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M V M   Mobility Vehicle Mechanics

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[1] KINEMATIC ANALYSIS OF PISTON MECHANISM IN VALVELESS INTERNAL COMBUSTION ENGINE WITH MORE COMPLETE EXPANSION

Jovan Dorić, Ivan Klinar, *Faculty of technical sciences, Novi Sad, Serbia*

UDC: 621.432:531.1

Abstract

This paper presents kinematic analysis of valveless internal combustion engines with more complete expansion of the working body. Radial-rotary valveless internal combustion engine with more complete expansion of the working body, is one of the possible ways of converting chemical energy of fuel into mechanical work. The engine is designed so that the changes of thermodynamic state of the working body are different than in conventional engines. Specific differences are reflected in more complete expansion of working body somewhere known as Miller cycle (modern version of Atkinson cycle), valveless gas flowing and full discharge the combustion chamber of the residual products of combustion. In this construction the movement of the piston is a different than in a conventional piston mechanism. Movement of the piston has brought another motion, ie. rotation of the cylinders around the axis which is placed at exactly defined position. Ratio between the drive shaft and the movable cylinder is (-1). The paper presents values of velocity and acceleration of the important points of the kinematical group consisting in described concept. An account of the basic values are also given in tables and chart form, also this paper presents analysis of differences between this IC engine and conventional mechanism. In this paper was used material of patent applications under the number 2008/0607 of Intellectual Property Office of the Republic of Serbia.

Key words: IC engine, kinematic, Miller cycle.

KINEMATSKA ANALIZA KLIPNOG MEHANIZMA BEZVENTILSKOG MOTORA SUS SA POTPUNIJIM ŠIRENJEM RADNOG TELA

UDC: 621.432:531.1

Rezime: U radu je prikazana kinematska analiza bezventilskog motora sa unutrašnjim sagorevanjem i potpunijim širenjem radnog tela. Radijalno-rotacioni bezventilski motor SUS sa potpunijim širenjem radnog tela predstavlja jedan od mogućih načina pretvaranja hemijske energije goriva u mehanički rad. Motor je tako dizajniran da su promene stanja radnog tela drugačije nego kod konvencionalnog motora SUS. Razlike se ogledaju u potpunijem širenju radnog tela poznato i kao Milerov ciklus (modernija verzija Atkinsonovog ciklusa), bezventilskom razvodu radnog tela i potpunom pražnjenju komore

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za sagorevanje od zaostalih produkata sagorevanja. Kod ove konstrukcije motora kretanje klipova je drugačije izvedeno. Kretanju klipa je dovedeno još jedno kretanje, tj. obrtanje cilindara oko ose koja je postavljena na tačno definisanom položaju. Prenosni odnos između kolenastog vratila i pokretnih cilindara iznosi (-1). U radu su takođe prikazane vrednosti brzine i ubrzanja najvažnijih tačaka novog kinematskog lanca. Dobijene vrednosti date su u vidu tabela i dijagrama, gde je takođe dat akcenat na bitne razlike između ovog i konvencionalnog motora SUS. U radu je korišćen materijal patentne prijave broj 2008/0607 zavoda za intelektualnu svojinu Republike Srbije.

Ključne reči: motor SUS, kinematika, Milerov ciklus

KINEMATIC ANALYSIS OF PISTON MECHANISM IN VALVELESS INTERNAL COMBUSTION ENGINE WITH MORE COMPLETE EXPANSION

Jovan Dorić [1], Ivan Klinar

UDC: 621.432:531.1

INTRODUCTION

Today, there is a very large number of successful construction of internal combustion engines, which are applied to various fields of science and technology. In some areas IC engines are so dominant without concurrence of other types of engines. This fact suggest that today's internal combustion engines are at a high technical level. However, construction of piston stroke internal combustion engines that are now used is based on inefficient thermodynamic and mechanical concept. It can be said that the main characteristics of today's engine is very small amount of work in relation to used fuel, in other words, today's engines have a very low coefficient of efficiency. Realistically speaking *Otto* engines today use about 25% of input energy, while *diesel* construction about 30% (in some cases can be expected a little more). Approximately 35% of the *Otto* engine and 30% of heat in the diesel engines goes through exhaust and around 33% goes for cooling the engine in both versions, other 7% is attributed to friction and radiation [1]. For illustration can be taken into account combustion of one liter of diesel in the classical combustion engines. Combustion of this amount of fuel frees approximately 39 MJ of power, the engine output shaft is generated only around 13 MJ, while with other 26 MJ engine heated environment.

Considering the present development trends, trends for more efficient use of fuel resources, the problem of global warming and other environmental factors, development of internal combustion engines will certainly move towards the reduction of fuel consumption[2,3]. In this paper one of the possible ways of reducing thermodynamic losses in the IC engine is shown.

KINEMATIC OF CONVENTIONAL IC ENGINE

Movement of the piston in conventional IC engines is based on relatively simple kinematic [4]. Fig 1. shows the kinematics of a crankshaft drive with crossing, in which the longitudinal crankshaft axel does not intersect with the longitudinal cylinder axel, but rather is displaced by the lenght e.

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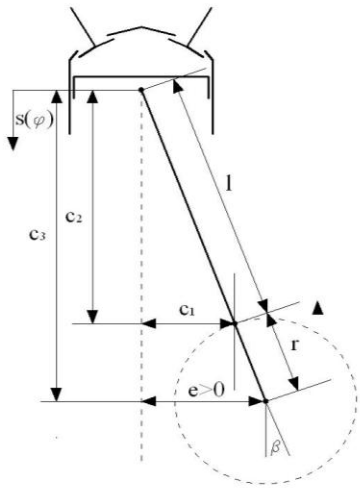

Figure 1: *Kinematic scheme of conventional IC engines*

For the piston path $s(\varphi)$, it follows from Fig 1.

$$s(\varphi) = c_3 - c_2 - r\cos(\varphi - \beta) \tag{1}$$

from which with

$$\sin\beta = \frac{e}{r+l} \quad \text{and} \quad \beta = \arcsin\left(\frac{e}{r+l}\right) \text{, respectively}$$
$$\begin{aligned} c_1 &= e - r\sin(\beta - \varphi) \\ c_2 &= \sqrt{l^2 - c_1^2} \\ c_3 &= \sqrt{(r+l)^2 - e^2} \end{aligned} \tag{2}$$

Finally

$$s(\varphi) = \sqrt{(r+l)^2 - e^2} - \sqrt{l^2 - [e + r\sin(\varphi - \beta)]^2} - r\cos(\varphi - \beta) \tag{3}$$

results. The derivative provides for the piston speed the relation

$$\frac{ds}{d\varphi} = r\sin(\varphi - \beta) + \frac{r[e + r\sin(\varphi - \beta)]\cos(\varphi - \beta)}{\sqrt{l^2 - \left[e + r\sin(\varphi - \beta)^2\right]}} \tag{4}$$

With the definition of the cylinder volume

$$V(\varphi) = V_C + D^2 \frac{\pi}{4} s(\varphi) \tag{5}$$

follows for the alteration of cylinder volume

$$\frac{dV}{d\varphi} = D^2 \frac{\pi}{4} \frac{ds}{d\varphi} \tag{6}$$

With the eccentric rod relation $\lambda = r/l$, it follows for the limiting case *e=0.*

$$s(\varphi) = r\left\{[1-\cos(\varphi)] + \frac{1}{\lambda_k}\left[1-\sqrt{1-\lambda_k^2 \sin^2(\varphi)}\right]\right\} \tag{7}$$

and

$$\frac{ds}{d\varphi} = r\left[\sin(\varphi) + \frac{\lambda_k}{2} \frac{\sin(2\varphi)}{\sqrt{1-\lambda_k^2 \sin^2(\varphi)}}\right] \tag{8}$$

Through this kinematics, movement of the piston is very limited. Conventional piston movement in Otto engine cause changes of working fluid described in Fig 2.

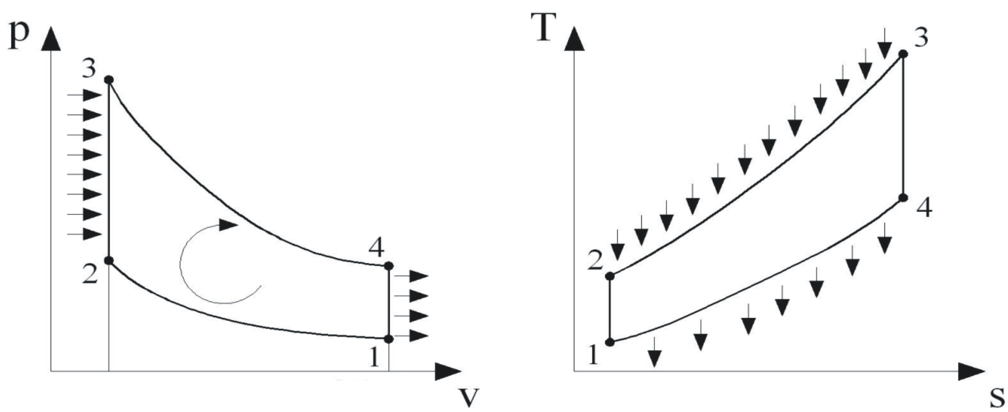

Figure 2: *PV and TS diagram of conventional IC engines*

As in Fig. 2, the compression 1-2 process is isentropic; the heat addition 2-3, an isochoric process; the expansion 3-4, an isentropic process; and the heat rejection 4-1, an isochoric process. For the heat addition (2-3) and heat rejection (4 -5) stages, respectively, it is assumed that heating occurs from state 2 to state 3 and cooling ensues from state 4 to state [5].

Miller cycle

The Miller cycle, named after its inventor R.H. Miller, has an expansion ratio exceeding its compression ratio. The Miller cycle, shown in Fig. 3, is a modern modification of the Atkinson cycle (i.e., a complete expansion cycle). In the Miller cycle, the intake valve is left open longer than it would be in an Otto cycle engine. In effect, the compression stroke is two discrete cycles: the initial portion when the intake valve is open and final portion when the intake valve is closed. TS and PV diagrma of Miller cycle are presented in Fig. 3.

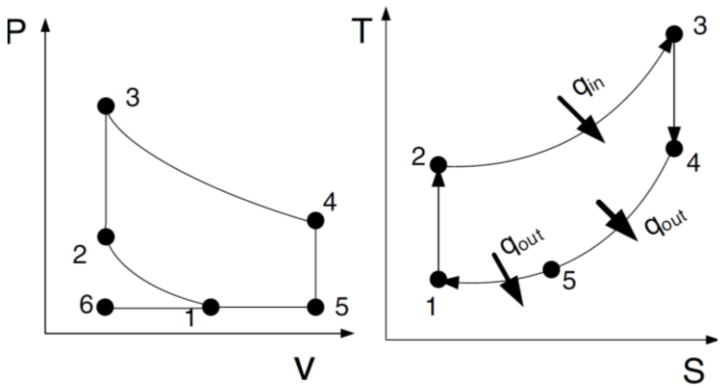

Figure 3: *Pv and TS diagram of Miller cycle IC engine*

It is obvious that is very important for thermodynamic efficiency that temperature at the end of expansion be as small as possible. This lower temperature will give a larger surface of TS diagram, larger surface of TS diagram means greater efficiency. As can be seen from PV diagram, in Miller cycle compression and expansion stroke are not same length geometrically speaking. With conventional piston mechanism is very difficult to achieve this motion. In new internal combustion engine this complex movement can be perform much easier. It is very important for modern IC engines to work on such cycle, because this thermodynamic cycle have many advatanges over standard Otto cycle, one of them are:

- Less fuel consumption.
- Less heating of the environment.
- Less pollution

DESCRIPTION OF ENGINE

In this mechanism movement of piston is different than in conventional IC engine. The main idea can be described with kinematic scheme in Fig. 4.

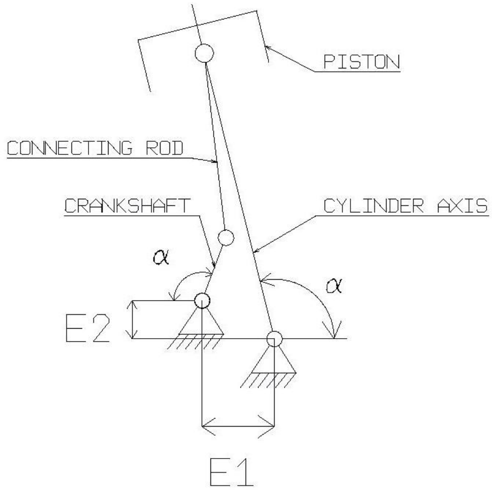

Figure 4*: *Kinematic scheme of new piston mechanism

As seen in Fig. 4, in this case to the conventional piston mechanism scheme is added one more movement (rotation) of cylinders around axis at exactly defined position. Precisely defined position of the crankshaft and movable cylinders is very important, because these values define more complete expansion of working fluid and full discharge combustion chamber of the residual products of combustion, impact of these values on more complete expansion are described in [6].

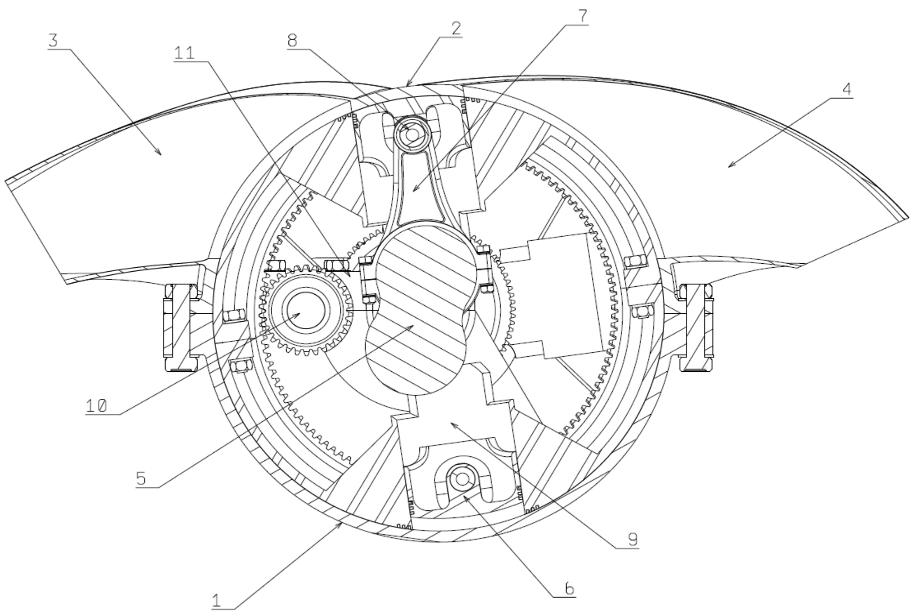

***Figure 5:** Cross-section of new IC engine*

Basic parts of engine are shown in Fig 5. The engine consist of the lower part of engine block (1) and the upper part of the engine block (2), movable cylinders - rotors (9), in which are placed pistons (6), connected to the crankshaft (5), through the piston rod (7) and piston pin (8). With (3) and (4) are presented intake and inlet manifold respectively, and with (10) and (11) are described gear mechanism. Pistons are placed radially to the crankshaft, whereby the rotation axes of crankshaft located at precisely defined position The Miller cycle, named after its inventor R.H. Miller, has an expansion ratio exceeding its compression ratio. The Miller cycle, shown in Fig. 1, is a modern modification of the Atkinson cycle (i.e., a complete expansion cycle). In the Miller cycle, the intake valve is left open longer than it would be in an Otto cycle engine. In effect, the compression stroke is two discrete cycles: the initial portion when the intake valve is open and final portion when the intake valve is closed. In new valveless IC engine with more complete expansion of working fluid Miller cycle is achieved in a different way.

KINEMATIC OF VALVELESS IC ENGINE WITH MORE COMPLETE EXPANSION

Projections of the kinematic schemes from the Fig. 4 may be expressed through the two axes.

Projection of the x-axis s is given in next Eg (9)

$$s \cdot \sin(\alpha) = L \cdot \sin(\beta) + r \cdot \sin(-\alpha) + l \cdot \sin(\gamma)$$

$$\gamma = a\sin\left(\frac{s \cdot \sin(\alpha) - L \cdot \sin(\beta) + r \cdot \sin(\alpha)}{l}\right) \tag{9}$$

and for y-axis Eg (10):

$$s \cdot \cos(\alpha) = L \cdot \cos(\beta) + r \cdot \cos(-\alpha) + l \cdot \cos(\gamma)$$

$$\gamma = \pi - a\cos\left(\frac{(-s) \cdot \cos(\alpha) + L \cdot \cos(\beta) - r \cdot \sin(\alpha)}{l}\right) \tag{10}$$

Values of speed, acceleration and position will depend on the selected kinematics parameters. In this case the tendency was the construction of the IC engine that can run the usual car. In accordance with the use of engine in the field of passenger motor vehicles the following parameters are selected.

Kinematics parameters of piston mechanism:

- r =30 [mm] - crankshaft radius,
- L=115 [mm] - connecting rod lenght,

- $\lambda_k = \frac{r}{L} = \frac{30}{115} = 0.26$ - rod relation,
- $\omega_k = \frac{\pi \cdot n}{30} = \frac{3.14 \cdot 3000}{30} = 314\left[s^{-1}\right]$- angular speed of crankshaft,
- $\omega_c = -\frac{\pi \cdot 3000}{30} = -\frac{3.14 \cdot 3000}{30} = -314\left[s^{-1}\right]$ - angular speed of rotor,
- $\omega_k = -\omega_c$,
- $E_1 = 13{,}8[mm]$,
  $E_2 = 2{,}3[mm]$.

Simple kinematic analysis shows that the lenght of certain strokes are:

- $s_u = 49.3[mm]$ - intake stroke
- $s_s = 44.5[mm]$ - compression stroke
- $s_š = 72.1[mm]$ - expansion stroke
- $s_i = 76.9[mm]$- exhaust stroke

Piston distance in function of the angle of crankshaft are shown in Table 1.

Table 1: *Piston distance from rotation axis of rotor*

angle *[^{o}]*	*distance* [mm]	*angle* *[^{o}]*	*distance* [mm]	*angle* *[^{o}]*	*distance* [mm]	*angle* *[^{o}]*	*distance* [mm]
0	147.924	90	99.4357	180	137.59	270	72.1387
4.5	147.421	94.5	100.545	184.5	134.059	274.5	73.4323
9	146.027	99	102.169	189	129.842	279	75.2788
13.5	143.826	103.5	104.289	193.5	125.068	283.5	77.6872
18	140.933	108	106.873	198	119.886	288	80.6633
22.5	137.479	112.5	109.877	202.5	114.457	292.5	84.2051
27	133.61	117	113.244	207	108.94	297	88.2981
31.5	129.478	121.5	116.898	211.5	103.488	301.5	92.9091
36	125.231	126	120.745	216	98.2372	306	97.9817
40.5	121.01	130.5	124.673	220.5	93.3021	310.5	103.433
45	116.94	135	128.557	225	88.7718	315	109.151
49.5	113.127	139.5	132.259	229.5	84.7104	319.5	114.999
54	109.661	144	135.636	234	81.1595	324	120.821
58.5	106.607	148.5	138.549	238.5	78.1423	328.5	126.446

angle [º]	*distance* [mm]	*angle [º]*	*distance* [mm]	*angle [º]*	*distance* [mm]	*angle [º]*	*distance* [mm]
63	104.017	153	140.863	243	75.6682	333	131.703
67.5	101.924	157.5	142.465	247.5	73.7383	337.5	136.428
72	100.352	162	143.262	252	72.349	342	140.474
76.5	99.3133	166.5	143.187	256.5	71.4958	346.5	143.719
81	98.8144	171	142.209	261	71.1758	351	146.073
85.5	98.8563	175.5	140.331	265.5	71.3889	355.5	147.481

Values from the Table 1 can be described through diagram, such a diagram is shown in Fig 10. From the diagram it can be observed that all four strokes have different lenghts. The values of the length of strokes can be easily changed depending on desired ratio of compression and expansion ratio. In this case, the tendecy was to accomplish the compression ratio of approximateley 10.3 , and expansion ratio of approximateley 15.

It is interesting to make an analysis of piston movement through space, it can be easily done by solving Eq (9) and Eq (10) with respect to values of kinematics parameters of piston mechanism. In this case movement of the piston is not a linear, because in this engine kinematics of piston mechanism forces the piston to move over curve. Piston is moving on very complex curve. This curve can be described by the following Fig. 6. As can be seen from this diagram this curve is very similar to ellipse. The parameters of this curve depends on the value of the following factors: crankshaft radius , connecting rod lenght, rod relation and also of values of eccentric parameters E_1 and E_2. It is clear that with these parameters it is easy to achieved very large number of different types of curves, in other words different type of piston motion law. In this type of kinematics chain small changes of values are reflected in wide variations. For example, angular velocity must be of the same intensity for the movable cylinders and crankshaft, but opposite directions. Also the gear ratio must be respected, only in this case engine can reach the proper cycle operation.

By changing described values, engine can achieve much higher expansion ratio than standard conventional IC engine. This is very important because with longer expansion stroke working fluid can reach much lower temperature at the end of expansion According to Fig. 3 it is clear that with the reduction of T_4 (temperature at the end of expansion) cycle achieves a greater surface of TS diagram, larger surface of TS diagram mens greater efficiency**.**

After analyzing the trajectories of the piston, it is necessary to perform the analysis of velocity and acceleration. Compared to traditional engine, in this concept velocity and acceleration have more complex components. Fig. 5 shows IC engine with kinematic scheme from Fig. 4, as can be seen from both pictures piston will be forced to complex movement. This complex movement will consist of two components. First one is the result of movement of piston through the cylinder, while the other component is velocity due rotation of cylinders. These two values of velocities are very different in terms of the intensity and law changes. Vector addition of these components gives absolute velocity of piston. This design of IC engine allows development of all four cycle for one revolution of crankshaft. For these reasons, this engine can achieve the same number of working cycles as conventional IC engine with the half angular velocity of crankshaft. Therefor in the further analysis was used angular velocity of crankshaft of 3000 rpm, because this values of angular speed corresponding to 6000 rpm in conventional IC engine, which is near the maximum for standard engines.

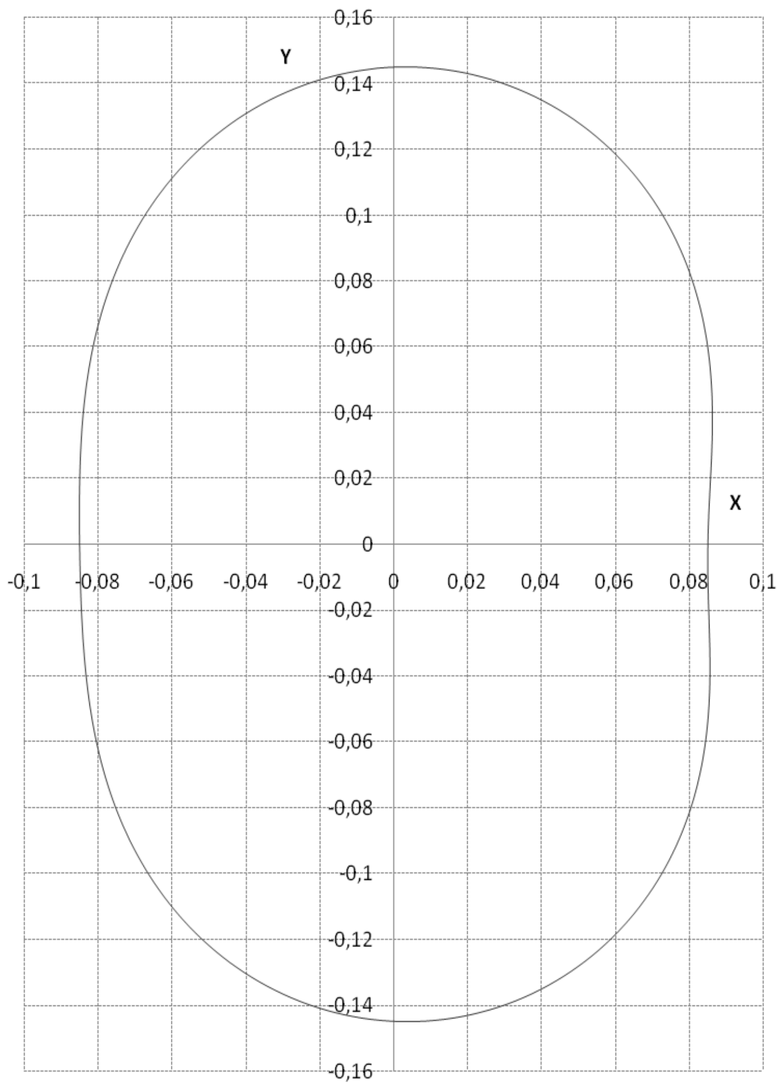

Figure 6: *Piston path in new IC engine*

The graphic in Fig. 7 represents the change in velocity of piston through the cylinder. As can be see from the same Fig. 7 law of motion is very similiar to standard motion law of piston in standard engine. However, noted is that the amplitude for some strokes are not the same, which is direct consequence of non-conventional kinematics, in other words the tendency to make a longer expansion than compression stroke. In the next diagram in Fig. 8

change of absolute velocity of piston is shown. Here can be observed the greatest differences between conventional and non-conventional kinematics. From the Fig. 8 it can be noticed that the velocity of piston is never equal to zero, in other words, piston in this engine never stops, also velocity is nearly constant in a large part of movement. As is well known in standard IC engine pistons need four times per whole cycle to stop. Finally, Fig. 9 represents the relative acceleration of the piston. This results is especially interesting, because he shows how the engine construction achieves more complete expansion. As is well know shape of the acceleration curve depends on the kinematics factor, i.e. the number of maximum values of acceleration function depends of ratio R/L. In this mechanism for the first part of movement acceleration curve have one maximum values and for the second part have two maximum, this can be concluded by observing Fig. 9. This feature achieve more complete expansion.

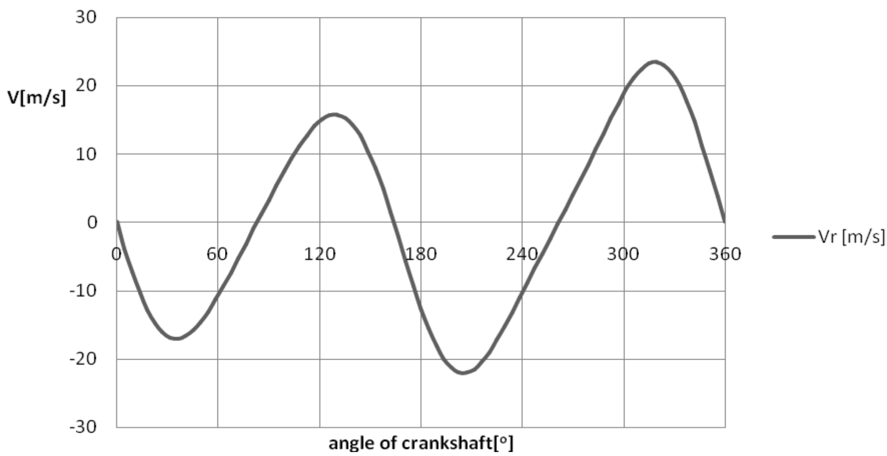

Figure 7*: *The relative velocity of piston at 3000 rpm (velocity of piston through the cylinder)

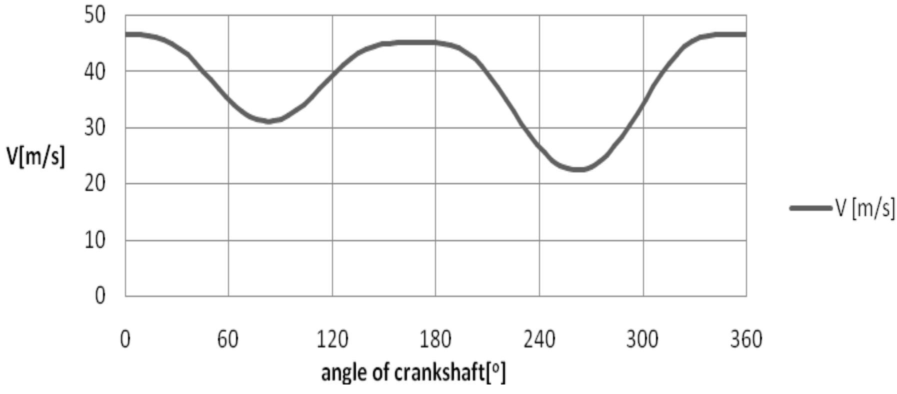

Figure 8*: *Absolute velocity of piston at 3000 rpm (velocity which the piston moves through engine housing)

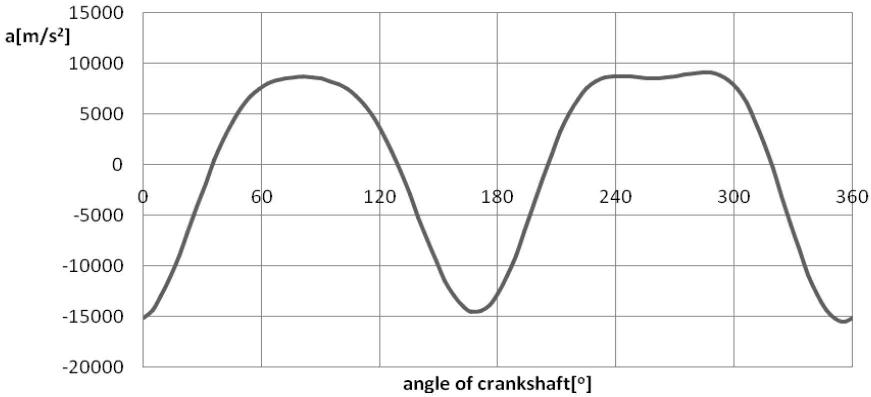

Figure 9: *Relative acceleration of piston at 3000 rpm*

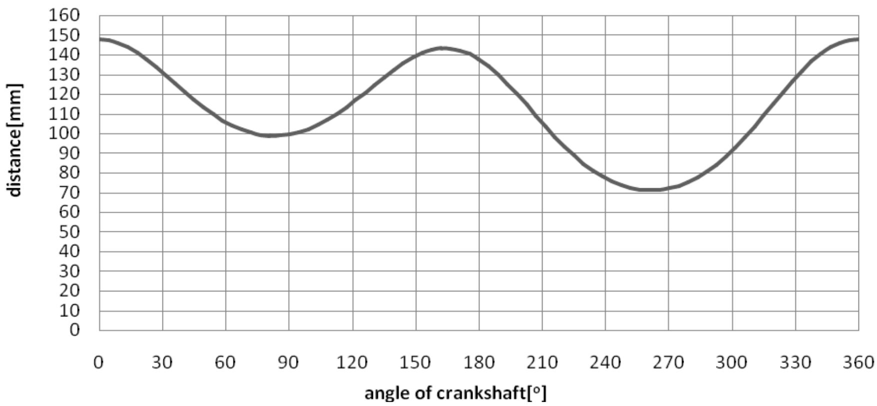

Figure 10: *Distance from piston and rotation axis of rotor*

CONCLUSIONS

This paper deals with kinematics analysis of valveless IC engine with more complete expansion. Final results are shown through diagrams and charts. As can be seen from Fig. 3, in this engine piston in-plane motion are translated to rotary motion of crankshaft, unlike conventional IC engines where pistons have translatory motion. Also, it can be seen that apsolute piston velocity are greater than in usually IC engine (with the same number of cycles per time), but the changes of piston speed per time are smaller than in conventional engine. From diagram in Fig. 10 are described how with this kinematics piston distance have different values for compression and expansion stroke, this feature is very important because in this way working fluid can reach much lower temperature at the end of expansion. Lower temperature at the end of expansion means more efficiency. Through described kinematics IC engines easy can achieve thermodynamic cycles with increased efficiency.

ACKNOWLEDGMENTS

This study has been supported by the Ministry of Science and Technological Development, Republic of Serbia (Project No. TR 20078).

REFERENCES

[1] Dorić J.: ’’Valveless IC engine with more complete expansion’’, Tractor and Power machines, Volume 14, Issues 2/3, p. 58-64.

[2] Gruden D.:’’Automobile development followed by prophency of ecological disasters’’, Mobility and Vehicle Mechanics, Volume 34, Issues 3, p. 4.

[3] Gruden D.:’’Technical measure to reduce carbon dioxide emissions on the road traffic’’, Mobility and Vehicle Mechanics, Volume 32, Issues 3/4, p. 54-69.

[4] Merker G., Schwarz C., Stiesch G., Otto F.:’’Simulating Combustion’’, Springer-Verlag Berlin Heidelberg 2006, Germany.

[5] Klinar I.:’’Motori sa unutrašnjim sagorevanjem’’, Faculty of Technical Sciences, Novi Sad, 2009.

[6] Dorić J.:’’Radial-rotary Valveless four-cycle IC engine with more complete expansion of the working group’’, patent application number 2008/0607, at the Intellectual Property Office of the Republic of Serbia, Belgrade 2008.

[1] EXPERIMENTAL RESEARCH OF DYNAMIC STRESSES OF MOTORCYCLE'S FRAME

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UDC: 629.118.6-58

Abstract

In this paper are presented the results of all three acceleration components of motorcycle's frame. For the experiment a motorcycle HONDA MTX 125R was used and B&K vibration analyzer, model 4447 measuring equipment. During the ride, the speeds were constant with variable intensity in the range of 30 km/h till 130 km/h. The experiment was performed on the straight and smooth asphalt road. The quality of the road is same as the quality of most of the main roads in Serbia. The length of the pathway was 1100m. On the other hand, a simple dynamic model of motorcycle was created, which allows the calculation of dynamic reactions of the ground by means of acceleration components measured on the frame. The goal of this paper was to identify dependence of vertical ground reactions on front and rear wheel for different constant speeds. Measurements have shown that even during the straight ride by motorcycle with constant speeds there is a considerable ratio of lateral acceleration component, which is not the case during the ride by car, for instance. The purpose of this paper is to indicate to risks of the ride with speeds over those identified on the Serbian roads, by means of experimental identification of the speed, which in some sense, for certain road quality, is the critical speed.

Key words: acceleration measurement, motorcycle's frame, dynamic model.

EKSPERIMENTALNA ISTRAŽIVANJA DINAMIČKIH NAPREZANJA RAMA MOTOCIKLA

UDC: 629.118.6-58

Rezime: U radu su prezentirani rezultati merenja sve tri komponente ubrzanja na ramu motocikla. Motocikl je Honda MTX 125R, a korišćena merna oprema je analizator vibracija B&K, model 4447. Vožnja je bila konstantnim brzinama različitog intenziteta i to od 30km/h do 130km/h. Eksperiment je izveden na deonici asfaltnog puta koji je ravan i prav. Kvalitet puta odgovara kvalitetu većine magistralnih puteva u Srbiji. Dužina deonice je bila 1100 m.

S druge strane, postavljen je jednostavan dinamički model motocikla koji omogućava izračunavanje dinamičkih reakcija tla korišćenjem izmerenih komponenti ubrzanja na

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ramu. Cilj rada je bio da se identifikuje zavisnost vertikalnih reakcija tla prednjeg i zadnjeg točka za različite konstantne brzine. Merenja su pokazala da i pri pravolinijskom kretanju motocikla konstantnom brzinom postoji značajan udeo bočne komponenete ubrzanja, što, na primer, kod automobile nije slučaj.

U radu se, eksperimentalnom identifikacijom brzine koja je u neku ruku za zadati kvalitet puta kritična, želelo da ukaže na rizik brze vožnje brzinama većim od identifikovane na putevima Srbije.

Ključne reči: merenje ubrzanja, ram motocikla, dinamički model.

EXPERIMENTAL RESEARCH OF DYNAMIC STRESSES OF MOTORCYCLE'S FRAME

Branislav Aleksandrović*[1]*, Marko Đapan, Aleksandra Janković

UDC: 629.118.6-58

INTRODUCTION

Motorcycle dynamic characteristics and the paths/roads along which they have been driven often do not have correlated characteristics, i.e. the quality of the road often is not appropriate to high dynamic performance of motorcycles.

In this paper work is considered the effect of vertical oscillations to dynamic reactions, which are in direct relation with the force transmitted by the wheel, both in lateral and in longitudinal direction. It's quite obvious that vertical vibrations have considerable effect to motorcycle safety and stability.

EXPERIMENTAL MEASUREMENT CONDUCTED ON HONDA MTX 125 R MOTORCYCLE

In figure 1. the entire HONDA MTX 125 R motorcycle is shown, while in figures 2. and 3. is shown the way how device should be installed to the motorcycle frame.

Figure 1.

Figure 2.

In figures 4, 5 and 6. is presented a driver with already installed measurement device before the experiment started.

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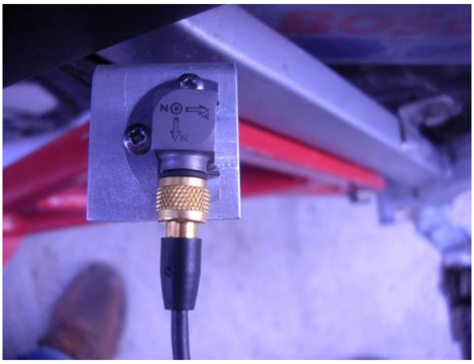


Figure 3.

Figure 4.

Figure 5.

Figure 6.

In the following figure 7. 1100m long test road where the experiment was performed is shown. The asphalt cover is made of good quality; the road is strait and even without longitudinal or lateral inclinations.

Figure 7.

Motorcycle was accelerated up to the certain speed. At the moment when desired speed was reached, that was observed directly on the instrument panel a device would be turned on and based on the marks previously set along the road, the acceleration along the road L=1100m was recorded.

The results are presented by direct transmission from device display to the paper (print screen). The marks given in the table have the following meaning:

- RMS – Root Mean Square
- MTVV – Maximum Transient Vibration Valve the highest RMS value during the measurement for the integration time 1sec.
- Peak
- CF- Peak/ RMS - crest factor
- F – factor of difficulty (is used during vibration measurement – body acceleration, in this case this factor is equal to 1)
- VTV – Vibration Total Value

DESCRIPTION OF MEASUREMENT EQUIPMENT–VIBRATION ANALYZER 4447

Human response to vibrations differs according to the frequency range of the vibration, amount of exposure and the point of contact. For power tool and vehicle operators, prolonged exposure to vibration can be harmful.

To protect workers, legislation such as EU Directive 2002/44/EC, sets forth minimum health and safety requirements for those exposed to risks arising from work-related vibration.

Figure 8: *Human Vibration Analyzer Type 4447*

Human Vibration Analyzer Type 4447 is a portable system designed for those who wish to monitor and reduce the exposure of potentially harmful vibration influences and ensure compliance to the EU directive. It is an effective and easy-to-use instrument that fulfils the international standard ISO 8041:2005, - Human response to vibration - Measuring instrumentation.

USES AND FEATURES

USES

- Hand-arm vibration measurements
- Whole-body vibration measurements
- Assessment of vibration exposure

FEATURES

- Compact, battery-powered instrument
- Four-button operation: Easy to use, ideal for field work, and can be operated using gloves
- Minimal cable connections: Only one transducer cable connection in the basic setup
- Triaxial and uniaxial measurements
- EU Directive parameters measured and displayed
- In-field assessment of vibration exposure all necessary data are displayed
- Simultaneous display of X, Y and Z axes. vibration, as well as total value
- USB connection: Transfer data to a computer for post-processing and archiving as well as battery charging
- Included PC software, 4447 Vibration Explorer
- BZ-5623, for data transfer, management and calculations on a PC

USER INTERFACE

Vibration analyzer Type 4447 can be operated easily with only four pushbuttons, as shown in Fig. 9.

Figure 9: *The four buttons on the front panel control the user interface of Type 4447*

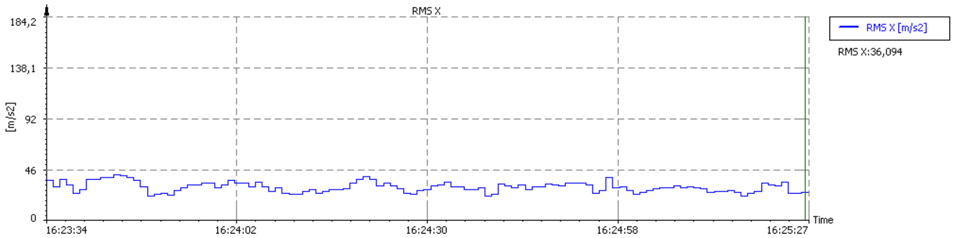
Type 4447.s graphical colour interface makes it easy to set up measurements and display results. Readings are by default in meters per second squared (m/s2), but can be displayed in g, dB re. µm/s2, m/s1.75 or g·s0.25.During a measurement, the results for the individual and combined axes are displayed. See Fig. 2 for an example of a measurement display. You can step through additional screen displays at any time during a measurement.

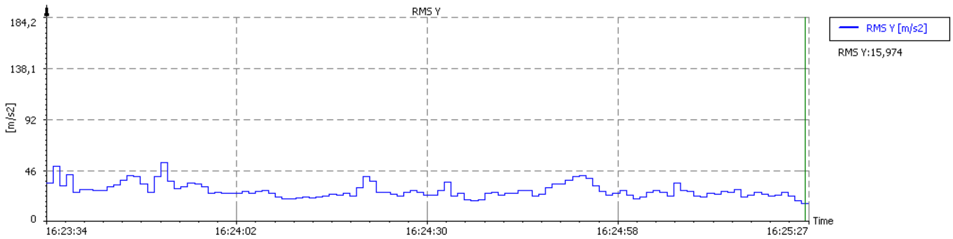
Results at the speed 30 km/h

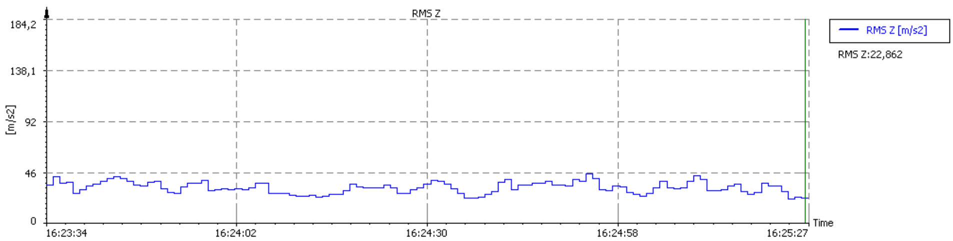
Table 1.

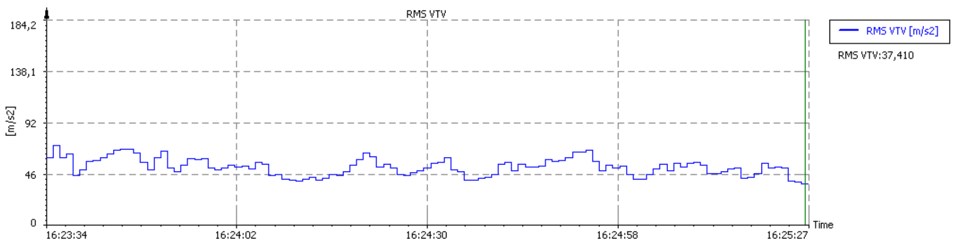
Name	Unit	X	Y	Z	VTV
RMS	[m/s2]	30,264	28,446	32,473	52,722
MTVV	[m/s2]	40,453	47,718	44,934	
Peak	[m/s2]	126,044	148,316	175,408	
CF		4,164	5,213	5,401	
Factor		1,00	1,00	1,00	
Overload		No	No	No	
Underrange		No	No	No	

Diagram 1.







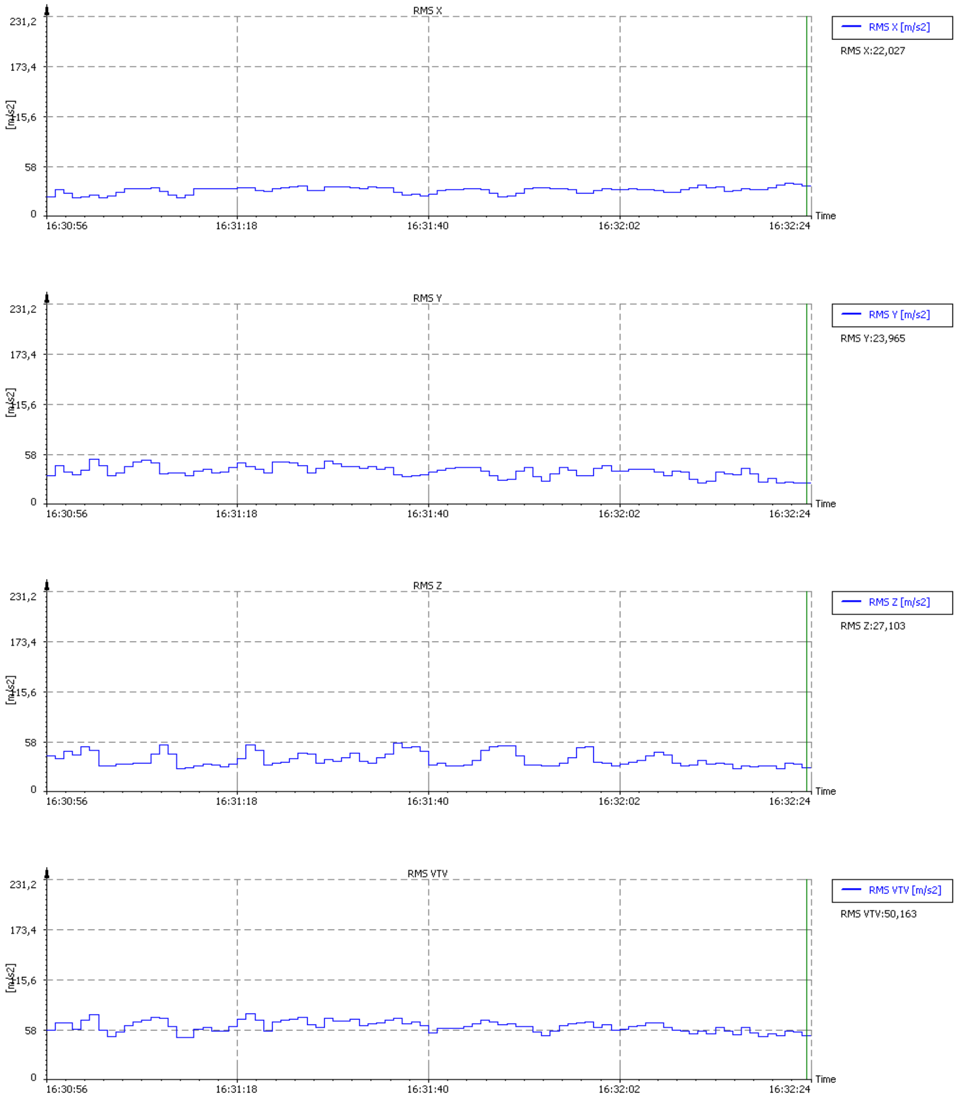


Results at the speed 50 km/h

Table 2.

Name	Unit	X	Y	Z	VTV
RMS	[m/s2]	30,119	38,264	37,819	61,657
MTVV	[m/s2]	37,658	49,212	53,824	
Peak	[m/s2]	118,179	159,552	220,146	
CF		3,923	4,169	5,820	
Factor		1,00	1,00	1,00	
Overload		No	No	No	
Underrange		No	No	No	

Diagram 2.

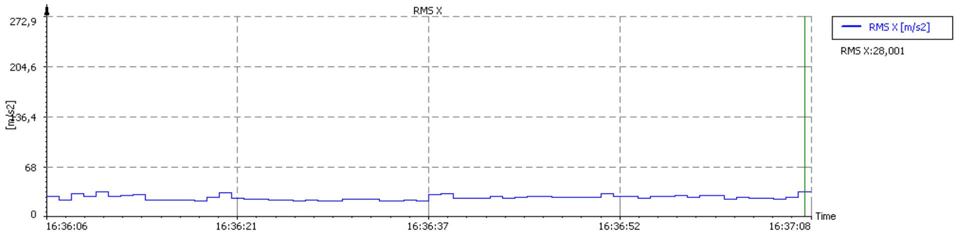


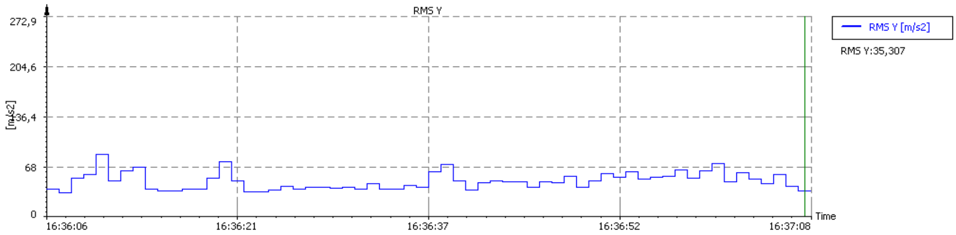
Results at the speed 70 km/h

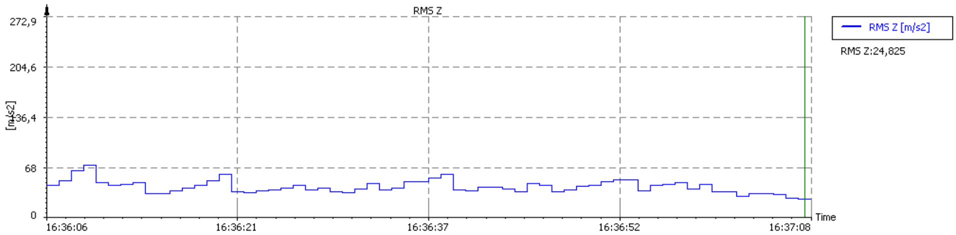
Table 3.

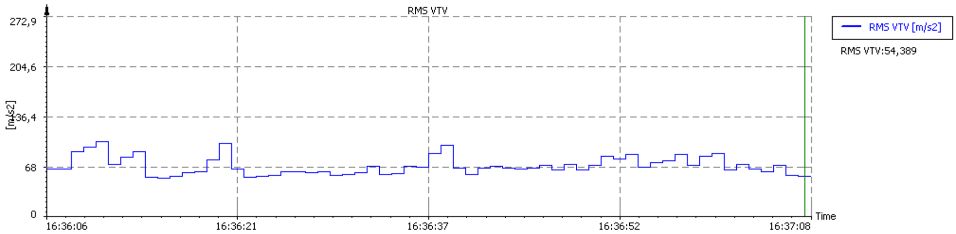
Name	Unit	X	Y	Z	VTV
RMS	[m/s2]	25,937	49,441	42,386	70,099
MTVV	[m/s2]	34,346	77,801	71,945	
Peak	[m/s2]	156,295	259,860	251,152	
CF		6,025	5,255	5,925	
Factor		1,00	1,00	1,00	
Overload		No	No	No	
Underrange		No	No	No	

Diagram 3.







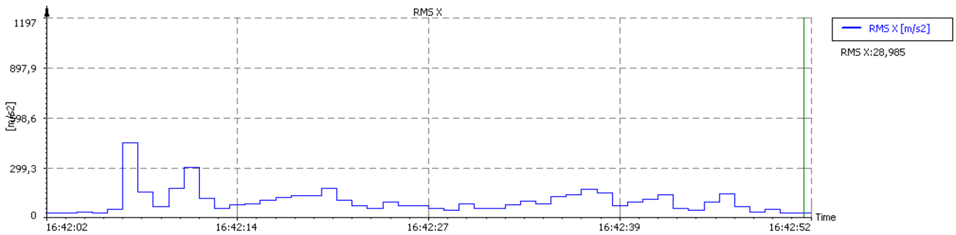


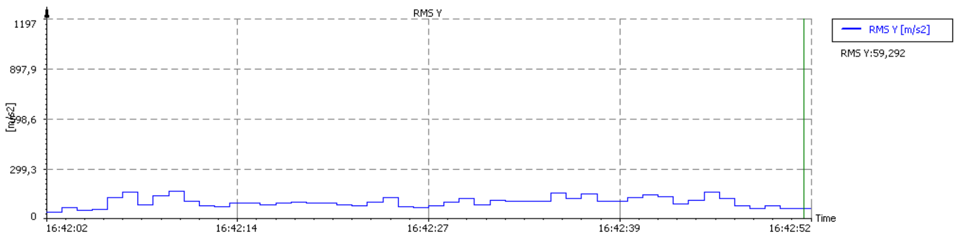
Results at the speed 90 km/h

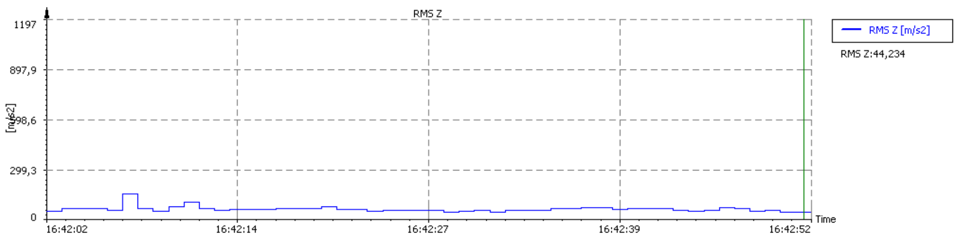
Table 4.

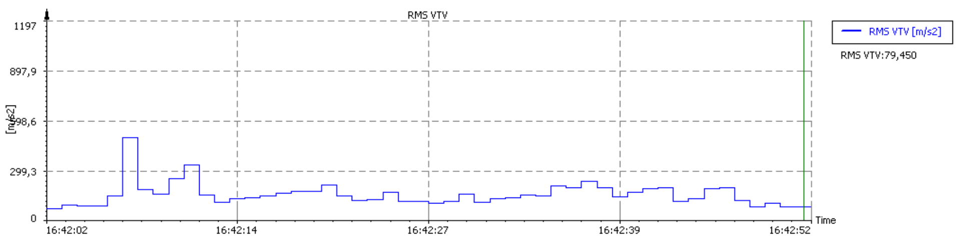
Name	Unit	X	Y	Z	VTV
RMS	[m/s2]	120,503	102,891	64,164	170,953
MTVV	[m/s2]	403,961	151,002	143,772	
Peak	[m/s2]	1140,248	415,123	487,023	
CF		9,462	4,034	7,590	
Factor		1,00	1,00	1,00	
Overload		No	No	No	
Underrange		No	No	No	

Diagram 4.







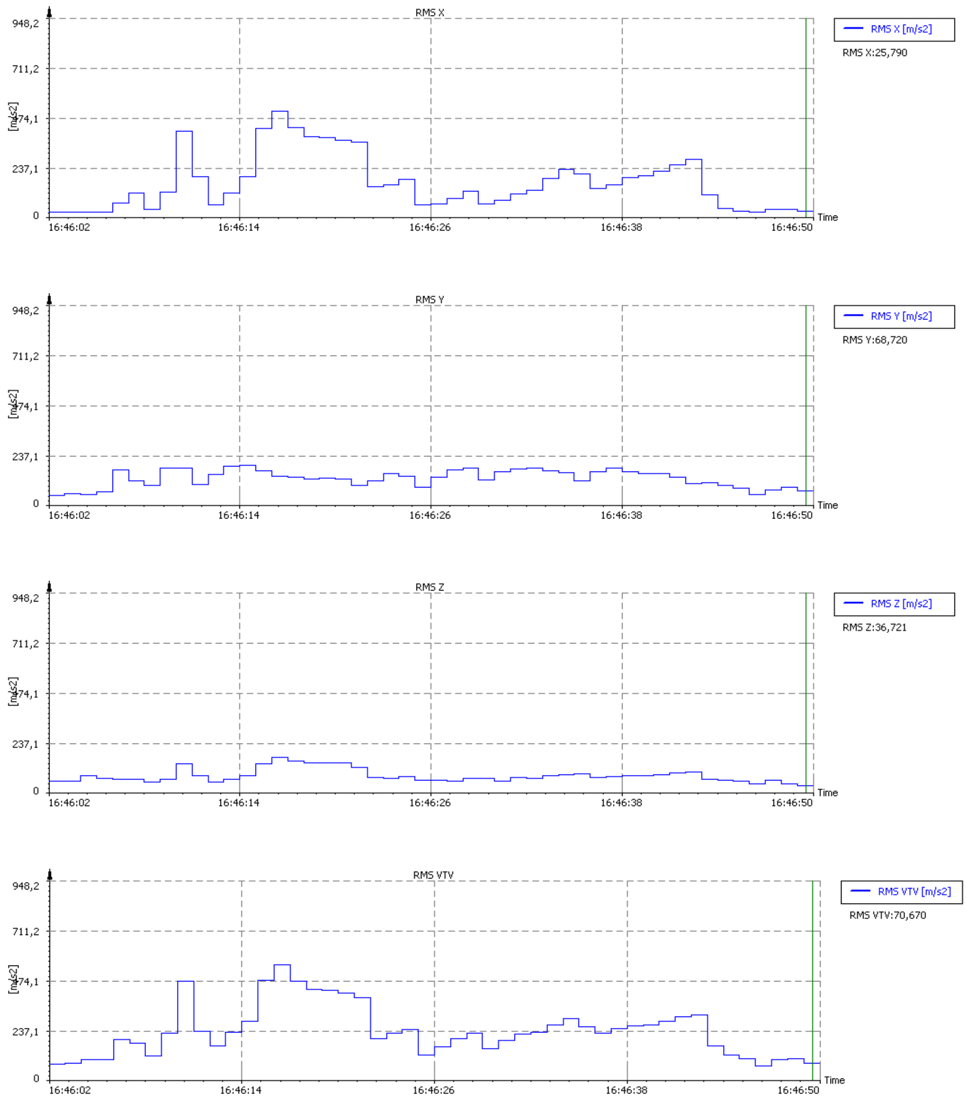


Results at the speed 110 km/h

Table 5.

Name	Unit	X	Y	Z	VTV
RMS	[m/s2]	208,751	134,426	87,027	263,099
MTVV	[m/s2]	460,810	185,903	155,154	
Peak	[m/s2]	903,082	444,949	489,238	
CF		4,326	3,309	5,621	
Factor		1,00	1,00	1,00	
Overload		No	No	No	
Underrange		No	No	No	

Diagram 5.

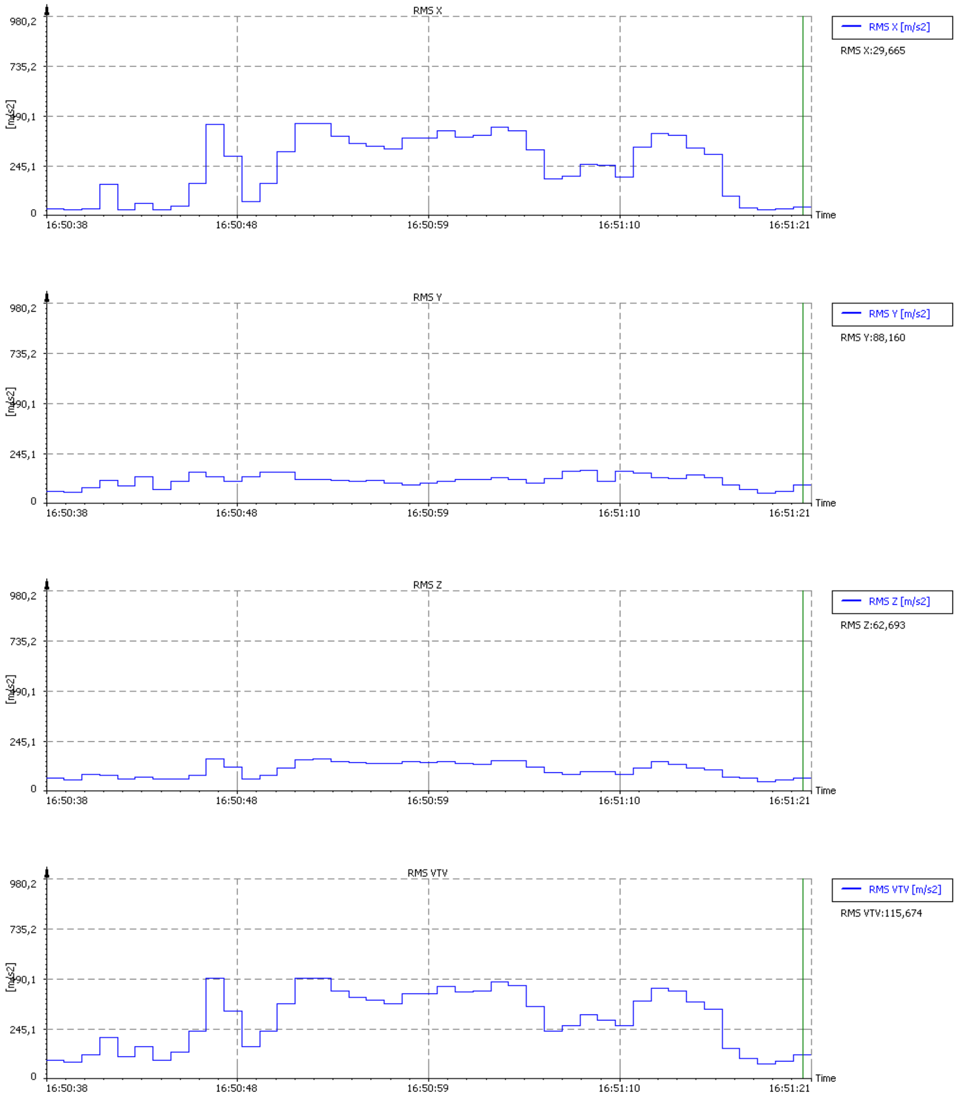


Results at the speed 130 km/h

Table 6.

Name	Unit	X	Y	Z	VTV
RMS	[m/s2]	284,232	114,140	108,265	324,865
MTVV	[m/s2]	438,575	154,949	151,731	
Peak	[m/s2]	933,535	395,511	558,559	
CF		3,284	3,465	5,159	
Factor		1,00	1,00	1,00	
Overload		No	No	No	
Underrange		No	No	No	

Diagram 6.



DYNAMIC REACTIONS

The simplified model based on which it's possible to calculate dynamic reactions is shown in the following figure.

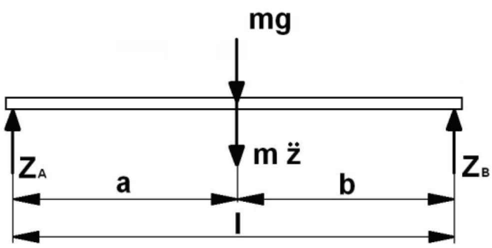


Figure 10.

Force equilibrium equation of all forces in "z" direction:

$$m \cdot (\ddot{z} + g) = Z_A + Z_B$$
$$\ddot{z}^* = \ddot{z} + g$$
$$m \cdot \ddot{z}^* = Z_A + Z_B$$

The torque equation at point "A" is:

$$m \cdot \ddot{z}^* \cdot a - Z_B \cdot l = 0 .$$

Based on these equations it's possible to calculate the values of dynamic reactions:

$$Z_A = \frac{m \cdot \ddot{z}^* \cdot b}{l} \text{ и } Z_B = \frac{m \cdot \ddot{z}^* \cdot a}{l}$$

where:

$\ddot{z}^*$ - acceleration measured along z – axis, Z_A - dynamic reaction of a front wheel and Z_B - dynamic reaction of a rear wheel.

Data necessary for calculation of forces are:

- motorcycle and driver's weight $m = 180kg$
- wheel base $l = 1{,}45m$ and
- center of gravity is defined by distances $a = 0{,}8m$ and $b = 0.65m$

Since the motorcycle was driven with the constant speeds: 30 km/h, 50 km/h, 70 km/h, 90 km/h, 110 km/h и 130 km/h, for each value will be calculated dynamic reactions at the beginning (I - turning device "on"), in the middle (II) and at the end of each measurement interval (III – turning device "off").

Motorcycle motion, same as specified interval points in which dynamic reactions are being calculated might be presented in the following diagram.

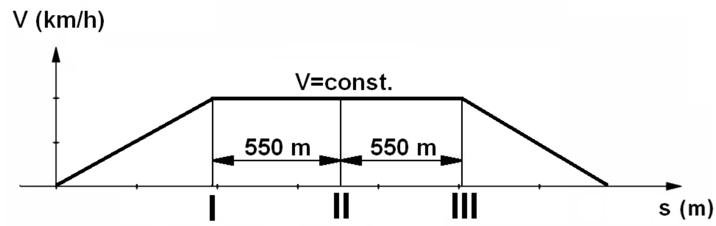


Figure 11.

Obtained results are presented in the following table:

		DYNAMIC REACTIONS (κN)					
	Speed (km/h)	30	50	70	90	110	130
	Interval duration (s)	1'53"	1'28"	1'02"	50"	48"	43"
I	Z_A	2.90	3.55	3.87	4.68	5.08	5.57
	Z_B	3.58	4.37	4.77	5.76	6.26	6.85
II	Z_A	3.15	3.95	4.19	5.00	5.24	7.67
	Z_B	3.87	4.87	5.16	6.16	6.46	9.43
III	Z_A	1.86	2.58	2.66	3.22	3.14	4.92
	Z_B	2.28	3.18	3.28	3.97	3.87	6.06

Obtained results could be presented by diagram.

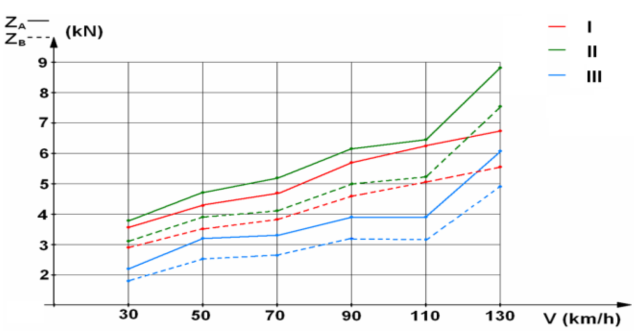


Diagram 7.

In order to make the analysis of dynamic reactions in longitudinal and lateral direction the equations of equilibrium should be settled based on figure 12.

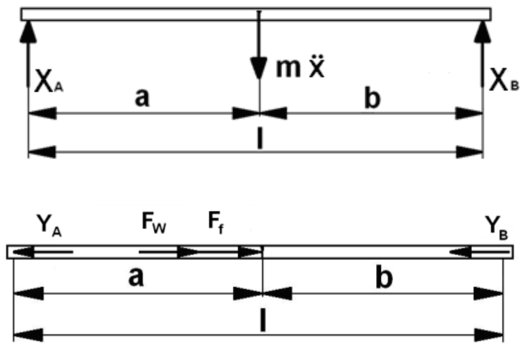


Figure 12.

This analysis, due to its complexity was not comprehended by this paper-work, but it might be the subject of some further researches.

COMMENTS ON THE RESULTS

Constant speed drive

Observing the recorded values during the constant speed drive, it can be noticed that the acceleration is present in all three directions - that means that the rolling effect is present, considering the fact that the device was placed on the motorcycle frame. These accelerations are more or less same up to the speed 90km/h when they became multiplied. This indicates the presence of resonant frequency of the frame at speeds higher than 90km/h.

In the measurement the influence of the road is also counted, i.e. the asphalt cover is not brand new, it is a bit rough and that impact to the measuring results. Also, the subjective behavior of driver has some influence to the results too. Further researches could be performed with the goal to eliminate these two effects – choosing some other, better test roads and some other drivers.

Dynamic reaction values

Based on the diagram where dynamic reaction values are shown, it might be concluded that within the interval and at the small speeds dissipation of the values is under 50%, while at higher speeds dissipation is about 70%.

The effect of a driver to the steering system and to dynamic reactions is highly expressed, because opposite to drive by car, where driver's position is much more stable, in a case of a motorcycle - driver's position has been constantly changing, because driver in this way keeps stability of a motorcycle.

REFERENCES

[1] Appel, H., Kramer, F. Morres, H.: Unfallforrschumg und Unfallmechanik, Institut fuer Kraftfahrzeugtechnik, TU Berlin 1990.

[2] Appel, H., Meissner, T. : Kraftfahrzeugtechnik I; Umdruck zur Vorlesung, TU Berlin 1998.

[3] Stoffregen Jürgen: Motorradtechik 6. Auflage Grundlagen und Konzepte von Motor, Antrieb und Fahrwerk – ATZ MTZ Wiesbaden 2006.

[4] Mukherjee, S. Chawia, A. Mohan, D.,Singh, M, Sakurai, M, Tamura, Y.: Motorcycle-Car Side Impact Simulation, JARI, Japan 2005.

[5] Willumeit, H.P.: Kraftfahrzeugdynamik, Umdruck zur Vorlesung, TU Berlin 1998.

[6] Janković, A., Simić, D.,. Joкović, S. Nikolić, B.: Matematical modeling of the frontal impact, Int. J. MVM Vol 20, Number 4, Decembar 1994, Kragujevac

[7] Hucher, S.,Smith, T.,Hermitte, T.: Vomputational Analzsis of Motorcycle Crach-Tests-a Basis for Motorcycle Accident Reconstructions, CEESAR; France 2007.

[8] Janković, A. Simić, D.: Bezbednost automobila, DSP Mecatronik, Kragujevac 2006.

[9] Aleksandrović B., Krstić G., Janković A.: Bezbednost motocikla pri kočenju, simpozijuim prevencija saobraćajnih nezgoda, Novi Sad, Oktobar 2006.

[10] Janković A., Aleksandrović B.: Motocikl danas u svetlu aktivne beybednosti, Naučno savetovanje na temu SAOBRAĆAJNE NEZGODE, Zlatibor, maj 2008,

[11] B. Aleksandrović,: “Neki aspekti aktivne bezbednosti motocikla“ – magistarska teza, Kragujevac, jul 2009.

[1] ONE CONTRIBUTION TO SYNCHROMESH TYPE GEARBOX AUTOMATION

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UDC: 629.113-585.3
621.83.069.2

Abstract

Solution of a car gearbox gear shifting automation with DC motor is presented in this paper. Gearbox shifting is performed by two types of motion, translatory and rotational. So far, mostly all solutions have had two DC motors. One DC motor would be for translation and and the other for rotation. Within this solution, both types of motion, are acquired by a single DC motor. This solution represents big material savings and of course simplification of entire system of automatic control. Converting of DC motors motion intomotion of gearboxes command lever is obtain through movement of a specially profiled plate.

Key words: manual gearbox, automation, DC motor, profiled spouts (plate).

MOGUĆE REŠENJE AUTOMATIZACIJE MANUELNOG MENJAČA

UDC: 629.113-585.3
621.83.069.2

Rezime: U radu je predstavljeno rešenje automatizacije promene stepeni prenosa, putem motora jednosmerne struje, manuelnog menjača na putničkom vozilu. Promena stepeni prenosa se vrši preko dva vida kretanja, translatornog i rotacionog. Sva dosadašnja rešenja su, uglavnom, bila sa dva elektromotora. Jedan elektomotor je zadužen za translaciju, dok je drugi zadužen za rotaciju. Kod ovog rešenja oba oblika kretanja, postižu se samo jednim motorom jednosmerne struje. Ovo rešenje predstavlja kako veliku uštedu u materijalu tako i pojednostavljenje celokupnog sistema automatskog upravljanja. Pretvaranje kretanja elektromotora u kretanje komandne poluge menjača je postignuto preko pomeranja specijalno profilisane ploče.

Ključne reči: manuelni menjač, automatizacija, elektromotor jednosmerne struje, profilisani otvori (ploča).

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ONE CONTRIBUTION TO SYNCHROMESH TYPE GEARBOX AUTOMATION

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INTRODUCTION

Synchromesh gearboxes gear shifting process is comprised out of several successive actions, such as: clutch engaging, selector levers movement, gears engaging/disengaging, gently releasing clutch pedal etc. This is only simplified display of what is actually happening during gear shifting process in a manual gearbox.

Because of the drive train configuration, command handle is almost never on a gearbox, but is actually connected to it by means of levers and joints. To engage or disengage a certain gear, this mechanism needs to move in a certain way. When divided in to simpler types of movement, lever mechanism movement is comprised out of two types of it, translatory and rotatory type.

Order in which this types of movement appear depends of a situation. All of this complicates driving process and contributes to drivers overall feeling of fatigue.

To resolve this problem, today we have automated gearbox systems, where drivers action are substituted with actions of actuators. In configuration like this small changes in drivers surroundings have been made, e.g. clutch pedal is gone, command handle, if there is one, now has only a few positions ("+", "-", N, R) and has totally different paths. Big changes have been made in gearbox surroundings. All actions between driver and gearbox are now replaced by actions of an actuators.

Main goal of this paper is to contribute further development of automated gearbox systems with one conceptual design and to present one of possible direction for future development.

GEAR SHIFTING AUTOMATION AT PRESENT

At present, solutions are based on two DC motors concept. Each of DC motors has its purpose. The first one is responsible for rotatory motion and the second one is responsible for the translatory motion.

One of the today's existing solution is presented in figure 1. System comprises out of several separate systems, all working together toward the same goal – automating the gear shifting process.

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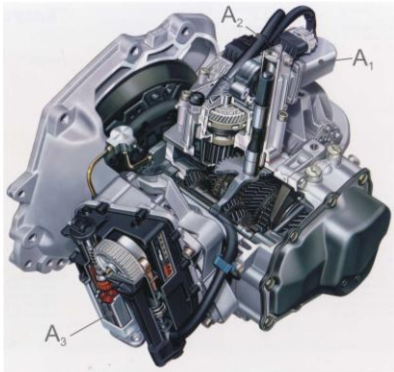

Figure 1: *Opel`s EasyTronic system*

As it can be seen in figure 1, this system is very compact and complex. All mechanical components are incorporated in gearbox housing and because of that, housing needed to be redesigned so all the parts could be easily mounted. This made the price goes up in compare to standard manual gearbox system.

Second disadvantage of this system is its complex software algorithm. Namely, system has to manages several different systems all at the same time. Among them are two DC motors which are need to be controlled sequentially and at the different speed rates and most of the times have to change rotation direction.

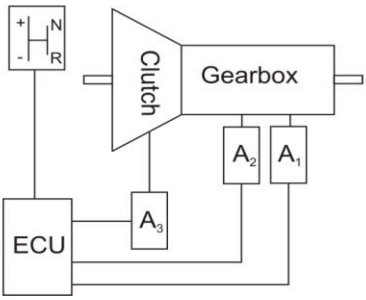

Figure 2: *Automated gearbox block scheme*

Automated gearbox system have tree actuators, marked as A_1, A_2 and A_3, in figure 1 and 2. A_1 and A_2 are DC motors and A_3 is clutch actuator. The clutch actuator unit is needed in any automated gearbox system with dry clutch system, so it can not be ignore in future constructions.

NEW DESIGN FOR GEAR SHIFTING AUTOMATION

The idea for only one DC motor for gearbox automation originates from gear shifting process in motorcycle gearboxes. Motorcycles often have a gearboxes with dog couplings, gears do not move axial and are engaged by one type of motion. Some gears are made with shaft out of one piece, while the rest of gears can freely rotate on shafts. All gears are coupled at all times, but when selector lever (fork) moves dog coupling engages particular pair of gears, see figure 3. To engage certain gear, driver moves command lever by foot or by hand and makes one type of movement, rotatory. Driver rotates cylinder which has channels on its surface, see figure 3. Selector forks are in conjunction with this cylinder by means of these channels.

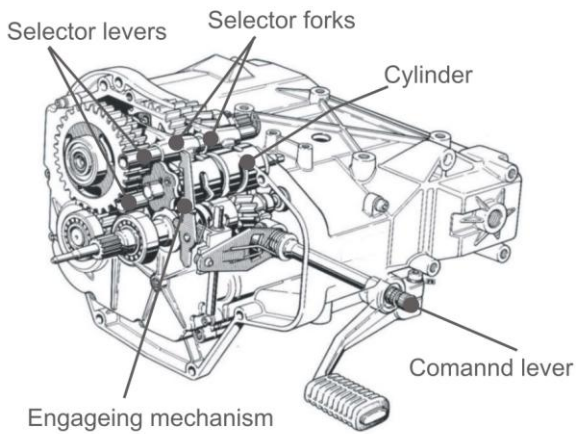

Figure 3: *Motorcycle gearbox*

If channel path is straight then there are no selector lever movement, therefore no change in gears. But if the path of the channel on the surface of the cylinder has a slope then one out of two things will happen, either gear will engage or it will disengage. So if a driver moves command lever, he is actually making rotatory movement by which he is moving, firstly, engaging mechanism, then cylinder and at the end dog couplings. Most important part of this system is cylinder with his channels, because it leads selector levers so they can engage gears.

DESIGN PRESENTATION

This one DC motor gearbox shifting mechanism is comprise from mechanical and electrical parts and in this paper only most important one's will be presented. Mechanical parts are: profiled plate, gear pair, cylinder with prongs, housing and flange. Electrical parts are: DC motor, DC motor controller, sensors, cables, command joystick, supporting electronics. The focus of this paper is at the mechanical part, because, choosing the DC motor and after that everything else is conditioned by hardware dimensions and construction.

Profiled plate is a key part of the system, because profiled plate is converting movement type. It is cuboid, dimensions varied by type of specific model of a gearbox. Main requirement for profiled plate dimensioning are stroke length and turning angle of the command lever. Beside dimensions of the profiled plate, more important thing is depth of its channels which are on the surface of the profiled plate, see figure 4.

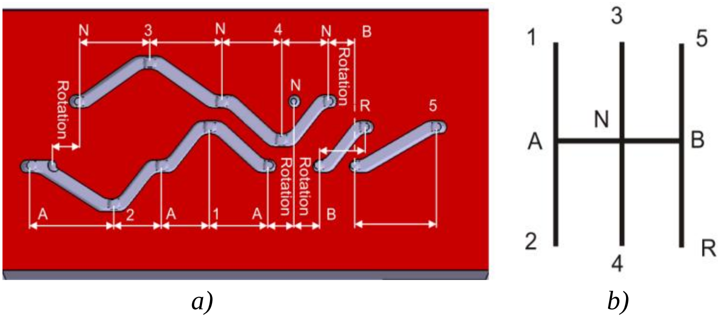

Figure 4: *a) Profile plate with gears displacement: 1…5 - number of a gear, N –neutral, A, B - neutral positions of a gearbox between two specific gears, b) H scheme of gearbox gear shifting process*

Layout of these channels is displayed on figure 4, as well as gear displacement. At the same figure, ordinary H scheme of gearbox gear shifting process is presented. In figure 4 rotation represents rotation of the cylinder with prongs i.e. moving gearbox command lever from neutral to neutral between two gears.

Length between two gears on the profiled plate directly proportional to stroke length of gearbox lever and is different for every gearbox model. Channels end positions are completely arbitrary but still very much depending of the type of a gearbox and they are also directly influencing DC motor speed change during gear shifting process. So it is better that this distances are equal in length, so that we could have constant DC motor speed at all time and at the end DC motor controller of a lower cost. End points span, between end points that are not jointed by the channel, is directly coupled with gearbox lever turning angle.

Cylinder with prongs is part of this mechanism that transform profiled plate translatory movement into command lever rotatory and translatory movement. There are three prongs, see figure 5. Prongs are joined together by a supporting cylinder. Supporting cylinder have two bases on which two shafts are welded, one on each base.. These shafts are supported by slide bearings in the way represented in figure 5, and bearing are embedded in the housing, later on. Slide bearings allow rotatory and translatory movement through them as well.

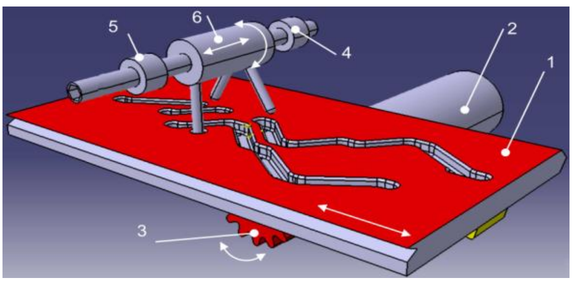

Figure 5: *Gear box shifting mechanism with single DC motor: 1 – profiled plate, 2 – DC motor, 3 – sprung gear on DC motor, 4, 5 – sliding bearings, 6 – cylinder with prongs*

When profiled plate moves translatory (left or right), cylinder is rotating and moving back and forward because one prong is always coupled with profile plate and is following path of the channel. In figure 6, cross section of the mechanism is presented.

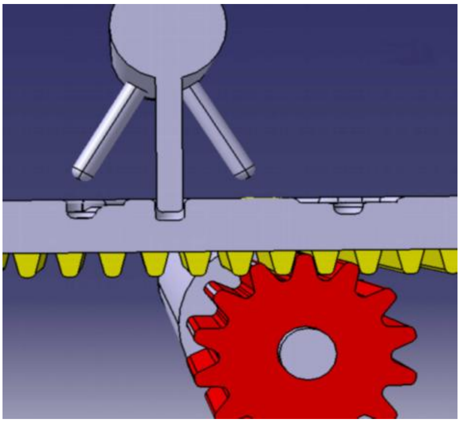

Figure 6: *Gearbox shifting mechanism cross section*

DESIGN ANALISYS

Profiled plate is a single metal piece that not necessary needs to be a plate. Older solutions similar to the profile plate tend to be round in shape. So now we have cylinder and disks with similar cannel displacement on surface of them. As presented in figure 6, prongs are always in contact with profile plate and are following channels paths, but there is a moment of time when switching in which prong is currently coupled with the plate. At this moment two prongs are coupled with the plate at the same time. Because of the cylinder rotatory movement one prong is exiting the channel while the other is entering it at the same time. At this point it is out of most importance the profile of top end of the channels, end on which prongs are sliding while entering or exiting the channel. This radius depends out of many factors, from which most importants are length of the prongs, rotatory angle of gearbox (synchromesh engaging stoke length, etc). These radiuses need to follow exiting/entering trajectory of the prong, so the resistance would be as less as possible. Channels path can vary from model to model of a certain gearbox, because of their difference in construction, so it is very important as well to plan carefully the paths when designing new profile plate.

Number of gears determents number of prongs on cylinder. Maximal number, which one prong can cover is two if shifting pattern is H type. So if there are five gears plus reverse that means that there are six gears to be covered with prongs, so that means tree prongs. If a vehicle would have six gears plus reverse, that would mean four prongs on the cylinder, etc. DC motor for this kind of application has to have big momentum and relatively low speed (speed is depending of the length of the profiled plate). Therefore the DC motor must have one suitable reducer, that reduce the motor speed and increases its torque. Needed ratio for suitable DC motor is around 200 or 300, because, DC motors with brushes usually runs at speeds of 2000 or 3000 rpm, and we need speed of 10 to 20 rpm.

There are many complete solutions for this type of application on market. Most important characteristics for choosing the right DC motor are nominal speed [min^{-1}], nominal torque [Nm], nominal output power [W], nominal current [A], starting torque [Nm] and mass [kg].

When completed, new system have less moving parts and less electrical ones. Block scheme is presented in figure 7. A_1 and A_2 are the two only actuators on the system. The A_1 is clutch engaging/disengaging actuator, and the A_2 is DC motor which moves profile plate.

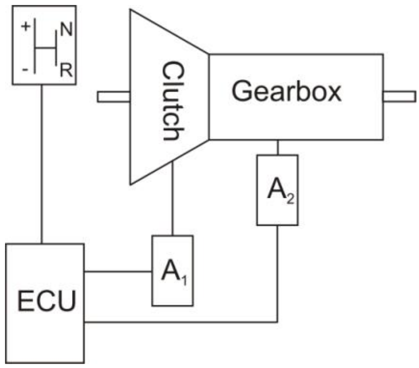

Figure 7: *Block scheme of the single DC motor gearbox shifting system*

The most important improvement of this system is that, that in comparing to the other solutions, this one is detachable from the gearbox. This system is only attached to the already existing system of transmission and it is not necessary to make any kind of changes to the gearbox housing, Another interesting feature of this single motor gearbox shifting mechanism is that it can be mounted under any angle as long as the command lever of gearbox and output shaft, of cylinder with prongs, stays coaxial.

Possible spots for embedding this system are vast, because this system is not limited with points of its connection with gearbox, except of command lever. This means that this system can be built in under any given angle as long as his cylinder shafts and command lever stays coaxial. Empty spots left after building in system like this on can be used to put a new command handle in a drivers cabin or it can even be pushed away from the gearbox if there is a straight corridor between them.

CONCLUSION

This paper presents actuator for gearbox shifting automation in automobiles with synchromesh gearboxes, that operates with one DC motor. This actuator represents upturn in comparison to today's two DC motors solution, because the system is simplifying. Mechanical part of a system comprises out of a profiled plate, cylinder with prongs, gear pair, DC motor and housing. Because of a small number of parts and simplicity, this mechanism is suitable for production and maintenance. Modular installation is possible on different types of manual gearboxes, with small necessary modifications such as changes in the channels paths, gear pair ratio, prongs lengths etc.

At present, described actuator does not exists in automobile production, but it seems worth researching it's potentials.

In course of future development, kinematics and dynamics of the system should be investigate as well as the possibility to replace profiled plate with profiled cylinder, which will simplify the presented system even more.

REFERENCES

[1] Bauer H. : Kraftfarhtechnisches Taschenbuch/Bosch, VDI-Verlag, 22 Auflage, Düsseldorf, 1995.

[2] Bauer H. : Automotive Electrics Automotive Electronics / Bosch, Robert Bosch GmbH, 2004.

[3] Braess H., Seifert U. : Vieweg Handbuch Kraftfarhzeugtechnik, Vieweg – Verlag, 2.Auflage, 2001.

[4] Fisher R., Hirt G. :Integrating of the Auto Shift Gearbox with the electrical machine, LuK GmbH 1999.

[5] Happian – Smith J. : An Introduction to Modern Vehicle Design, Butterworth – Heinemann, 2002.

[6] Isermann R. : Mechatronic Systems Fundamentals, Springer – Verlag London Limited, 2005.

[7] Lechner G., Naunheimer H. : Automotive Transmissions – Fundamentals, Springer 1999.

[8] Legg A. K., Gill T. : Vauxhall/Opel Corsa - Service and Repair Manual, Haynes 2003

[9] Milenkovic P., Vitosevic N. :Automated manual transmission for "zastava" vehicles, The second international conference "power transmissions `06", 2006

[10] Reif K. : Automobile – Electronic, Vieweg – Verlag, 2006.

[11] Self - Study Programme 221, Electronic Manual Gearbox, Design and Function, Volkswagen AG, Wolfsburg 1999.

[12] Self - Study Programme 237; Manual Gearbox. Design and Function, Volkswagen AG, Wolfsburg 2000.

[13] Wallentowitz H. ,Reif K.: Handbuch Kraftfahrzeugelectronik, Vieweg – Verlag, 2006.

[1] MESH TYPE ANALYSIS FOR SIMULATING CAVITATION PHENOMENA IN INJECTION NOZZLE: THEORETICAL AND NUMERICAL ANALYSIS

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UDC: 621.43:519.6

Abstract

The evaporation of diesel fuel in the injection nozzle significantly influences the injection characteristics and spray formation process. In present paper the influence of different type of mesh and density of mesh on the cavitation phenomena is analyzed. The theoretical backgrounds of cavitation occurrence presented in the first part of the paper are followed by the numerical analyses of two-phase flow (liquid and gas phase) and single phase flow. The numerical analysis is made for two different type of fluid diesel (D2) in biodiesel (B100) using computation fluid dynamic (CFD) program FIRE. Numerical analysis also includes different type and density of meshes and their influence on results. The two-phase flow is analyzed using the two-equation approach, where all conservation equations are solved for every phase. The results are compared for different meshes and different types of fluid (D2 and B100). The result show how important is the structure of used mesh and its density.

Key words: two-phase flow, cavitation, injector nozzle, CFD, nodalisation analysis.

ANALIZA TIPA MREŽE U SVRHU SIMULACIJE POJAVE KAVITACIJE U MLAZNICI BRIZGAČA: TEORETSKA I NUMERIČKA ANALIZA

UDC: 621.43:519.6

Rezime: Isparavanje dizel goriva u mlaznici brizgača značajno utiče na karakteristike ubrizgavanja i proces formiranja mlaza. U ovom radu se analizira uticaj različitih tipova i gustina mreža na proces stvaranja kavitacije. Prikazane su osnove nastanka kavitacije, navedene u prvom delu rada, potom sledi numerička analiza dvofaznog (tečnost i gas) i jednofaznog protoka. Numerička analiza se odnosi na dva različita fluida, dizel gorivo (D2) i biodizel (B100) koristeći kompjuterski CFD program FIRE. Numerička analiza takođe uključuje različite tipove i gustine mreže i njihov uticaj na rezultat. Dvofazni protok smo analizirali koristeći pristup sa dve jednačine u kojima se jednačine održavanja rešavaju za svaku fazu. Upoređivali smo razultate za različite tipove mreže i gustine goriva (D2 i B100). Rezultati ukazuju na značaj strukutre mreža i njihove gustine.

Ključne reči: dvofazni protok, kavitacija, mlaznica brizgača, CFD, analiza mreže.

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MESH TYPE ANALYSIS FOR SIMULATING CAVITATION PHENOMENA IN INJECTION NOZZLE: THEORETICAL AND NUMERICAL ANALYSIS

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UDC: 621.43 :519.6

INTRODUCTION

The development of the modern compression ignition engine is mainly connected with rising of injection pressure and the possibility of the injecting several jets during the single injection cycle. Both modifications influence positively the engine characteristics and the emission formation processes. On the other side rising of the injection pressure results in higher flow velocities in the nozzle hole channels and in evaporation of fuel in the nozzle holes. The fuel evaporates on the sharp edge at the nozzle inlet, where the static pressure falls below the fuel vapour pressure. The vapour is spreading along the nozzle hole and could also reach the outlet. The evaporation of fuel and cavitation process in the nozzle hole significantly influence the in-nozzle flow and spray formation process.

THEORETICAL BACKGROUNDS

Cavitation bubbles form because of low static pressure that occurs near a sharp inlet corner in the nozzle flow. If the corner of the inlet is sufficiently sharp, the flow tends to separate and form a contraction inside the nozzle, which reduces the area through which the liquid flows. This reduced area is accompanied by increase in velocity, as predicted by conservation of mass. Conservation of momentum predicts that the acceleration of the liquid through the vena contracta causes a pressure depression in the throat of the nozzle. The low pressure inside the throat of the nozzle may fall below the vapour pressure of the liquid, causing cavitation. Cavitation flow does not, however, strictly adhere to this simple idealization. The formation of the bubbles is sensitive to the geometry of corner and any imperfections in the nozzle shape. The cavitation is also very sensitive to the quality of the liquid. Furthermore, cavitation inception may occur at pressure below the vapour pressure. Another complication is that the cavitating flow is transient and fully three-dimensional. The location of the vapour is not steady and it is usually also not symmetrical.

Nozzle geometry

Analyses were made for one-hole nozzle with sac volume and sharp edges at the nozzle hole inlet side. Dimension of the tested nozzle are presented in Table 1.

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Table 1: *Nozzle dimensions*

Nozzle hole diameter	d_d	*0,68 mm*
Nozzle hole channel length	l_s	*1 mm*
Sac chamber diameter	D_E	*1,5 mm*
Needle seat diameter	D_A	*1,36*
Needle tip cone angle	α	*120°*
Needle seat cone angle	σ	*60°*
Maximal needle lift	h_{mx}	*0,30 mm*

Flow coefficient definitions

Flow coefficient, despite its simplicity, represent one of the most important values, representing the fuel injection conditions at the nozzle. It is defined as ratio between the measured or real (V_{real}) and theoretical (V_{th}) volume flow injected through the nozzle. According to Bernoulli equation, the theoretical outflow velocity can be derived from the pressure difference (Δp) and fuel density (ρ):

$$\mu = \frac{\dot{V}_{real}}{\dot{V}_{th}} = \frac{\dot{V}_{real}}{A_d \cdot \sqrt{\frac{2 \cdot \Delta p}{\rho}}} \qquad (1)$$

A_d represents the sum of the nozzle hole cross-section area, while ρ is the density of the fluid. Following the presented equations, higher flow coefficient values result in bigger quantity of fuel injected per time unit, higher outflow velocities and better fuel spray atomization.

NUMERICAL ANALYSIS

Numerical analyses were made using the CFD program FIRE. Two-phase flow is calculated with two-equation model, with a continuous liquid and a dispersed vapour phase.
Mass conservation equation:

$$\frac{\partial \alpha_k \rho_k}{\partial t} + \nabla \cdot \rho_k \mathbf{v}_k = \sum_{l=1;l\neq k}^{N} \Gamma_{kl} \qquad (1)$$

α_k is volume fraction of phase k, v_k is phase k velocity, and Γ_{kl} presents the interfacial mass exchange between phases k and l. The compatibility condition must be fulfilled:

$$\sum_{k=1}^{n} \alpha_k = 1 \qquad (2)$$

Momentum conservation equation:

$$\frac{\partial \alpha_k \rho_k \mathbf{v}_k}{\partial t} + \nabla \cdot \rho_k \mathbf{v}_k \mathbf{v}_k = -\alpha_k \nabla p + \nabla \alpha_k \left(\tau_k + T_k^{'}\right) + \\ + \alpha_k \rho_k \mathbf{g} + \sum_{l=1;l\neq k}^{N} \mathbf{M}_{kl} + \mathbf{v}_k \sum_{l=1;l\neq k}^{N} \Gamma_{kl} \tag{3}$$

where f is the body force vector which comprises of gravity g; M_{kl} represents the momentum interfacial interaction between phase k and l, and p is pressure. Pressure is assumed identical for all phases. The phase k shear τ_k, equals:

$$\tau_k = \mu_k \left[(\nabla v_k + \nabla v_k^t) - \frac{2}{3} \nabla \cdot v_k \right] \tag{4}$$

μ_k is molecular viscosity. Reynolds stress T_k^t equals:

$$T_k^t = -\rho_k \overline{v_k^{'} v_k^{'}} = \mu_k \left[(\nabla v_k + \nabla v_k^t) - \frac{2}{3} \nabla \cdot v_k \right] - \frac{2}{3} \nabla \cdot v_k I \tag{5}$$

Turbulent viscosity is modelled as:

$$\mu_k^t = \rho_k C_\mu \frac{k_k^2}{\varepsilon_k} \tag{6}$$

Since the analyses in the present paper were made at constant temperature the enthalpy conservation equation will not be described in the details here.

Turbulent kinetic energy (TKE) conservation equation:

$$\frac{\partial \alpha_k \rho_k k_k}{\partial t} + \nabla \cdot \rho_k v_k k_k \nabla \cdot \alpha_k \left(\mu_k + \frac{\mu_k^{'}}{\sigma_k} \right) \nabla k_k + \alpha_k P_k + \alpha_k P_{B,k} - \\ - \alpha_k \rho_k \varepsilon_k + \alpha_k \frac{dp}{dt} + \sum_{l=1;l\neq k}^{N} K_{kl} + k_k \sum_{l=1;l\neq k}^{N} \Gamma_{kl} \tag{7}$$

Turbulence dissipation equation (TED):

$$\frac{\partial \alpha_k \rho_k \varepsilon_k}{\partial t} + \nabla \cdot \rho_k v_k k_k \nabla \cdot \alpha_k \left(\mu_k + \frac{\mu_k^{'}}{\sigma_k} \right) \nabla \varepsilon_k + \alpha_k C_1 P_k \frac{\varepsilon_k}{k_k} + \alpha_k C_2 \rho_k \frac{{}_k\varepsilon_k^2}{k_k} + \\ + \alpha_k C_3 \rho_k max\left(P_{B,k}, 0\right) \frac{\varepsilon_k}{k_k} - \alpha_k C_4 \rho_k \varepsilon_k \nabla \cdot v_k + \sum_{l=1;l\neq k}^{N} D_{kl} + \varepsilon_k \sum_{l=1;l\neq k}^{N} \Gamma_{kl} \tag{8}$$

In two-equation model several mass exchange terms could be introduced (according to the type of two-phase flow). In the present analysis the cavitation model is employed.
Mass exchange term is defined with following equations

$$\Gamma_c = \frac{1}{C_{CR}} sign(\Delta p) \cdot 3{,}85 \cdot \frac{\rho_d}{\sqrt{\rho_c}} N'''^{1/3} \alpha_d^{1/2} |\Delta p|^{1/2} \tag{9}$$

$$\Gamma_d = sign(\Delta p) \cdot 3{,}85 \cdot \frac{\rho_d}{\sqrt{\rho_c}} N'''^{1/3} \alpha_d^{1/2} |\Delta p|^{1/2} \tag{10}$$

Where the effective pressure difference equals:

$$\Delta p = p_{sat} - \left(p - C_E \frac{2}{3} \rho_c k_c\right) \tag{11}$$

C_E is the Egler coefficient, which varies between 1 and 1,4. C_{CR} is the condensation reduction factor. N''' is the bubble number density, which is calculated with assumed diminishing linear ramp:

$$N''' \begin{Bmatrix} N_0''' & \alpha_d \leq 0{,}5 \\ 2(N_0''' - 1)(1 - \alpha_d) + 1 & \alpha_d > 0{,}5 \end{Bmatrix} \tag{12}$$

Where N_0''' represents the initial value of the bubble number density.
Interfacial momentum exchange term is defined with:

$$M_c = C_d \frac{1}{8} \rho_c A_i''' |v_r| v_r + C_{TD} \rho_c k_c \nabla \alpha_d = -M_d \tag{13}$$

Numerical models

Numerical analyses were taken by using the CFD program FIRE.

Computation model

To analyse the flow characteristics of the in-nozzle flow different nozzle models were made. Since some analysis shown, that the pressure drop in nozzle is significant only in the area of the needle seat, sac chamber and nozzle holes, the meshes were modelled only for the above mentioned parts. For the maximal needle lift of 0,35 mm, two different nozzle models, representing real size and one half of the nozzle were made. First analyses show no significant changes between the results of the real size and one half model. For this reason we used one half model in further research.

Initial and boundary conditions

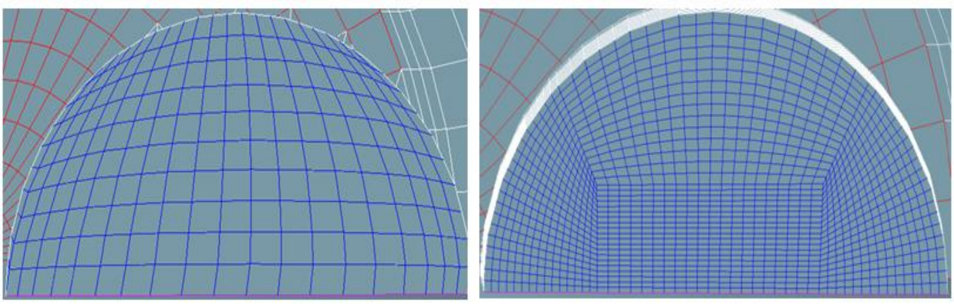
When analysing steady state flow, pressure boundary conditions at the inlet and outlet are specified. The fluids used for analysis are the diesel D2 and biodiesel B100, with the temperature of 293,15 K, the density 825 kg/m^3 for D2 and 875 kg/m^3 for B100 and dynamic viscosity of 2,45 mm^2/s for D2 and 2,6 mm^2/s for B100. K-ε turbulence model is employed. In all simulations the fuel was considered to be incompressible. Since the maximal velocities of fuel are much smaller (less than 50%) than the velocity of sound we believe this assumption is correct.

Used types of meshes and their various density

Numerical analysis includes various densities and types of meshes. Densities and their related type of used meshes are presented in Table 2 in Figure 1. Standard approach is to use block-structured types of meshes, while we had a chance to compare (time) it with structured, to study the differences between them.

Table 2. *Various densities and related types of meshes*

Name	Number of used elements	Type
mesh 1	21.440	structured
mesh 2	41.920	structured
mesh 3	162.050	structured
mesh 4	241.240	structured
mesh 5	376.960	structured
mesh 6	610.750	structured
mesh 7	39.000	block- structured
mesh 8	95.000	block- structured
mesh 9	319.360	block- structured
mesh 10	505.760	block- structured



A B

Figure 1: *Structured (A) and block-* structured *(B) types of used meshes*

RESULTS

Results were taken on two different positions on the injector hole. The positions are presented on the Figure 2.

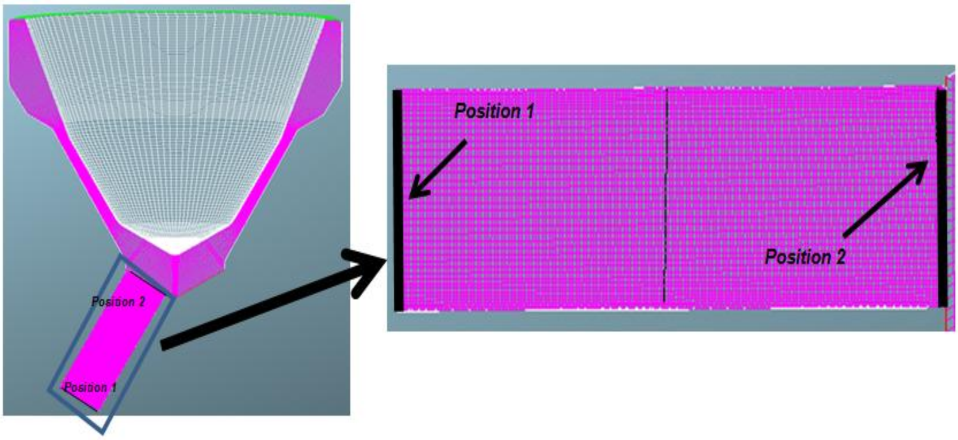


Figure 2: *Positions of point were the result are taken*

Velocity profiles and volume fraction distributions derived from the CFD analyses for different meshed models are presented on following figures. Figures 3-6 show the results of volume fraction distribution in nozzle hole with various mesh densities.

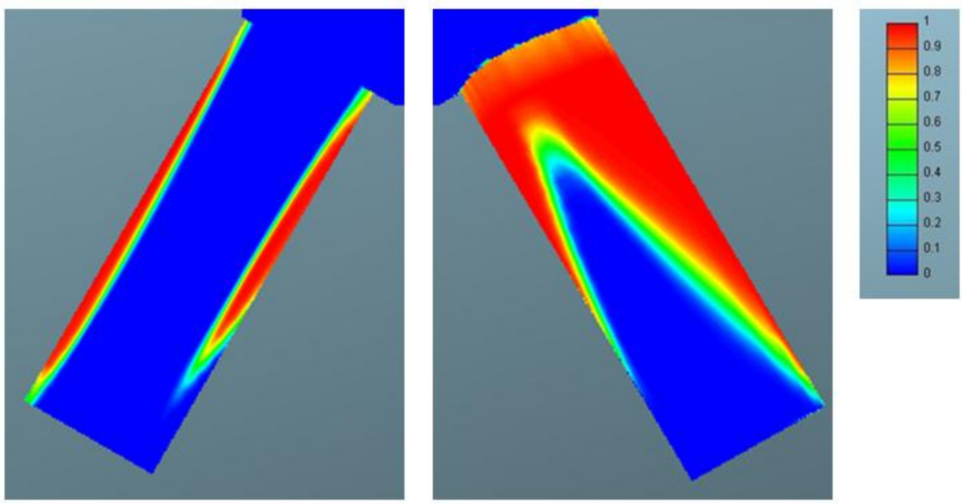


Figure 3: *Volume fraction (mesh 7)*

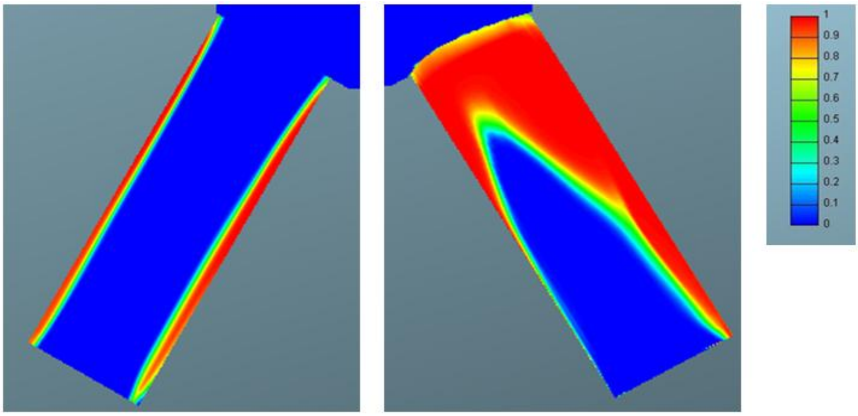


Figure 4: *Volume fraction (mesh 8)*

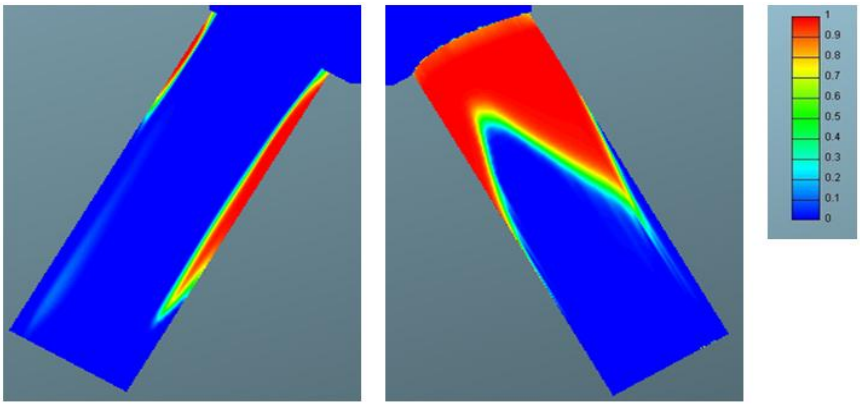


Figure 5: *Volume fraction (mesh 9)*

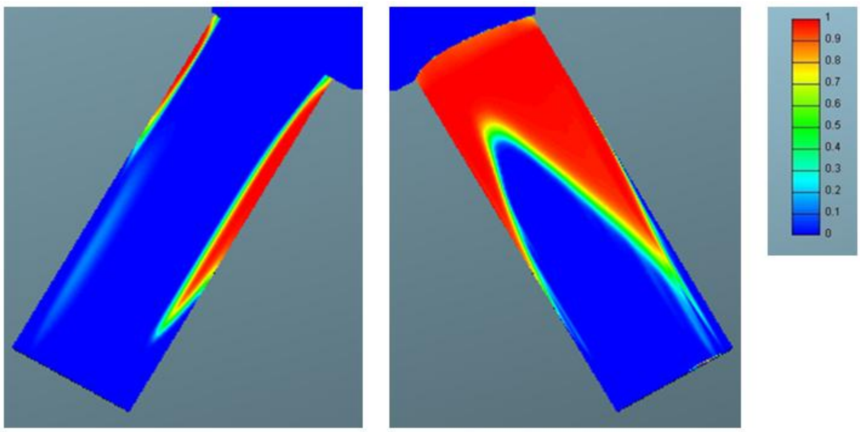


Figure 6: *Volume fraction (mesh 10)*

The velocity profiles and volume fraction distributions in nozzle holes with various densities, figures 3-6 are almost identical. The results indicate already known fact that the outflow velocity is higher at holes with smaller inclination angles, what results in higher flow coefficient at those holes.

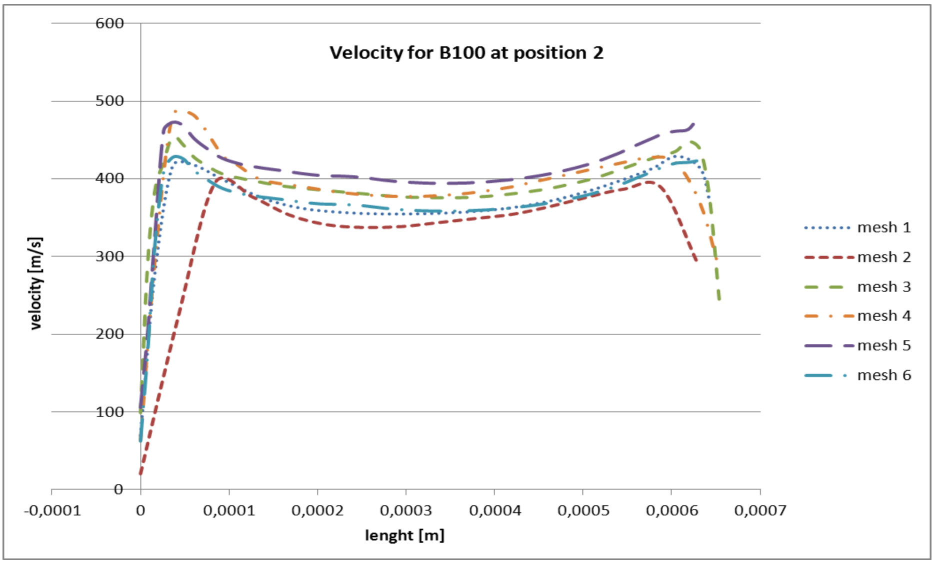


Figure 7: *Velocity profiles for structured type and various densities of used meshes at position 2*

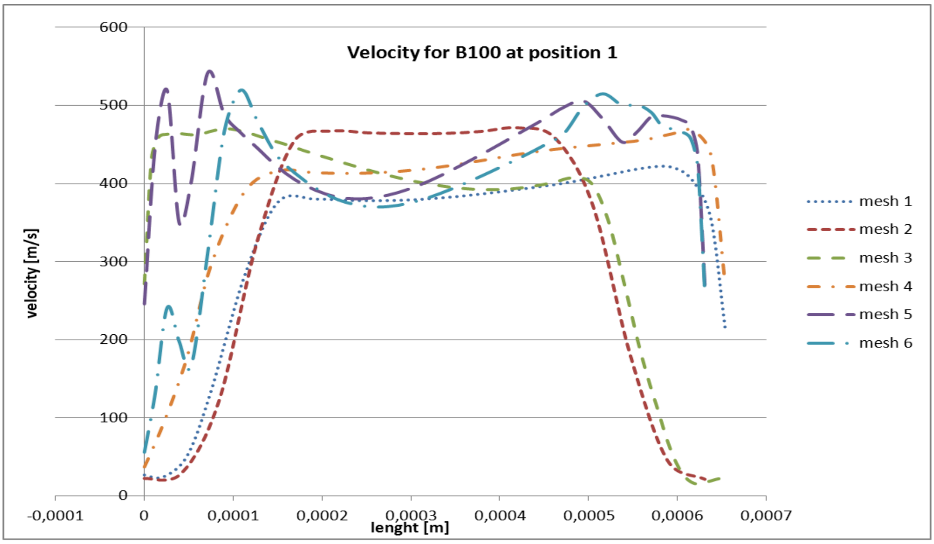


Figure 8: *Velocity profiles for structured type and various densities of used meshes position 1*

The results of the numerical analysis for velocity profiles for structured type and various density are presented on figures 7 and 8, while figure 9 shows the results for block-structured type. From figures 10-12 it is obvious that the numerical result of the volume fraction for structured and block-structured are not significantly different.

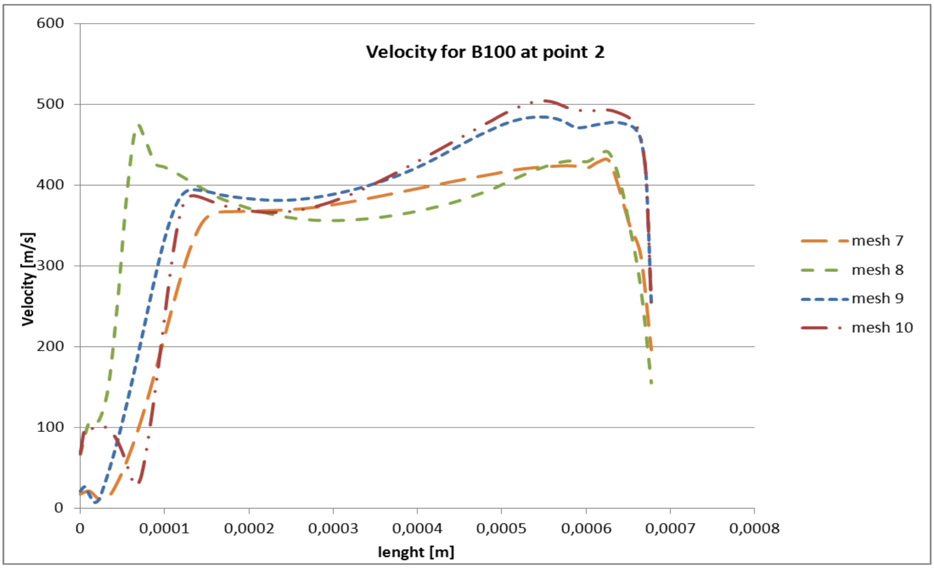


Figure 9: *Velocity profiles for block-structured type and various densities of used meshes position 1*

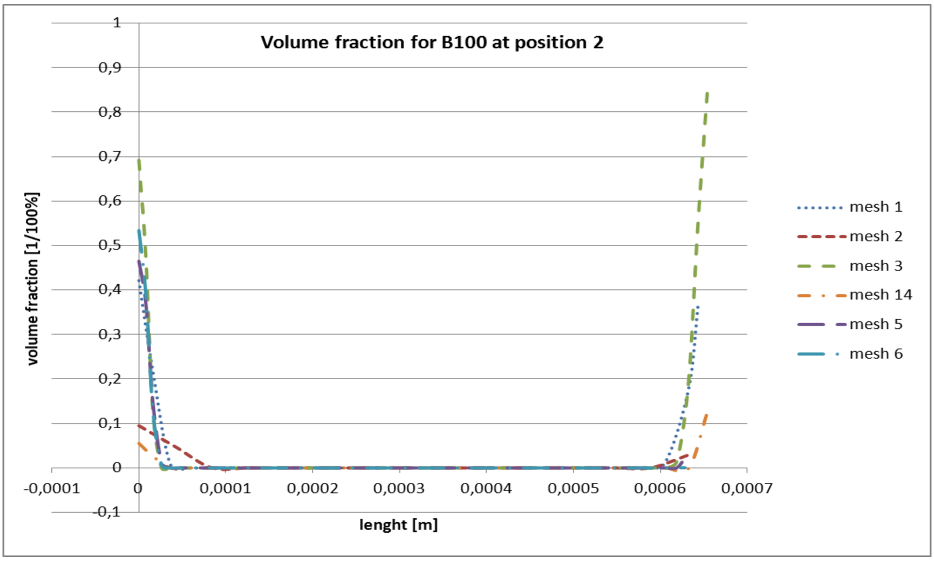


Figure 10: *Volume fraction for structured type and various densities of used meshes position 2*

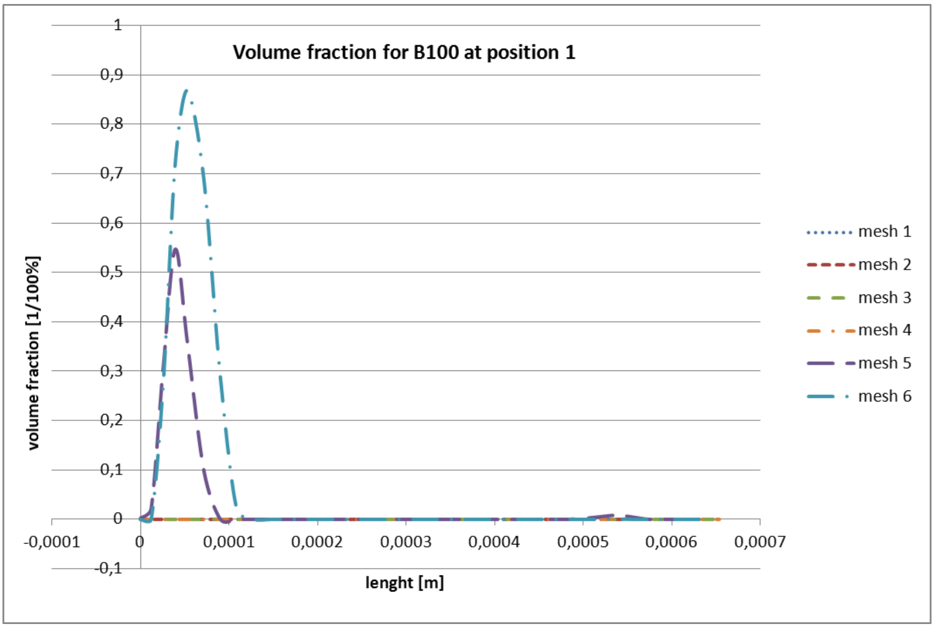


Figure 11: *Volume fraction for structured type and various densities of used meshes position 1*

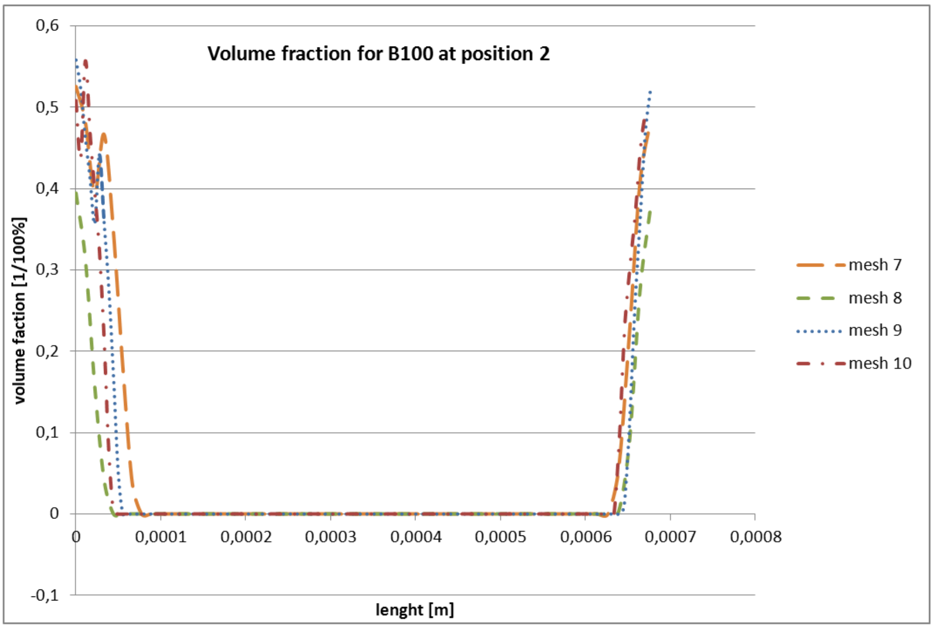


Figure 12: *Volume fraction for block-structured type and various densities of used meshes position 2*

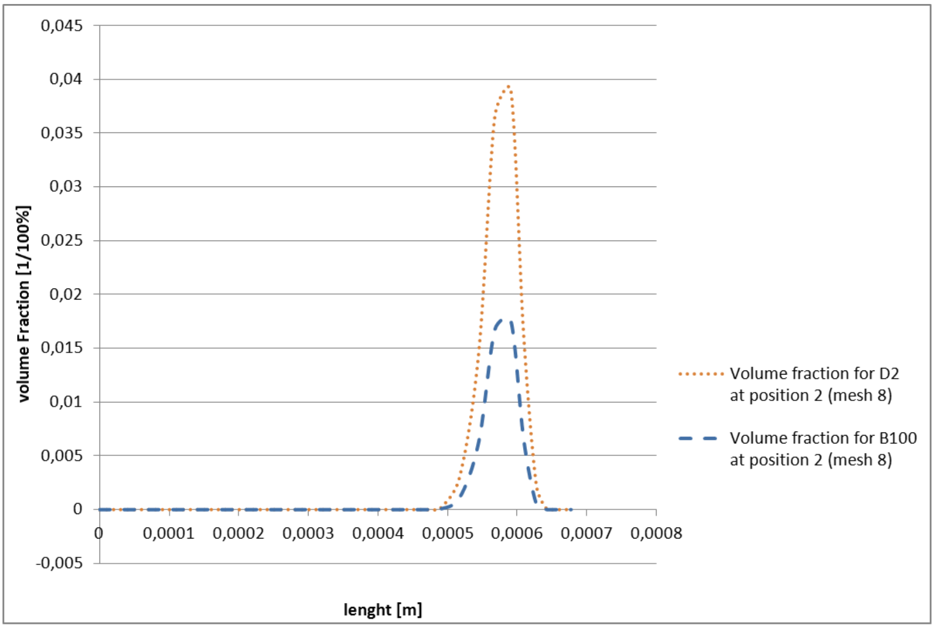


Figure 13: *Volume fraction for different type of fluid (D2 and B100) at position 2*

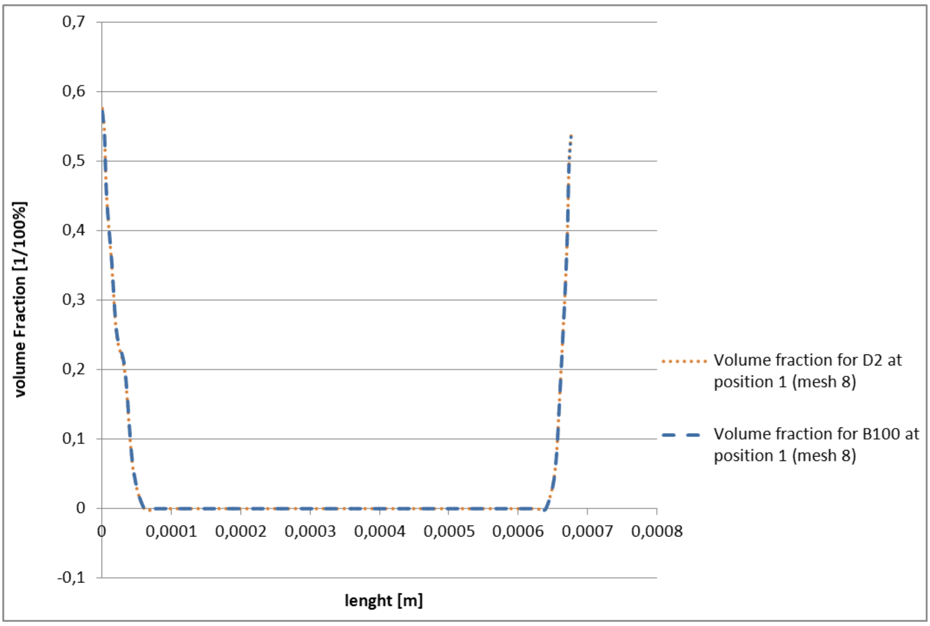


Figure 14: *Volume fraction for different type of fluid (D2 and B100) at position 1*

The results of the numerical analysis show that the block-structured type (figures 7-12) of mesh is better for numerical analysis in nozzle holes. Comparison of the results in Figure 8 an 9 show how different type significantly influences the velocity profiles. When we use structured mesh the velocity profiles are rougher then in case of block-structure. By comparison of result for volume fraction there is no difference between structured and block-structured type of mesh (figure 10-12). When we compare the results for various density meshes there is no significant difference between meshes with higher densities compared to meshes with lower density. The numerical analysis for different type of fluid (D2 and B100) are presented on figures 13-15.

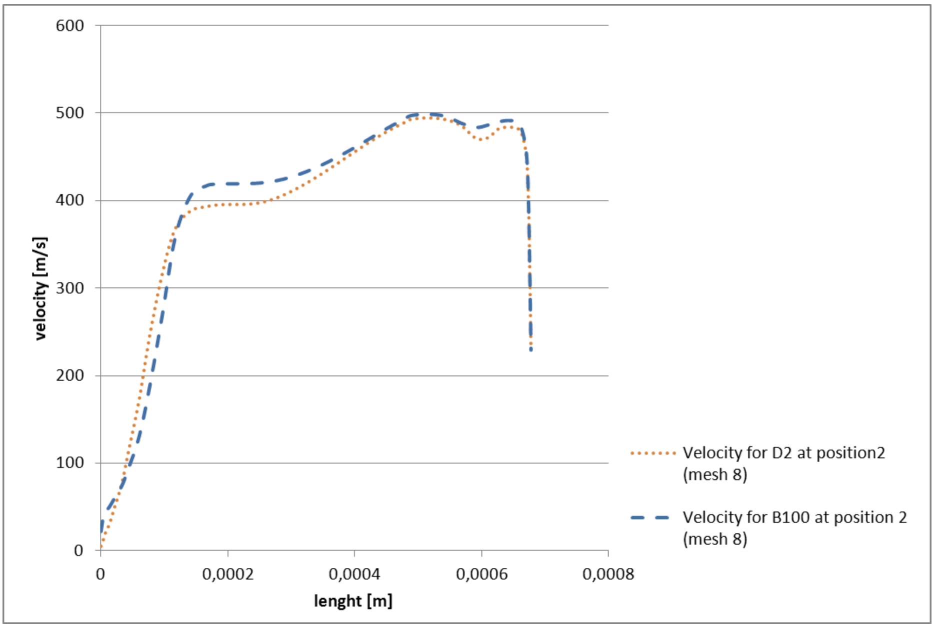


Figure 15: *Velocity profile for different type of fluid (D2 and B100) at position 2*

As already stated the results of influence of different type of fluid (D2 and B100) in nozzle showed no significant difference. Velocity field, velocity profiles and volume fraction distributions on the outlet are comparable. The results are comparable because the density of used fluids is similar and not so much different (difference is about 50 kg/m^3). The difference between dynamic viscosity is also very small and has no effect on the results. Values for mass-flow, viscosity were different (taken from experiment), pressure and temperature were the same for booth fuels.

The differences presented in figures 13-15 are very small. Some volume fraction (indicator for cavitation) analysis were made for different fuels and the difference is even smaller, so it cannot be presented properly in form similar as figures 3-6.

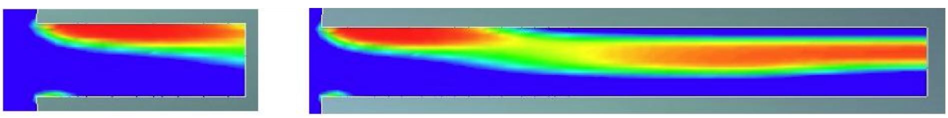


Figure 16: *Volume fraction in nozzle hole*

The vapour fraction for different types of fluid is presented on figure 16. When comparing the results the following could be observed. The structure of the cavitation is not dependent on the length of the nozzle hole. The shape of cavitation in the first part of the ole is almost identical. Later on the vapour cloud is spreading through the hole till in reaches the outlet.

The pressure distributions presented in figure 17 show the very low pressure area in the re-circulation zone. This means, that the velocity at the outlet is higher and it is also more uniform. Higher output velocities result also in better atomisation of injected fuel.

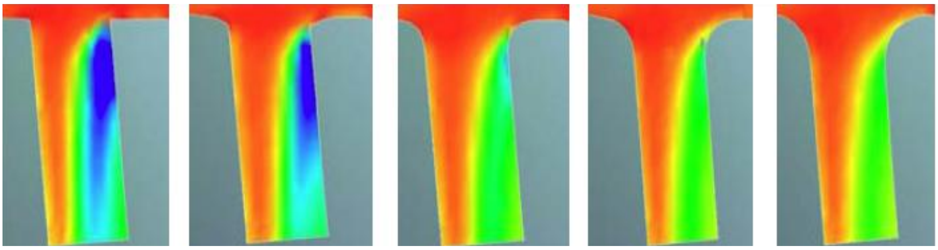


Figure 17: *Pressure distribution in the nozzle hole*

From this first analyses no concrete conclusion could be made. We could only speculate the shorter nozzle hole could have better influence on the spray formation. Further we could also suppose that region where the vapour volume fraction is lowered significantly some damages due to the bubble collapse could be expected.

CONCLUSIONS

Considering above-mentioned results the following conclusions could be made:

- The numerical analysis shows that higher pressure differences yield more cavitation.
- The results of numerical analysis are comparable for different types of fluid (D2 and B100).
- The numerical result are better when we use block-structured meshes .
- Comparison between structured and block-structured meshes show, that the velocity profiles are much rougher with structured mesh than in case of block-structured meshes.
- To cover all areas of cavitation that could appear in the nozzle hole meshes with higher density should be used .
- By introducing the chamfer at the inlet edges of nozzle hole the cavitation could be lowered significantly.
- First analyses show no significant influence of the nozzle hole length on the shape of the cavitation.

REFERENCES

[1] T. Yoda, T. Tsuda, Inflence of injection Nozzle improvement on DI Diesel Engine, SAE paper 970356

[2] Y.Oishi et.al., A computational Study into the Effect of the Injection Nozzle Inclintion Angle on the Flow Characteristics in Nozzle Holes, SAE paper 920580

[3] C.Arcoumanis et. al., Analysis of the Flow in the Nozzle of a Vertical Multi Hole Diesel Engine Injector, SAE paper 980811

[4] M.Kato et al., Flow Analysis in Nozzle Hole in Consideration of Cavitation, SAE 970052

[5] Y.Oishi et.al., A Computational Study into the Effects of the Injection Nozzle Inclination Angle on the Flow Characteristics in the Nozzle Holes, SAE paper 920580

[6] K.Melcher, J.Chomiak, Experimentelle Untersuchung der Stroemung durch Dieseleinspritzduesen in stationaer betriebenen Grossmodel, Bosch Techn.Berichte 5(1976) 4

[7] ISO-4113:1988, Road Vehicles- Calibration Fluid Diesel Injection Equipment

[8] H.Hardenberg, Die Nadelhubabhaengigkeit der Durchflussbeiwerte von Lochduessen fuer Direkteinspritzdieselmotoren, MTZ 46(1985) 4, p. 143-146

[9] FIRE Version 9 – Users manual, Multiphase flow, AVL, 2009

[10] L.C.Ganippa et.al., The structure of cavitation and its effect on the spray pattern in a single hole diesel nozzle, SAE paper 2001-01-2008

[11] L.C.Ganippa et.al., Comparison of cavitation phenomena in transparent scaled-up single hole diesel nozzles, 4th international symposium on cavitation, California institute of technology, Pasadena, 2001

[12] B.Göschel, Einspritzdüsenverschleiß, Forschungverinigung Verbrenungskraftmachinen e.V., Heft R245, 1974

[13] Kavitation. Abschlüßberichtüber die Ergebnise des Schwerpunktprogramm 1966-1972, Deutsche Forschungsgemeinschafts, Boppard, 1974

[14] G.König et.al., Analysis of flow and cavitation Phenomena in diesel injection nozzles and its effects on spray and mixture formation, 5. Internationales Symposium für Verbrennungsdiagnostik der AVL Deutschland, 6.-7.Juni 2002, Baden Baden, 2002

[15] C.Arcoumanis et. al., Visualisation of cavitation in diesel engine injectors, Mec.Ind. (2001) 2, 375-381

[16] C.Arcoumanis et.al., Investigation of cavitation in a vertical multi-hole injector, SAE paper 1999-01-0524

[17] C.Arcoumanis et.al, Cavitation in real-size multi-hole diesel injector nozzles, SAE paper 2000-01-1249

[18] C.Badock et.al, Investigation of cavitation in real size diesel injection nozzle, International Journal of Heat and Fluid Flow 20 (1999)

[19] K.Jung et.al., The breakup characteristics of liquid sheets formed by like-doublet injectors, Journal of Propulsion and Power (sent), http://rpl.snu.ac.kr/

[20] D.Radonjić, Application possibilities of similarity theory and dimensional analysis in designing process of internal combustion engines intake manifold, Mobility and Vehicle Mechanics, Vol. 31, No. 3-4, p.11-31, 2005.

INCREASE OF THE ENERGY EFFICIENCY OF PASSENGER CARS USING DIFFERENT TYPES OF TRANSMISSIONS [1]

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UDC: 621.43.018.3-63:620.92

Abstract

One of the main parameters to increase the energy efficiency is to reduce fuel consumption. For this purpose a number of vehicles fuel economy standards (FE) are introduced. They are being implemented all over the world in order to conserve energy and for reduction in carbon dioxide emissions.

In this paper, it has been discussed how different types of transmission technology could contribute on fuel economy and energy efficiency of passenger cars. Different types of transmission (automatic transmission, manual gear transmission or continuously variable transmission - CVT) differently influence fuel consumption. For examples, the CVT offers high fuel economy, presumably because it ensures a low brake specific fuel consumption (BSFC) driving condition with its continuously variable ratio characteristics. Also, it is shown that automatic transmissions are almost always less energy efficient than manual transmissions due mainly to viscous and pumping losses. The practical use of the increase of the energy efficiency of passenger cars using different types of transmissions is based on the comparison reviews investigating fuel consumption and acceleration characteristics of passenger cars with different type transmission concepts which show the significant advantages offered by new transmission concepts currently being launched as volume production models.

Key words: transmission, fuel consumption, energy efficiency.

POVEĆANJE ENERGETSKE EFIKASNOSTI PUTNIČKIH VOZILA PRIMENOM RAZLIČITIH TIPOVA TRANSMISIJA

UDC: 621.43.018.3-63:620.92

Rezime: Jedan od glavnih parametara za povećanje energetske efikasnosti je smanjenje potrošnje goriva. U tu svrhu uveden je veliki broj standarda koji definišu potrošnju goriva (FE). Oni se sprovode širom sveta u cilju uštede energije i smanjenje emisije ugljen-dioksida.

U ovom radu je analizirano kako različite vrste prenosnika snage mogu doprineti ekonomičnosti potrošnje goriva i energetskoj efikasnosti putničkih automobila. Različite

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U ovom radu je analizirano kako različite vrste prenosnika snage mogu doprineti ekonomičnosti potrošnje goriva i energetskoj efikasnosti putničkih automobila. Različite vrste transmisija (kao na pr. automatski menjač, mehanički menjač ili kontinualno varijabilna transmisija - CVT) različito utiču na potrošnju goriva. Na primer, CVT ostvaruje visoku efikasnost pri potrošnji goriva, pre svega jer obezbeđuje nisku specifičnu potrošnju goriva (BSFC) u voznom stanju pri kontinualno promenljivim prenosnim odnosima. Takođe, pokazalo se da automatski prenosnici snage su skoro uvek manje energetski efikasni u odnosu na mehaničke prenosnike snage, uglavnom zbog gubitaka na trenje. Praktična primena povećanja energetske efikasnosti putničkih automobila ostvaruje se korišćenjem različitih vrsta transmisija i ona se zasniva na poređenju istraživanja potrošnje goriva i karakteristike ubrzanja putničkih vozila. Danas se primenjuju različiti tipovi prenosa snage koji pokazuju značajne prednosti koje nude novi koncepti trenutno lansirani na tržištu.

Ključne reči: transmisija, potrošnja goriva, energetska efikasnost

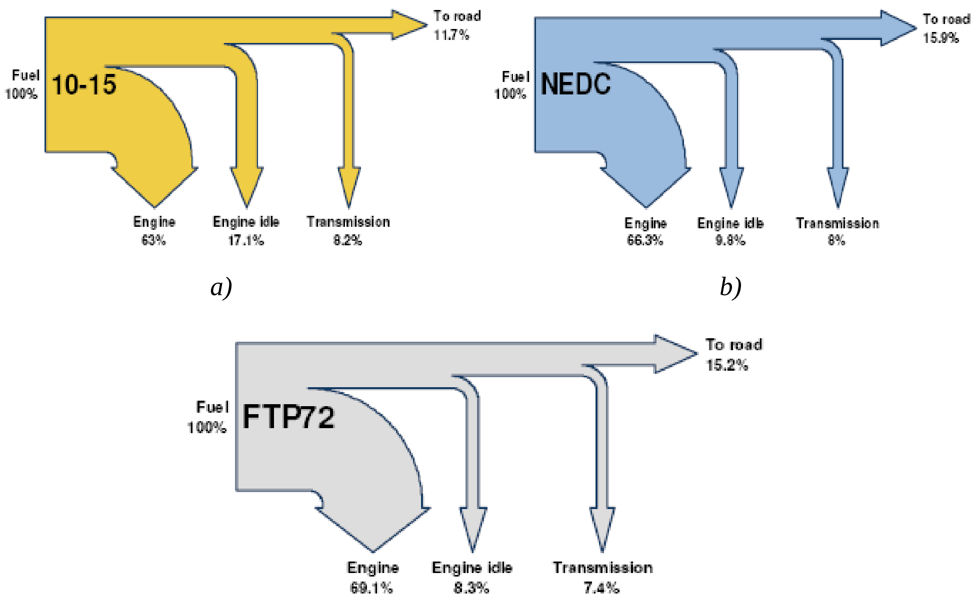
INCREASE OF THE ENERGY EFFICIENCY OF PASSENGER CARS USING DIFFERENT TYPES OF TRANSMISSIONS

Vanja Šušteršič[1]*, Dušan Gordić, Milan Despotović*

UDC: 621.43.018.3-63:620.92

INTRODUCTION

Vehicle fuel economy (FE) norms are being implemented world over to conserve energy and for reduction in carbon dioxide emissions. The European Union standards are based on fleet averaged carbon dioxide emissions, while the Japan standards are based on vehicle weight. The EU has already set the standards applicable for the model year 2012 and Japan for the year 2015. The US and Japan standards are mandatory. The EU standards are voluntary in nature so far but become mandatory from the year 2012. [1, 2]



a) *b)* *c)*

Figure 1: *Calculation results of energy losses for*
a) the Japanese 10-15, b) the European NEDC cycle and c) the US FTP72 cycle [3]

On the Figure 1 is shown energy lost in the transmission and other parts of the driveline. The new technologies, such as automated, manual transmission and continuously variable transmission, are being developed to reduce these losses. Also, in recent years, power

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transmission systems are using more number gears going to 6-, 7- and 8- gear transmissions and continuously variable transmission (CVT) to reduce fuel consumption and emission of CO_2. The new transmission designs lead to a fuel consumption reduction in the New European Driving Cycle (NECD) of 6 to 8%. In parallel, there is also an increase in acceleration from 0 to 100 km/h of 4 to 10%.

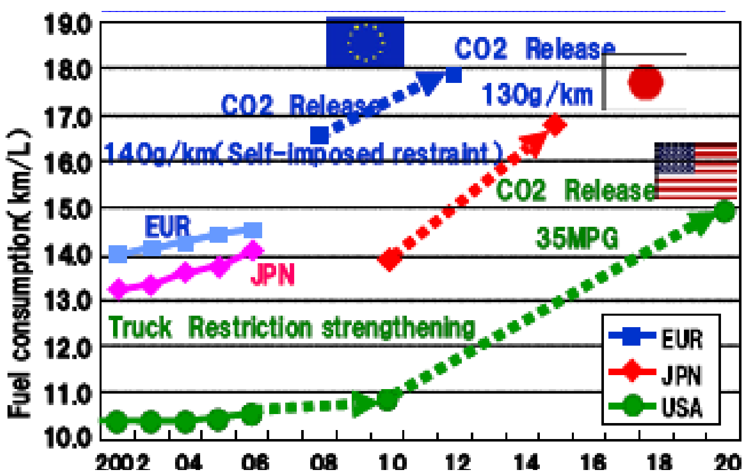

Figure 2: *Fuel economy target (EU, JPN, US) [4]*

On the Figure 2 is shown fuel economy target and CO_2 reduction for EU, Japan and US.

EFFECT OF TRANSMISSION TECHNOLOGY ON FUEL ECONOMY

Knowing that, engine performance map shows relationship between fuel consumption, engine torque and speed, it is demonstrated that the best engine fuel efficiency is obtained when it operates in medium to low speed range and at high loads. Engine specific fuel consumption (mass of fuel consumed per kilowatt-hour) increases as the operating point moves away from the best efficiency point. For example, if a vehicle has been designed for a 5-gear transmission and it is operated in 4^{th} gear at constant speed of 100 km/h the specific fuel consumption of the engine may typically increase by nearly 15% as the engine would be operating at higher engine speed and lower torque [1]. The highest possible gear is therefore, selected to keep the engine speed low and torque high. If more gears are available than it is more likely that the engine would operate close to the best efficiency point at all the vehicle speeds. A larger number of gears say, 5 to 6 in comparison to 4 forward gear ratios also results in better fuel economy and CO_2 reduction.

A reduction in fuel consumption is approximately half as expensive when achieved with investment in the drive train compared to sophisticated engine concepts. The transmission affects fuel consumption in two ways. One factor is its own transmission losses; the other is providing suitable ratios for fuel – efficient utilisation of engine power. Geared transmissions are still the most efficient, although there is now a significant factor of continuously variable transmission. But the main factor affecting consumption is still the driver. This main factor affecting reducing fuel consumption includes the follows:

- Improving the efficiency of the internal combustion engine, particularly by reducing part-load consumption.
- Appropriate engine performance characteristics, i.e. the vehicle must be neither over-powered nor under powered.
- Reducing driving resistance, for example rolling resistance and drag.
- Reducing the power draw of accessories such as servo pumps, air conditioning, etc.
- Improving the efficiency of the transmission. This relates principally to continuously variable transmission.
- Traffic management system to reduce stationary periods.
- Improved driving. Intelligent control system, which protect the driver against his own misjudgement. There are many factors involved in determining how far “usurping” of control can go [5].

A drive train components supplier has to deliver these fuel economy improvements while enhancing comfort. Coming from traditional components such as the self-adjusting clutch, dual mass flywheel and clutch release system, these components have been optimised over the years and they continue to evolve and are becoming more efficient, reliable and comfortable.

Effect of number of gear ratios and other changes in power transmission on vehicle fuel economy are given in Table 1. Also, in Table 1 is given how different types of transmission could influence on the reduction of CO_2. In further text, it would be explained each of the transmission technology improvements mentioned in Table 1.

Automatic gear transmission is another technology that influences fuel economy. As automatic transmission has been developed, more forward speeds have been added to improve fuel efficiency, performance and improve a vehicle’s market position. Increasing the number of available ratio provides the opportunity to operate an engine at more optimized condition over a wider variety of vehicle speeds and load condition.

Automated shift manual transmission (AMT) operates similarly to a manual transmission except that it does not require clutch actuation or shifting by the driver. Automatic shifting is controlled electronically (shift-by-wire) and performed by a hydraulic system or electric motor. In addition, technologies can be employed to make the shifting process smoother than conventional manual transmissions. This system can deliver a 15% improvement in fuel economy compared to conventional automatic transmission [6]. Figure 3 shows the possible savings resulting from automating manual transmission such as the parallel shift gearbox or by means of mild hybridisation.

Continuously variable transmission (CVT) is unique in that it does not use gears to provide ratio for operation [7]. Instead, the most common CVT design use two V-shaped pulleys connected by a metal belt. Each pulley is split in half and a hydraulic actuator moves the pulley halves together or apart. This causes the belt to ride on either a larger or smaller diameter section of the pulley which changes the effective ratio of the input to the output shafts. Ideally, a continuously variable transmission (CVT) provides best means to implement the strategy of engine operation near best efficiency point at all the vehicle speeds (Figure 4).

Elimination of hydraulic torque converter improves fuel economy as the fluid slippage increases energy losses. With hydraulic torque converter the engine idling speed is to be kept at higher levels compared to manual transmission increasing vehicle fuel consumption.

Table 1: *Effect of Transmission Technology on Fuel Economy and CO_2 Reduction of Passenger Cars [8]*

Transmission Technology Improvement/Change	**Fuel Economy* [%]**	**Reduction CO_2 [%]**
Use of 5-gear automatic instead of 4 -gear automatic transmission (aggressive shift logic)	1 - 2	2.5
Use of 6-gear automatic instead of 4 -gear automatic transmission	3 - 5	4.5 – 6.5
AMT (automated shift manual transmission) instead of 4 -gear automatic transmission	7 - 9	9.5 - 14.5
CVT in small FWD instead of 4 -gear automatic transmission	3 - 8	6
Elimination of torque converter	2 - 3	0.5

*NAS report 2002, NESCCAF report 2004

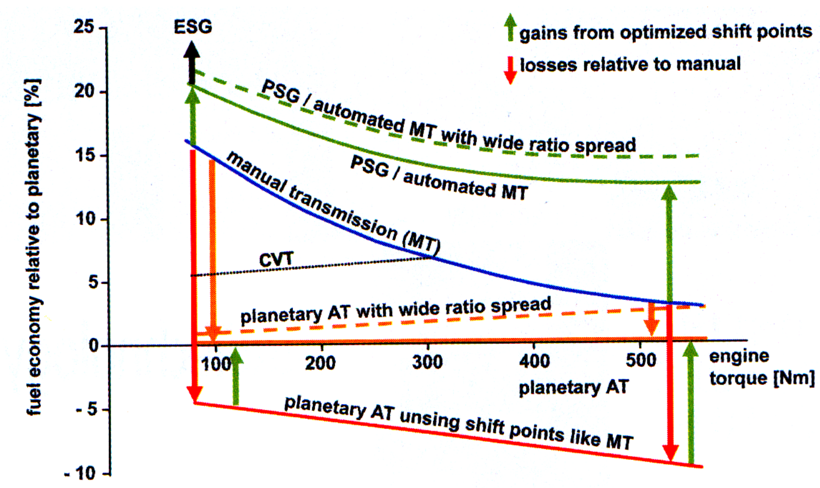

Figure 3: *Differences in fuel consumption for several transmission concepts [5]*

- PSG – parallel shift gearbox,
- ESG – electronic shift gearbox,
- AT – automatic transmission,
- AMT – automated manual ttransmission

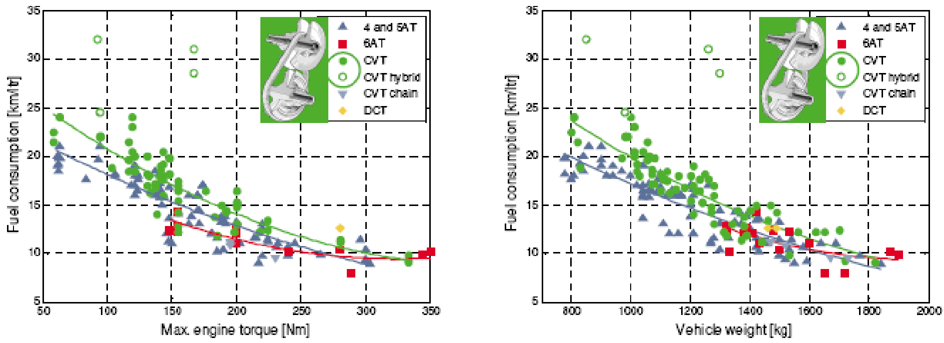

Figure 4: *Fuel consumption of FWD vehicles with several transmission types as currently available on the Japanese market (OEM data of European applications on the Japanese market) [3]*

ENERGY EFFICIENCY OF DIFFERENT TYPE OF TRANSMISSION

The increase of energy efficiency is resulted by the reduction of fuel consumption which is different in order to transmission (automatic or manual). The energy efficiency of automatic transmission has increased with the introduction of the torque converter lock-up clutch, which practically eliminates fluid losses when engaged. Modern automatic transmissions also minimize energy usage and complexity, by minimizing the amount of shifting logic that is done hydraulically. Typically, control of the transmission has been transferred to computerized control systems which do not use fluid pressure for shift logic or actuation of clutching mechanisms.

Hydraulic automatic transmissions are almost always less energy efficient than manual transmissions due mainly to viscous and pumping losses; both in the torque converter and the hydraulic actuators. A relatively small amount of energy is required to pressurize the hydraulic control system, which uses fluid pressure to determine the correct shifting patterns and operate the various automatic clutch mechanisms.

Manual transmissions use a mechanical clutch to transmit torque, rather than a torque converter, therefore avoiding the primary source of loss in an automatic transmission. Manual transmissions also avoid the power requirement of the hydraulic control system, by relying on the human muscle power of the vehicle operator to disengage the clutch and actuate the gear levers, and the mental power of the operator to make appropriate gear ratio selections. Therefore, the manual transmission requires very little engine power to function, with the main power consumption due to drag from the gear train being immersed in the lubricating oil of the gearbox.

The on road acceleration of an automatic transmission can occasionally exceed that of an otherwise identical vehicle equipped with a manual transmission in turbocharged diesel applications. Turbo-boost is normally lost between gear changes in a manual whereas in an automatic the accelerator pedal can remain fully depressed. This however is still largely dependent upon the number and optimal spacing of gear ratios for each unit, and whether or not the elimination of spool down/accelerator lift off represent a significant enough gain to counter the slightly higher power consumption of the automatic transmission itself.

CONCLUSIONS

This paper presents the effect of different transmissions on fuel consumption which affect energy efficiency and emission of CO_2. A reduction in fuel consumption is approximately half as expensive when achieved with investment in the drive train compared to sophisticated engine concepts, however the effect of different transmission is about 8%. Some of the transmission technology improvements affect the fuel economy by small but important percentage such as use of 5-gear or 6-gear automatic instead of 4 -gear automatic transmission; use of CVT in small FWD instead of 4 -gear automatic; elimination of torque converter; use of AMT (automated shift manual transmission) instead of 4 -gear automatic etc. To conclude, it is shown that hydraulic automatic transmissions are almost always less energy efficient than manual transmissions due mainly to viscous and pumping losses; both in the torque converter and the hydraulic actuators.

After price, reliability and fuel efficiency are primary decision factors for car buyers. When buying a new car, drivers want to have confidence that their cars are reliable and long lasting, with excellent fuel efficiency and low CO_2 emissions. The comparison reviews investigating fuel consumption and acceleration characteristics of passenger cars with different type transmission concepts show the significant advantages offered by new transmission concepts currently being launched as volume production models. Therefore, these new transmission make a considerable contribution towards reducing fuel consumption and assist the automotive industry in significantly reducing fleet consumption and exhaust emissions. At the same time, they offer the vehicle driver greater performance and increased driving comfort.

REFERENCES

[1] B. P. Pundir: „Fuel Economy of Indian Passenger Vehicles - Status of Technology and Potential FE Improvements“, Greenpeace India Society, 2008

[2] F. An, A. Sauer: „Comparison of Passenger Vehicle Fuel Economy and Greenhouse Gas Emission Standards Around the World“, Pew Center on Global Climate Change, 2004

[3] F. Van der Sluis at all: „Fuel Consumption Potential of the Pushbelt CVT“, FISITA, 2006

[4] T. Shibayama, H. Yada, Y. Morita, M. Fujikawa: “Introduction of the latest hydraulic control system for automatic transmission”, Proceedings of the 7th JFPS International Symposium on Fluid Power, Toyama, 2008, pp.137-142

[5] G. Lechner, H. Naunheimer: “Automotive transmission - Fundamentals, Selection, Design and Application”, Springer, 1999

[6] P. Gutzmer: “Global Trends in Transmission Development”, Auto Technology 2, pp. 70-71, 2004

[7] M. Terruso, G. Virzi’ Mariotti: “Prediction of Driving Performance of a Hybrid Vehicle with CVT transmission”, Mobility and Vehicle Mechanics, vol. 35, num. 3, pp. 25-42, September 2009.

[8] EPA Staff Technical Report: Cost Effectiveness Estimates of Technologies Used to Reduce Light-duty Vehicle Carbon Dioxide Emission, EPA, United State Environmental Protection Agency, 2008

[9] John B. Heywood**:** More Sustainable Transportation: The Role of Energy Efficient Vehicle Technologies, A report for OECD for the International Transport Forum, Leipzig, Germany, 2008

[10] REGULATION (EC) No 443/2009 OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 23 April 2009 setting emission performance standards for new passenger cars as part of the Community's integrated approach to reduce CO_2 emissions from light-duty vehicles

[11] Jovičić N., Šušteršič V, Gordić D., Babić M.: "Computer Aided Engineering in Modeling of Hydrodynamic Coupling", Mobility Vehicle and Motors, vol. 30, Number 2&3, pp. 41-51 September 2005

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