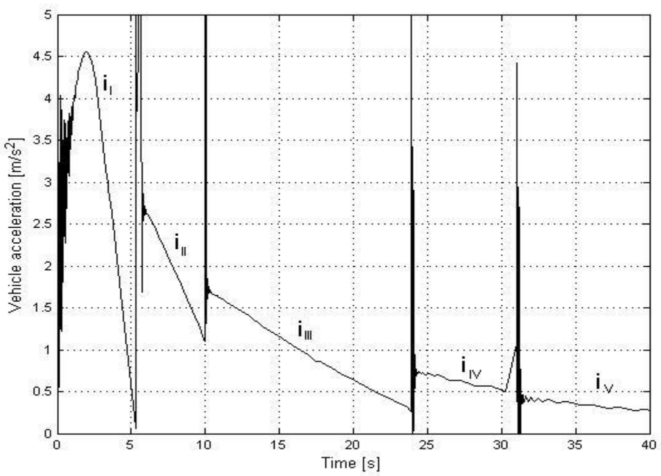

Figure 16.

Comparing the simulation results obtained for the presented laws of the ratio gear alternation, it can be discerned that the most convenient output parameters of the vehicle are obtained while using the law which is defined by the line *d* (P_{emax}), and then by the line *a*. The simulation results for the *Florida 1.4 l* vehicle with the step–shaped gearbox, a characteristic of which is displayed by the line *a* complete with the ratio gear alternation at maximum speeds in the particular ratio gears according to the recommendations given by the producer – are presented in the Figures 15 and 16. The alternations of the vehicle distance *S* and speed *v* are given in the Figure 15, and the Figure 16 represents the vehicle acceleration *a* in the particular ratio gears realized during the acceleration process.

5. SIMULATION RESULTS

A complete model, developed in order to simulate the overtaking process of vehicles, is included in the block diagram given in the Figure 13 complete with the model which, being in accordance with the relations given in the Figures 1, 2 and 3, determines the time of the encounter t_b of the vehicles 2 and 3, complete with the appropriate distances up to that moment: S_b and $S_1(t_b)$. Also, it should be emphasized that the distance indicated in the Figures 2 and 3 with S_1 *(accel.)*, represents the output value from a vehicle model block given in the Figure 13.

What has been made is a research of an influence of the internal combustion engine characteristics upon the traffic safety in the regimes of overtaking, based on the example of a road with two – way traffic (the Figure 3), by using the depicted model presented with the input values characterized in the following manner: $L_1 = L_2 = 4\ m$; $x_{3o} = 150\ m$; $v_{1o} = 90\ km/h = const.$, $v_2 = 80\ km/h = const.$, $v_3 = 80\ km/h = const.$, the coefficient of adhesion: $\varphi = 0.75$ and the rolling resistance force coefficient: $f_o = 0.018$.

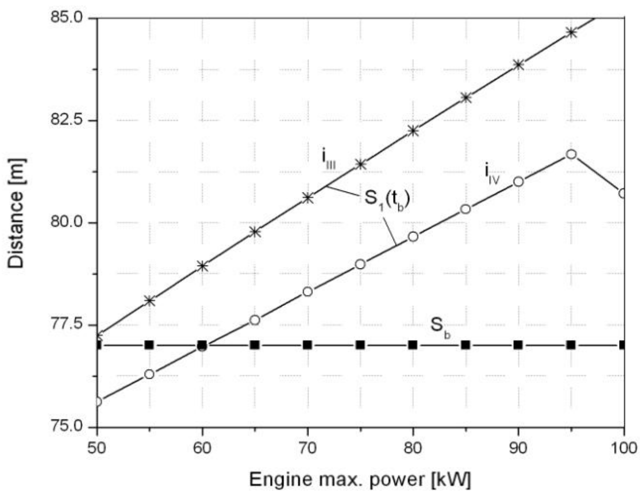

Figure 17.

The time of the encounter of the vehicles 2 and 3 and the appropriate distance, based on the predetermined input values, are characterized in the following manner:

$t_b = 3.285\ s, \qquad S_b = 77\ m.$

An influence of the maximal engine power has been presented by diagrams given in the Figure 17, in order to show the case of an overtaking manoeuvre performance within the third and fourth ratio gears. The character of these dependencies confirms the well–known fact related to the engine characterized by the greater maximal power enabling the same vehicle to achieve greater accelerations, owing to which the time of an overtaking performance is being shortened. Considering the previously introduced criterion for a safe overtaking performance, given within the relation $S_1(t_b)>S_b$, the simulation results presented in the Figure 17, are displaying the fact that in the fourth ratio gear – overtaking can be performed in a safe manner exclusively by means of the engine, the maximal power of which is greater than *60 kW*. In other words, it means that vehicles provided with engines producing less power have to perform the overtaking process throughout the third ratio gear or to commence the same process at the point when:

$x_{3o} > 150\ m.$

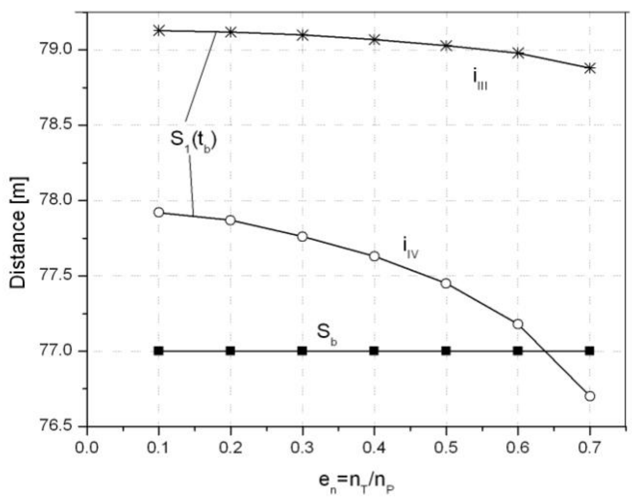

Figure18.

The diagram given in the Figure 18 displays an influence of the compliance coefficient e_n upon the overtaking manoeuvre. This influence reflects the character of the torque alternation depending on the value of the coefficient e_n which has been presented in the Figure 9. Namely, an increase in values of the coefficient e_n, leads to a decrease in values of the maximal engine torque within the empirical model.

An influence of the maximum throttle opening has been presented in the Figure 19. For the predetermined input values, a safe overtaking process can be performed only if the throttle openings are greater than 80%.

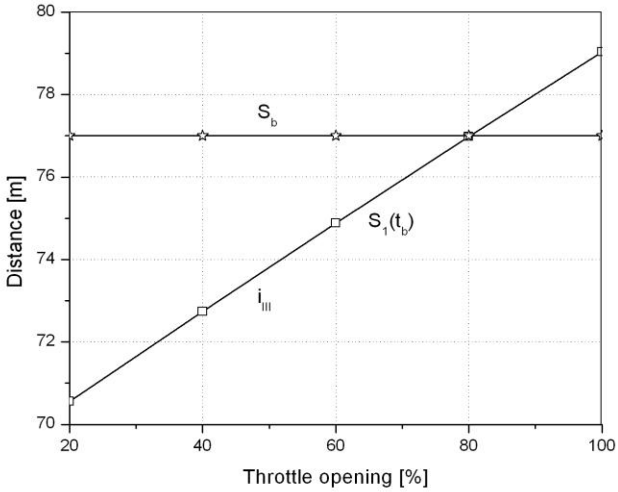

Figure 19.

For the purpose of a verification of the simulation results obtained by using the developed simulation model, characteristics of modern medium class passenger vehicles have been analyzed according to the criterion of maximum accelerations which can be achieved in the third ratio gear at speeds ranging from 60 to 100 km/h and in the fifth ratio gear at speeds ranging from 80 to 120 km/h. The results have been presented in diagrams given from the Figure 20 to the Figure 24. As it has previously been emphasized, vehicles with greater acceleration values have better performances regarding the overtaking manoeuvre process.

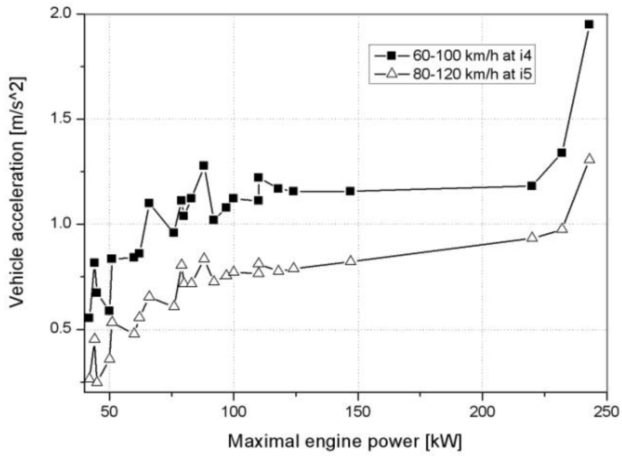

Figure 20.

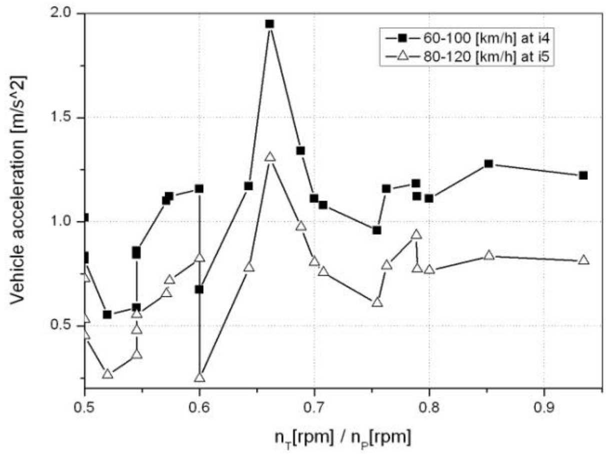

Figure 21.

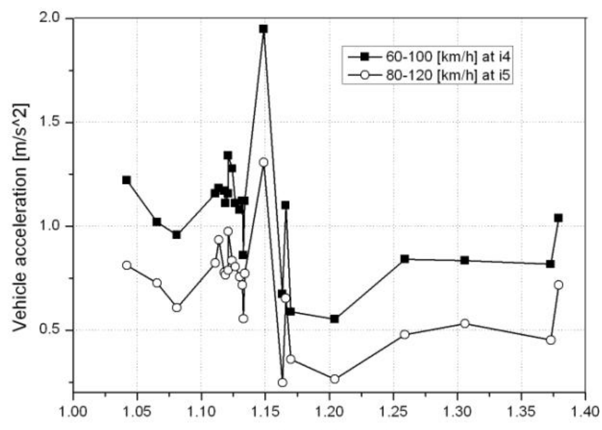

Figure 22.

Diagrams given in the Figures 20, 23 and 24, display a vehicle acceleration dependency on the maximal engine power, that is, on the maximal engine torque. These dependencies show the same direction of the influence just like the results obtained owing to the simulation models: an increase of the maximal engine power, that is, the maximal engine torque leads to the increase of acceleration values, and therefore, the time required for the overtaking manoeuvre performance is being shortened.

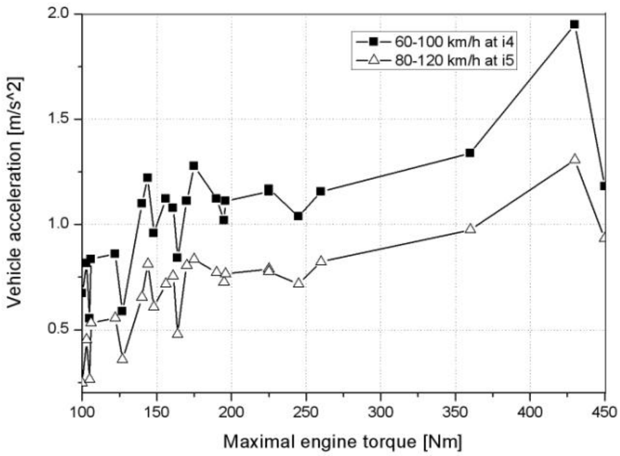

Figure 23.

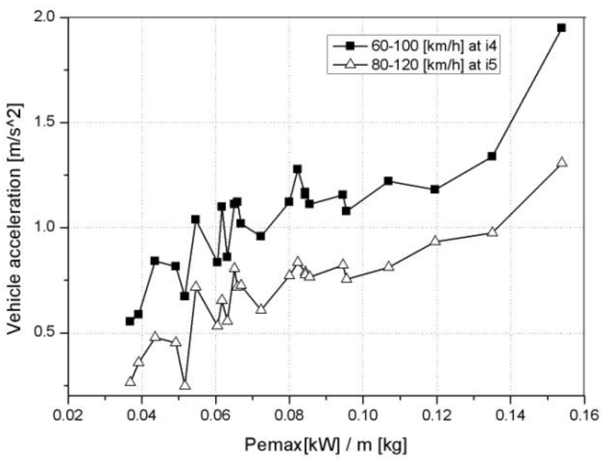

Figure 24.

The influence of the compliance coefficients displayed within diagrams given in the Figures 21 and 22 primarily depends on the maximal engine torque value which is equal to the given value of the coefficient itself. Therefore, this influence has a stochastic character. As regards modern engines used for passenger vehicles' drive units, and owing to the application of variable systems, the engine torque curve is characterized by the constant value of the maximum in a wide area of a number of revolutions (1000-5000 o/min), which renders the influence of the compliance coefficients invariant .

6. CONCLUSIONS

The developed model of the overtaking manoeuvre of vehicles enables an assessment of influences of a greater number of parameters, commencing with the engine characteristics, the intensity of its effect upon the throttle control linkage, the friction clutch connection process, transmission characteristics and vehicle characteristics. The engine influence as a vehicle's drive unit has been analyzed in detail by presenting a concrete case.

Obtained results confirm the fact that the engine characteristics are the ones having a crucial influence upon the safety of vehicles' overtaking manoeuvre performance. The fundamental parameter significant for the overtaking process is, by all means, a vehicle acceleration which primarily depends on the intensity of its effect upon the throttle control linkage. All of these influences have been analyzed by using the above presented model and interpreted by displaying appropriate diagrams.

Having a certain knowledge of the engine characteristics regarding the maximum acceleration values which can be achieved by means of the given vehicle – is significant particularly when the overtaking process is performed in cases when it is required to avoid a critical situation owing to the misjudgement of speeds and following distances. Except for the analysis of influences of the engine, transmission and vehicle's characteristics, the model indirectly enables the assessment of a driver's reaction regarding his/her evaluation of the following distances (x_{1o} ,x_{3o}) and speeds (v_2 ,v_3). By varying these values, the boundary valves can be established, the same ones owing to which the overtaking manoeuvre can be performed in a safe manner by using the given engine and vehicle.

7. REFERENCES

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