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RESEARCH METHODS FOR INVESTIGATION OF PREDICTORS ASSOCIATED WITH USING OF THE CHILD RESTRAINT SYSTEMS IN VEHICLE

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UDC:629.067

ABSTRACT: Traffic accidents present one of the most causes for child injuries in many countries, and almost 50% of all children up to 14 years have been killed in traffic accidents like passenger in vehicle. To prevent that, many car manufactures are developed different protection systems in cars, especially safety belts and child protection seats. Research studies have been investigated the impact of different factors to child restraint systems using level and set different risk levels associated with using or not using protection systems in vehicles. In addition, some studies also investigated how gender, education level, number of passengers, cultural and social characteristics in some areas, length of destination and other factors influenced on child restraint using level. Taking that, in this work we show some important investigation methods, results and recommendations for future investigations which will provide basis for taking measures for improvement of child restraint using systems in vehicle.

KEY WORDS: safety belt, child restraint systems, risk levels, vehicle, traffic accidents

METODE ISTRAŽIVANJA ZA ISTRAŽIVANJE PREDIKTORA POVEZANIH SA KORIŠĆENJEM BEZBEDONOSNIH SISTEMA ZA DECU U VOZILU

REZIME: Saobraćajne nezgode predstavljaju jedan najčešćih uzroka povreda deteta u mnogim zemljama, i skoro 50% od sve dece do 14 godina su ubijeni u saobraćajnim nezgodama kao putnici u vozilu. Da bi sprečili to, mnoge fabrike automobila su razvile različite sisteme zaštite u vozilima, naročito sigurnosne pojaseve i zaštitna dečija sedišta. Istraživačke studije su istraživale uticaj različitih faktora na bezbednosni sistem za decu koristeći nivo i set različitih nivoa rizika povezanih sa korišćenjem ili ne korišćenjem sistema zaštite u vozilima. Osim toga, neke studije su takođe istraživale kako pol, nivo obrazovanja, broj putnika, kulturološke i sociološke karakteristike u nekim oblastima, dužina destinacije i ostali faktori utiču na dečije sedište koristeći nivo. Uzimajući to, u ovom radu ćemo pokazati neke važne metode istraživanja, rezultate i preporuke za buduća istraživanja koja će omogućiti osnov za preduzimanje mera za poboljšanje dečijeg sedišta koristeći sisteme u vozilu.

KLJUČNE REČI: sigurnosni pojas, bezbednosni sistem za decu, nivoi rizika, vozilo, saobraćajne nezgode.

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RESEARCH METHODS FOR INVESTIGATION OF PREDICTORS ASSOCIATED WITH USING OF THECHILD RESTRAINT SYSTEMS IN VEHICLE

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INTRODUCTION

Worldwide, traffic accidents present the 15th leading cause of death for children less than 4 years, and the second leading cause among children aged 5 to 15 years [33]. Traffic accidents are also the largest cause of death for children over 1 year in the territory of the US [6].

The most effective protection system for preventing child mortality in traffic accident is using of child restraint systems (seat belt, seats, booster and other) in vehicles [8].

Many researchers have been investigated different impact factors associated with child restraint systems. For children under one year, there are UN ECE R44.04 or R44.03 standards for safety seats. Beside that, there are many different standards in other countries like SAD and other.

Law obligation for using child restraint system is not the same in all countries. There are many different roles for using of the child seats in terms of child height and age. In this system, the most important, that using of child seats must be putted in the traffic law or other traffic acts.

In the literature review there are different methods for investigation of child restraint using level in vehicles. Some countries many years before have been investigated this segment and putted their methodologies on their web presentation. Besides that, there are also some other methodologies who investigated causes and consequences associated with child restraint systems in vehicles.

EFFECTS OF THE USING CHILD RESTRAINT SYSTEMS AND RISK LEVELS OF TRAFFIC ACCIDENTS

Using of child restraint systems is associated with traffic accident risk levels. Children who are not using protection systems and their parents can be involved by high risk level in traffic accident. This risks are associated with light and serious injuries and also with death.

However, there is important to respect all standards of protection systems by child age, because sometime using inappropriate protection system can be very dangerous in case of traffic accident. Because of that, researches have different attitude about risk levels for child's age.

Porter et al. [26] suggested that parents are the most responsible for child restraint systems using in vehicles.

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In the last 15th years many research studies have been evaluated the risk levels and child injuries associated with the vehicles involved in traffic accidents [10].

There are many sides who suggested that using appropriate child restraint can minimize the child injuries in traffic accidents [4], [3], [10],[28], [29], [30].

In the US investigation are made by a lot of databases from different sources and because of that is very difficult to compare data with other countries.

Problem also can bee the premature graduation of child restraint system using according with child's age and height [24]. Also, using of appropriating protection system minimize the risk level for injuries in traffic accidents [22].

Child restraint systems in vehicles are designed to provide protection and to prevent or reduce the consequences resulting from a traffic accident [26].

Studies that have been researching the distribution use of the system of protection for children who were killed in traffic accidents, came to the conclusion that 73% of the total number killed children were not properly used protection systems in the United States, and 79% in Australia and New Zealand, and 64% in children under 1 year and 56% in children aged 1-4 years [26]. Because of that, the National Highway Traffic Safety Administration in US (NHTSA) recommends that all children under the age of 13 years must be on rear seats in vehicles [23].

Misconceptions among parents play the most important role in knowledge transfer for importance of child restraint systems [5]. It is therefore necessary to carry out continuous education of parents to ensure that children use the protection systems that is appropriate to their age.

The using of safety belts by drivers affect the using of child restraint systems in vehicles [13], [18], [2], [16], [14], [9].

Correlation between the drivers using safety belts and child protection system is stronger than the driver who is not using a seat belt and the child who does not use protection system [9].

The percentage of using the protection system have been significantly changed with the position of child in vehicle, and accordingly, the back seat is much better than the front seat in vehicle [14], [16].

The percentage of using the protections system decreases with children's age [7], an also depends on the location of research area and is higher in the recreational areas than in school zones.

Eby et al. [14] concluded that the level of use of the system of protection of children is higher in areas where all restraints used more and where drivers use seat belts, sports cars and commercial vehicles, as well as passengers in the front seat. They did not conclude that there are significant differences in the use of protection systems in the days of the week, sex of the child, and the types of locations in which data are collected.

For children over five years old, the research studies have been showed that there is no significant difference between using of the protection system according to the gender, time of day, and type of vehicle [14]. Higher levels using of the protection systems can be identified when: driver seat belt use; when the driver is female [26] when the vehicle is expensive; kindergarten compared to malls.

Based on the literature review, in Table 1. we summarized the effects of child protection systems to reducing risk level in traffic accidents.

Table 1 Review of research results of the child restraint systems using effects to minimizing the traffic accident risk consequences

Autori	Investigation results
NHTSA [24]	Using of child protection systems can minimize risk of death by 71% to the child’s under one year and also minimize risk of death by 54% for the child’s aged by 1-4. years old.
Elliott et al. [15]	There is 28% minimize risk of death in traffic accidents for children aged by 2-6 years who used all child restraint system in compare with children who used just safety belt for adults.
Durbin et al. [12]	Risk of injury in traffic accident can be reduced almost 60% for children (4-7 years) properly putted in the safety seats in compare with children who used just safety belt for adults.
Lee, Schofer [20]	Application of safety belts reduce injuries risks in traffic accidents to 45% at front seat, and 50% death injuries.
Arbogast et al. [1]	Properly used of safety seats for children aged over 4 years can reduced injuries in traffic accident by almost 70% in compare with using safety belt for adults.
Durbin et al., [12]	Children, age 4-7 years who use booster seats in compare with using safety belt for adults, have almost 50% better safety protection.
Herz, 1996.	Taking data from NHTSA Report for period from 1988. to 1994. year, investigator concluded that protection system for children (under 1 year old) reduce risk of death by 71% in car vehicles and 58% in vans. Also, the risk of death for children age 1-4 years old can be reduce by 54% in car vehicles and 59% in vans.
Rice i Anderson [27]	From NHTSA report for period 1996-2005. year, researchers concluded that using child protection systems reduce risk of death by 67% for children under 3 year and by 73% children under 1 year when compared to children who not used protection systems.
Durbin et al. [11]	Using of booster seats reduce risk of injuries by 59% for children age 4-7 years.
Durbin et al. [10] Lennon et al., 2008	For children under 13 years old, when they travel in the rear seat the injures accident risk is bigger for 40% [10] and two times bigger for children under 4 years old (Lennon, Siskind, Haworth, 2008).

SUMMARY OF METHODS FOR RESEARCH THE IMPACT FACTORS FOR USING THECHILD RESTRAINT SYSTEMSIN VEHICLES

The National Occupant Protection Use Survey (NOPUS) represents the only research on the use of child restraint systems in the vehicles, which being implemented on the entire territory of the United States. Research methodology involves a detailed selection of locations at which conducts research to obtain information in which nation the children are most protected with the child restraint systems in vehicles. All collected data are processed by the National Center for Statistics and Analysis in NHTSA. Statistical Reports are published annually.

Authors in [10] implemented their research methodology for collecting data on 66 locations in 31 district in seven states. They used locations like places of attractions with high volume of vehicles who have been transported children (shopping centers, medical facilities, parks with playgrounds for children, restaurants, fast food, etc..). In particular, taking into consideration the safety location, and when they can ease to collect data. Data were collected by two researchers, of which one was responsible for an interview with the drivers and the other is responsible for collecting the data on the use of child restraint systems in vehicles. Only data for vehicles with one child under 5 years aged were taken into account. The researches first asked the drivers if they want to participate in the study and about 5-10% of the total number of drivers did not want to participate. All researchers have completed special training by the NHTSA and they get a certificate. They are expected to well estimate which vehicles can participate in research and in conducting research to show the maximum degree of professionalism. All collected data were processed and analyzed according to defined procedures. For data analysis they used a descriptive statistics.

Vassentini, Willems [31] have investigated predictors who impacts on using child restraint system in the federal unit of Flanders in Belgium. Children aged under 12 years were a target group in this study. Data collection was taken from 30 recreation places (swimming pools, recreation centers, zoo etc..), and at the primary schools. The primary school taken 20 locations combined with kindergartens and the recreation areas taken at 10 locations. Based on the approval of the director of primary schools and managers of institutions, the researchers collected the data about safety belt using by driver, using child restraint system and they conducted a brief interview with the drivers/parents. During a interview, the researchers asked parents to provide them the information about child’s age, the weight and height. Researchers taking into account only vehicles that were parked in recreational areas or school zones. During data collection, the children who were sitting on the lap of an adult parent are classified as the children who did not use protection systems. During investigation, researchers also analyzed the type of protection system which been used by children. Drivers also need to give information about travel destination. In order to define which of all variables significantly affected on using child restraint systems in vehicles they used the logistics regression analysis. During analysis they used personal variables like weight and height of children, and also used explanatory variables like travel time, using safety belt by driver, position in vehicle, type of restraint for children, number of children in vehicles and travel destination.

Eby et al.[14] also investigated some predictors in connection with using of child restraint systems for children aged 4-15. years in Michigan. To make some budget savings during the investigation they choose 28 areas and defined 128 locations for researchers. Locations were food restaurants, movie theaters, shopping malls, parks for recreation etc. All places are the generally categorized into the schools and others. All researchers took some education course during five days. After that, they have demonstrated the practical knowledge on some location. During investigation they collected information about older children in vehicle, the restraint system that they have been used, to recognized in which category children are selected, driver gender and gender of child in the vehicle. Investigation period was 30 minutes. If in on some location researcher could not work, he need to find the new researching location. Researchers need to include as many vehicles as possible. For children they recorded data about using restraint system, gender and location in vehicle, and vehicle type. Child who have been used safety belt for adults are compared with children who used child restraint systems. For data analysis they used descriptive statistics.

Porter et al. [26] have conducted investigation about using child restraint systems in Turkey. Investigation was conducted in period of 90-120 minutes on defined locations.

During investigation they observed children in car vehicles, vans and other vehicles. All children was categorized into 3 groups: up to 1 year, 1-4 years and 4-8 years. Researcher also documenting the data about driver gender and age, number of children in vehicle, children's age, type of vehicle and child restraint system types. The key dependent variables were the using safety belt by driver, child position in vehicle, and also does child seating in parents lap. All data was documenting in day traffic on 10 different roads in Turkey. For data analysis they used correlation method.

Brixey et al. [5] have investigated the effects of the introduction of new mandatory used of the child restraint systems in vehicles in the area of Milwaukee. Research was conducted in cooperation with health center and two non-government offices. They used data from reports who are made by parents. Key target of investigation was to determine using levels before, in grace period and after mandatory the fines. Researcher documented data about child’s weight, height, gender, ethnicity, and the area in which he resides. Parents also had asked to give information about the type of child transport to the hospitality and type of restraint which had been used. Data set also included some information about child position in vehicles and does child had been involved into traffic accident in the last three months. Investigation focus was related to the children less than 8 years. All data were collected by the volunteers. Chi-square test was used for categorical variables, and the Multivariable logistic regression used to assess the using of the child restraint systems before and after implementation of traffic law. Taking that all data are simultaneously collected in the same location, we applied the method of regression analysis to generate the variables.

Williams et al. [32] are conducted the pilot program with the aim to improve the using of child restraint system in the area of Durham, North Carolina. They send some flayers to the children parents with the key messages about using child restraint systems in vehicles and penalties. Children in primary schools and health centers have attended the training program about importance of using child restraint systems in vehicles. After that, they conducted the evaluation process and concluded that using level of child restraint system is significantly bigger. Researchers in this study recommended that using level of restraint systems could be bigger if there is a lot of education programs and short-term actions. In this study they used a descriptive statistics.

Omari, Baron-Epel [25] in their study they tried to measure the rate child and adults using restraint systems in order to identify the associations between fatalistic beliefs and child restraint system use among Arab children. A random sample of 380 Arab drivers transporting children 8 years and younger in Israel were interviewed after observing 835 children traveling in 400 vehicles. Proper restraint ranged from 41% among children aged one to three to 9% among booster seat-eligible children. In a logistic regression model driver seat belt use, fatalistic beliefs, knowledge regarding the law on CRS, number of children in the car, age and gender were associated with all the children being restrained in the car. They concluded that drivers with higher levels of fatalistic beliefs had a lower odds ratio of restraining their children in the car, after adjusting for the other confounding variables (OR = 0.80, CI = 0.65, 0.97). They also mentioned that high levels of fatalism and low levels of knowledge in addition to other factors may inhibit Israeli Arab parents from restraining their children in cars. In this research they mentioned that children in communities are at risk of injury or death in motor vehicle crashes and there is a need for tailored interventions specific for this population.

Nambisan, Vasudevan [21] conducted a study with the aim to compare the using levels of restraint systems by children and adults in the two key situations in vehicles: Frist ,when driver used the safety belt: Second, when driver do not used the safety belt. They conducted the research in the period of three years (2003-2005) in 50 cities at the area of

Nevada. Research sample was 20.000 driver during every year. Data collection was done by special researchers. Data was documented for location inside and outside of building areas and the special focus in posted on the driver gender and gender of passenger in the front seats. The null hypothesis in their study was: 'The level of using the seat belts in the passenger front seat is in the same regardless of the degree of use of seat belts by drivers'. An alternative hypothesis was: 'The percentage of passengers who used the seat belt in the front seats is higher when the driver was used the seat belt in relation to the average value of the use of seat belts for all passengers (without taking into account the degree of use of seat belts by the driver). Hypothesis testing was performed using the Z test.

Greenspan, et al. [17] in the framework of a researching study have conducted a assessment of the using level of the protection systems at the national level and to determined which children prematurely using safety belt for adults or riding in the front seat, in the study period of the 30 days. The survey was conducted by the National Center for injury prevention and control. Investigation period was by 23th July, 2001. to 7th February, 2003. year. The telephone survey was used for the investigation. Parents who have at least two children were questioned to give the answers. The parents were asked to give information about the type of restraint systems for children that they had used in the period before 30 days. In this investigation data collection was for children under 13 years. Average time, for every person was 20 minutes during investigation. Taking whole sample, 48% was approved to be a part of the investigation. Investigation results concluded that many of children in the front seats do used the child restraint systems by the law. Percentage of using child restraint systems has been higher when children are older.

CONCLUSION

In this work the presented research results and effects of using the child restraint systems in vehicles gives the obvious needs for taking some preventive and educational activities by key government institutions and others. During 2013. year in the Republic of Serbia was conducted research for documenting the most important traffic safety performance indicators of using child restraint systems in vehicles. Based on the results of that study, it was found that children used the protective systems at a very low level in the vehicles.

Future investigations in the Republic of Serbia need to give detailed information about key predictors who are in correlation with the use of child restraint systems in vehicles. There is need to investigate the reasons why using level is low and why children do not use the child restraint systems. After that, there is need to take some activities in cooperation with the key subjects in Republic of Serbia to take a higher level of children safety in vehicles.

REFERENCES

[1] Arbogast, K.B., Durbin, D.R., Cornejo, R.A., Kallan, M., Winston, F.K. (2004). An evaluation of the effectiveness of forward facing child restraint systems. Accident Analysis and Prevention. 36, 585–589.

[2] Agran, Ph.F., Anderson, C.L., Winn, D.G. (1998). Factors associated with restraint use of children in fatal crashes. Pediatrics 102 (3), 39–43.

[3] Berg, M.D., Cook, L., Corneli, H.M., Vernon, D.D., Dean, J.M. (2000). Effect of seating position and restraint use on injuries to children in motor vehicle crashes. Pediatrics 105 (4), 831–835.

[4] Braver, E.R., Whitfield, R., Ferguson, S.A. (1998). Seating position and children's risk of dying in motor vehicle crashes. Injury Prev. 4, 181–187.

[5] Brixey, S., Ravindran, K., Guse, C. (2010). Legislating child restraint usage-Its effect on self-reported child restraint use rates in a central city. Journal of safety research, Vol.41, pp.47-52.

[6] CDC. (2012). Injury mortality statistics. Centres for disease control and prevention. National center for injury prevention and control. US.

[7] Decina, L.E., Lococo, K.H.. (2004). Misuse of child restraints. U.S. Department of Transportation. National Highway Traffic Safety Administation Washington, DC.

[8] Decina, E., Lococo, H., (2007). Observed LATCH use and misuse charasteristics of child restraint systems in seven states. Journal of Safety Reseach, Vol. 38, 273–281.

[9] Decina, L.E., Lococo, K.H. (2005). Child restraint system use and misuse in six states. Accident Analysis and Prevention. 37, 583–590.

[10] Durbin, D. R., Chen, I., Smith, R., Elliott, M. R., & Winston, F. K. (2005). Effects of seating position and appropriate restraint use on the risk of injury to children in motor vehicle crashes. Pediatrics, 115(3), e305−e309.

[11] Durbin, D. R., Elliott, M. R., & Winston, F. K. (2003b). Belt-positioning booster seats and reduction in risk of injury among children in vehicle crashes. JAMA, 289(21), 2835−2840.

[12] Durbin, R., Elliott, R., Winston, K. (2003). Belt positioning booster seats and reduction in risk of injury among children in vehicle crashes. The Journal of the American Medical Association, 289(21), 2835-2840.

[13] Ebel, B.E., Koepsell, Th.D., Bennett, E.E., Rivara, F.P. (2003). Too small for a seatbelt: predictors of booster seat use by child passengers. Pediatrics 111 (4), 323–327.

[14] Eby, D.W., Kostyniuk, L.P., Vivoda, J.M. (2001). Restraint use patterns for older children passengers in Michigan. Accident Analysis and Prevention. 33, 235–242.

[15] Elliott, R., Kallan, J., Durbin, R., Winston, K. (2006). Effectiveness of child safety seats vs. seats belts in reducing risk for death in children in passenger vehicle crashes. Archives of Pediatrics & Adolescent Medicine, 160 (6), 617-621.

[16] Edgerton, E.A., Duan, N., Seidel, J.S., Asch, S. (2002). Predictors of seat belt use among school-aged children in two low income Hispanic communities. Am. J. Prev. Med. 22 (2), 113–116.

[17] Greenspan, A., Dellinger, A., Chen, J. (2010). Restraint use and seating position among children less than 13 years of age: Is it still a problem? Journal of Safety Research, 41, 183-185.

[18] Glassbrenner, D. (2005). Child restraint use in 2004—overall results. National. US Department of Transportation DOT HS 809 845.

[19] Hertz, E., 1996. Research Note: Revised Estimates of Child Restraint Effectiveness. National Highway Traffic Safety Administration, Washington, DC.

[20] Lee, H., & Schofer, L. (2003). Restraint use and age and sex characteristics of persons involved in fatal motor vehicle crashes. Journal of the Transportation Research Forum, Vol. 1830 (pp. 10−17). Washington, DC: Transportation Research Record.

[21] Nambisan, S., Vasudevan, V. (2007). Is seat belt by front passengers related to seat belt usage by their drivers? Journal of Safety Research. 38. 545-555.

[22] Nance, L., Lutz, N., Arbogast, B., Cornejo, A., Kallan, J., Winston, K. (2004). Optimal restraint reduces the risk of abdominal unjury in children involved in motor vehicle crashes. Annals Surgery, 239(1), 127-131.

[23] National Highway Traffic Safety Administration-NHTSA (2010). Child Passenger Safety.

[24] National Highway Traffic Safety Administration-NHTSA. (2009b). Traffic Safety Facts 2008: Children.

[25] Omari, K., Baron-Epel, O. (2013). Low rates of child restraint system use in cars may be due to fatalistic beliefs and other factors. Transportation Research Part F. 16, 53-59.

[26] Porter, B., Lajunen, T., Ozkan, T., England Will, K. (2010). A behavioral observation study of Turkish drivers and children's safety belt use. Procedia Social and Behavioral Sciences, 5. 1607-1609.

[27] Rice, T.M., Anderson, C.L. (2009). The effectiveness of child restraint systems for children aged 3 years or younger during motor vehicle collisions: 1996 to 2005. Am. J. Public Health 99 (2), 252–257.

[28] Smith, K.M., Cummings, P. (2004). Passenger seating position and the risk of passenger death or injury in traffic crashes. Accid. Anal. Prev. 36, 257–260.

[29] Smith, K.M., Cummings, P. (2006). Passenger seating position and the risk of passenger death in traffic crashes: a matched cohort study. Injury Prevention. 12, 83–86.

[30] Starnes, M. (2005). Child passenger fatalities and injuries, based on restraint use, vehicle type, seat position, and number of vehicles in the crash (Technical Report).Washington, DC: National Highway Traffic Safety Administration.

[31] Vassentini, L., Willems, B. (2007). Premature graduation of children in child restraint systems: A observational study. Accident Analysis and Prevention. 39, 867-872.

[32] Williams, A., Wells, J., Ferguson, S. (1997) Development and evaluation of programs to increase proper child restraint use. Journal of safety research. Vol. 28, No.2, 69-73.

[33] World Health Organization. (2009). Global status report on road safety: Time for action. Geneva.

OPTIMIZATION THE PERIODICITY OF MANAGING OF PREVENTIVE MAINTENANCE OF TECHNICAL SYSTEMS

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ABSTRACT: There is given methodology of determination the periodicity of managing the processes of preventive maintenance of electronic devices, within strategy of its preventive maintenance. This methodology is possibly to apply when there one can manage the revision which doesn't change intensity failure of analyzed part after every failure.

KEY WORDS: motor vehicle, electronic devices, technical systems, preventive maintenance

OPTIMIZACIJA PERIODIČNOG UPRAVLJANJA PREVENTIVNOG ODRŽAVANJA TEHNIČKIH SISTEMA

REZIME: Data je metodologija utvrđivanja periodičnog upravljanja procesima preventivnog održavanja elektronskih uređaja, u okviru strategije svog preventivnog održavanja. Ova metodologija je moguće primenljiva kada postoji jedno upravljanje revizijom gde se ne menja intezitet neuspeha analiziranog dela posle svakog neuspeha.

KLJUČNE REČI: motorno vozilo, elektronski uređaj, tehnički sistemi, preventivni remont

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OPTIMIZATION THE PERIODICITY OF MANAGING OF PREVENTIVE MAINTENANCE OF TECHNICAL SYSTEMS

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INTRODUCTION

Timely managing of preventive maintenance operations and quality of their conduction represent effectiveness of preventive maintenance of technical systems. Frequency of failure occurrence in vehicles dictates times of preventive maintenance operations. Quality of failure identification depends on kind of discovering of defective elements, applied methods of defects prediction and periods predicted for preventive maintenance. Efficiency of preventive maintenance depends in essence on skill of electronic devicesusage.

Some of basic methods of failure prevention are:

- Quality control of devices functionality on the base of outgoing parameters. This method is based on the fact that change of intake parameters lead to interrupted functionality, which lead to changes of outgoing parameters. Here it is not straightforward possible to discover element which caused failure. For identification of defect is necessary to detect defective element and its maintenance.
- Use of statistical probability of part proper operation until first failure, obtained on the base of long term operation experience. In this case is possible, with certain probability, to predict moment of failure and to make steps to prevent it.
- Control of physically chemical changes of structure of considered parts if prediction devices are available.

Above methods are the most often applied for prevention of failures of electro mechanical devices and elements of technical systems, for which statistical rules of failure appearance are established.

ACUMULATION OF DEFECT TECHNICAL SYSTEMS

Planning of preventive maintenance on time mainly dictate its effectiveness in technical systems. Recognition of failure regularity of technical systems lead to schedule of execution of preventive maintenance. Too early maintenance causes unnecessary and irrational delays of technical systems, but too long periodT_{pr} leads to increase of failures caused with unfixed defects. Thus there is optimal periodicity of preventive operations$T_{pr\ opt}$, which leads to the best results, that is to maximal effectiveness of system.

Depending on method of defects prevention character of defect accumulation process in time may be described as follows

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1. If for some element predicting parameter is known, then probability of its operation without failure for time t may be estimated according following expression

$$P(t)=\int_{t}^{\infty}\Psi_{p1}(t)dt \quad , \tag{1}$$

where Ψ_{p1}(t)- function obtained by calculation in time.

2. For parts of same type there are failures that can be prevent, as well as failures that cannot be prevent, and their statistical laws of distribution are known, leading to probability of work without failure given as follows

$$P(t)=\int_{t}^{\infty} f(t)dt \quad , \tag{2}$$

where

$$f(t)=C_1 f_p(t)+C_2 f_n(t) \quad , \tag{3}$$

f_p(t), f_n(t) - specific probability of distribution of appearance of unavoidable failures;C_1,C_2, - coefficients that determine contributions of f_p(t) and f_n(t), that compose f(t).

Lately there are more and more effective objective methods for evaluation of technical condition of mobile systems, based on implementation of automatic diagnostic systems. Coefficients C_1 and C_2 satisfy relation $C_1+C_2 = 1$.

In the first case, when prediction parameter is known, it is usual to determine certain boundaries of part qualities, which may be controlled in working process. Quality of element gradually decreases, when it approaches to moment after which failure occurs. Thereby it may be determined preliminary degree of controlled parameter after which one detects and replace defective parts. This level is called degree of prognosis. During analyzing of prognosis parameter changes in time following quantities are employed:

- $\overline{\alpha}_0$ - mathematical expectation of initial values dissipation,
- $\overline{\alpha}_{cr}$ - critical degree of operating ability (functionality) – mathematical expectation of limiting parameter values in which failure occurs,
- $\overline{\alpha}_{pr}$ prediction degree (preventive control),
- T_{pr} - prediction period – period between two preventive control,
- $\Delta\overline{\alpha}_{cr}$, $\Delta\overline{\alpha}_{pr}$ total and preventive reserve of parts reliability, which may be expressed in the following forms

$$\Delta\overline{\alpha}_{kr}=\overline{\alpha}_0-\overline{\alpha}_{kr}\,,\ \Delta\overline{\alpha}_{pr}=\overline{\alpha}_{pr}-\overline{\alpha}_{kr}\,, \tag{4}$$

With knowledge of statistical low of change of controlled parameter with time allows that on the basis of its measuring in the moment $\bar{t}_{cr}=\bar{t}_{pr}+T_{pr}$, represents precondition for prevention of failure of technical system.

Forecasting period depends on rates of changes of parameters with time. On that basis any prediction parameter$\alpha_{(t)}$ may be evaluated according coefficient of its change in time:

$$K_\alpha = \frac{\overline{\alpha}_0 - \overline{\alpha}(t)}{t}, \tag{5}$$

That coefficient characterizes process of failure multiplication in time.

Distribution of time of occurrence of unavoidable failure, with acceptable accuracy, may be approximated according exponential lows [1]:

$$f_N(t) = \lambda \exp(-\lambda t), \tag{6}$$

Whereλ - intensity of unavoidable failure.

For failures that may be avoided it is possible to assume distribution described with truncated (partial) normal law [1]. In that case superposition of described laws, according expression (3), having in minds equations (6) and (2), will lead to expression

$$f(t) = C_1 \frac{c}{\sigma\sqrt{2\pi}} \exp\left[-\frac{(t-T_{sr})^2}{2\sigma^2}\right] + C_2 \lambda \exp(-\lambda t), \tag{7}$$

In expressions (3) and (7) coefficient C_1, which determines number of failures which may be prevent, represents coefficient of failure character $A(T_e)$.
Having in minds relation

$C_1+C_2 = 1$, it may be obtained that is $C_2 = 1 - A(T_e)$, (8)

It may be shown that substitution of (7) in (2), using (8), expression (2) may be transformed as follows

$$P(t) = A(T_e) \frac{\Phi\left(\frac{t-T_{sr}}{\sigma}\right)}{\Phi\left(\frac{T_{sr}}{\sigma}\right)} + \left[1 - A(T_e)\right] e^{-\lambda t}, \tag{9}$$

were

$\Phi\left(\frac{t-T_{sr}}{\sigma}\right), \Phi\left(\frac{T_{sr}}{\sigma}\right) -$ are tabulated function of probability integral [2].

Expression (9) characterizes statistic distribution during process of multiplication of failures without existence of prediction parameter. If, in expression (7), coefficients C_1 and C_2 are expressed with coefficient of failure character $A(T_e)$, it may be obtained expression for determination of frequency of failure appearance$\lambda_c(t)$ in case of superposition of exponential and truncated normal law:

$$\lambda_c(t)=\frac{A(T_e)\frac{c}{\sigma\sqrt{2\pi}}\exp\left[-\frac{(t-T_{sr})^2}{2\sigma^2}\right]+\left[1-A(T_e)\right]\lambda\exp(-\lambda t)}{A(T_e)\frac{c}{\sigma\sqrt{2\pi}}\int_0^t\exp\left[-\frac{(t-T_{sr})^2}{2\sigma^2}\right]dt+\left[1-A(T_e)\right]\exp(-\lambda t)}, \quad (10)$$

Pre-request for timely undertake of procedure for preventive maintenance technology of technical systems is knowledge of laws of appearance of failure in time (rate of change of prediction parameterK_α in the first case, and statistical distribution of probability of work without failure in second case).

POSSIBILITIES OF FAILURE DETECTIONS

Efficiency of preventive maintenance works on technical system depends not only on well timed recognition and quality of maintenance, which in turn depends on general timing, but on preventive maintenance according to in advance set schedule. This quality depends on all staff competency, equipment quality and time devoted to it. In practice is important to determine timing of preventive maintenance, when skill level of staff and prognostic equipments are known. Time necessary for preventive maintenance of any technical system consists of time for detection t_B, time for repair t_y, and time for subsidiary works, such as tool and accessories preparation, assemblies etc. Time necessary for detection of defect parts depends on kind of work. These works are connected to time random processes of defect parts discovery and, therefore, they have random character. Maintenance time may be calculated as

$$T_{pc}=\sum_{i=1}^{d}t_{pi}, \quad (11)$$

where t_{pi} – time for performance of i-th work, and d – number of different kinds of work.

Intervals for necessary works cannot be set in advance. Intervals necessary for discovering of defect parts usually is much greater than those for amendment, that is $t_B>>t_y$. Remain preventive works such as replacement of defect parts, re-assemblages, examinations, cleanings, lubrications, etc, have routine character. Time necessary for their fulfillment may be calculated as

$$T_{py}=\sum_{j=1}^{r}t_{pj}, \quad (12)$$

where r - number of different kinds of work.

Preventive, which include both random and determined works and all working processes may be set on the basis of amendment intervals. Same as efficiency of any process, which may be estimated on the basis of number of products in time and time necessary for production, preventive efficiency may be estimated on the basis of number of detected and amended failures., Number of detected, checked and amended elements during preventive works, in general, are not linearly dependent because of random character of failure detection.Productivity of preventive maintenance is determined with number of checked elements in time unit. Intensity of detection of failed elements means number of detected defect elements, that is prevented failures, in tome unit in comparison with their number in moment t.

Now we are going to establish connection between probability and frequency of detection of defect parts, and time interval necessary for preventive work T_p. In random detection of defects frequency n(t), in analogy with failure intensity, may be determined with rate of number of detected defect parts in unit time, in relation to undetected defects. Therefore one obtains:

$$\nu(t) = \frac{dn_{pv}}{\left[n_p\left(T_{pr}\right) - n_{pv}(t)\right]dt}, \tag{13}$$

Where n_P (T_{pr}) – number of preventively fixed defects, accumulated up to beginning of preventive operation, n_P (T_{pr}) – npr (t) – number of defects undetected up to moment t.

After some simple algebra it may be obtained

$$\nu(t) = \frac{P_{pv}^{/}}{1 - P_{pv}}, \tag{14}$$

Integration of expression (14) and easy transformations lead to

$$P_{pv} = 1 - \exp\left[-\int_0^{T_p} \nu(t)\,dt\right]., \tag{15}$$

There fore in preventive maintenance random process of detection of defected elements may be approximated with expression (15). In the case of scheduled process of failure detection probability of failure prevention, mainly, depends on prediction accuracy and on time used for it. It may be assumed that probability of defect part overlooking, because of inaccuracy of measuring equipments, mistakes made by technicians and shortage of time Q(L), independent. In that case probability of failure detection may be expressed as

$$P_{pv} = [1 - \delta]\left[1 - Q(L)\right],, \tag{16}$$

where

$$Q(L) = \begin{cases} 1 - LT_p, \, pri 0 \le T_p \le T_{p_0}; \\ 0, \, pri T_p > T_{p_0}. \end{cases},$$

T_{p_0} $-$ *time necessary for preventive maintenance,*

$L = \dfrac{1}{T_{p_0}} -$ time norm for preventive operations,D - relative error of real system.

Knowing laws of failure detection process (eqns. 15 and 16), leads to reliable determination of time for preventive work.

CONCLUSIONS

Effective preventive maintenance means timely determination of necessary preventive operations, and quality of their performances. Time of preventive operations is governed with speed of defect appearance. Quality of defect detection depends on applied prediction methods, on way of defect detection and time planned for preventive operations.

It is obvious that in both cases efficiency of preventive maintenance depends on skilled exploitation of equipments. In analysis of preventive measures are often imposed questions connected with quantitative estimations and comparisons of different methods of preventive maintenance. Mathematical model, shown here, allows quantitative estimation of preventive arrangements influence on reliability of technical systems. In analysis of preventive arrangements often it is not reliable benefit only criterion, but it should be considered cost of achieving it.

Mathematical model, shown here, may be employed for estimation of quality of preventive works, as well as for quantitative cost analysis. Usage of obtained characteristics may lead to estimation of preventive maintenance influence, depending on cost of its application, on reliability of technical systems.

REFERENCES

[1] Krstić, B., Tehnical serviceability of motor vehicles and engines, University of Mechanical Engineering, Kragujevac, 2009. (In Serbian)

[2] Bass. M.S., Kwakernak H.: Rating and Ranking of Multiple Aspect Alternatives Using, Fuyy Sets Automatics, Vol.13, No. 1, 1977, p. 47-58,

[3] B. Krstić, V. Lazić, R. Nikolić, V. Raičević, I. Krstić, V. Jovanović: Optimal strategy for preventive maintenance of the motor vehicles clutch, Journal of the balkan tribologikal association, Vol.15, No 4, (2009), 611-619

[4] Vukadinović S.:Elementi teorije verovatnoće i matematičke statistike, Privredni pregled, Beograd, 1980. , str 525.

[5] B. Krstić, V. Lazić, V. Krstić: Some views of future strategies of maintenance of motor vehicles, Tractors and power machines, Vol.15, No.1, 2010, p.42-47.

CIMPORTANCE OF PROPER DETERMINATION OF VEHICLE DECELERATION FOR TRAFFIC ACCIDENT ANALYSIS

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UDC:629.016

ABSTRACT: In the analysis of traffic accidents is one of the many tasks set before an expert traffic engineering profession is properly calculate or estimate deceleration motor vehicles that were involved in the accident. The problem becomes much simpler if done immediately after the accident inspection, and the court records are no data on the measured braking forces. However is not uncommon for a person who performs inspection data is entered on the weight of the vehicle when performing emergency technical inspection. As usual site investigation documentation contains information on vehicle weight, or information on the type of engine and associated equipment, it comes to the situation where the measured brake forces of vehicles, while data on vehicle weight cannot be determined. Considering to the weight of the same brand and type of vehicles can vary more than 500 kg, depending on the type of engine and equipment, it is also the maximum value of the braking coefficient, and therefore the maximum speed of the vehicle to trace braking can differentiate and more than 30 km/h. Depending on the used vehicle weight, will depend on the calculated speed of the braking trace, and therefore possible gaps participant accident.

KEY WORDS: site investigation, data, deceleration the vehicle, traffic accident analysis

VAŽNOST PRAVILNOG ODREĐIVANJA USPORAVANJA VOZILA ZA ANALIZU SAOBRAĆAJNE NEZGODE

REZIME: U analizi saobraćajnih nezgoda jedan od mnogih zadataka koji se postavljaju pred stručni inženjering saobraćaja je da pravilno izračunaju i procene usporavanje motornih vozila koji su bili uključeni u saobraćajnoj nezgodi. Problem je daleko jednostavniji ukoliko se uradi odmah nakon pravljenja zapisnika udesa sa podacima o kočionim silama. Međutim, nije neuobičajeno da osoba koja obavlja inspekciju unese podatke o težini vozila prilikom izvođenja hitnog tehničkog pregleda. Kao i obično, uviđajna dokumentacija sadrži podatke o težini vozila, ili informacije o tipu motora i prateće opreme, i dolazi se do situacije gde je izmerena kočiona sila vozila, dok se podaci o težini vozila ne mogu odrediti. S obzirom da težina iste marke i tipa vozila može varirati više od 500kg, u zavisnosti od vrste motora i opreme, takođe maksimalna vrednost koeficijenta kočenja, a samim tim i brzina vozila prilikom ostavljanja traga kočenja može biti i više od 30 km/h. U zavisnosti od korišćene težine vozila, zavisiće i izračunata brzina prilikom traga kočenja, pa takođe i mogući nedostaci učesnika nesreće.

KLJUČNE REČI: mesto uviđaja, podaci, usporavanje vozila, analiza saobraćajnih nezgoda

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INTRODUCTION

Traffic accident experts are often faced with the problem related to precise definition of vehicle deceleration, which was involved in a car accident. Defining of deceleration, that the vehicle had at the time of traffic accident occurrence, enables accurate calculation of important parameters of the expertise of traffic accident, such as the speed of the vehicle at the time of driver's reaction, the speed at the beginning of the skid marks, the speed at the moment of impact, characteristic positions of accident participants in certain phases of the collision and determining the possibility of accident avoidance. On the other hand, there are situations where precise speed identification is crucial for accurate determination of the collision spot. The above parameters of traffic accident analysis are important for definition of omissions of accident participants, so it can be concluded that determination of the deceleration represents one of the most important elements for traffic accident expertise.

Reliability of the analysis of traffic accidents is proportional to the quality of the investigation documents and conducted investigations. In some cases, certain results are a comparative analysis injury, damage, marking, or statements of participants of a traffic accident. In some cases, the precise conclusions require a comparative analysis of the two elements (for example, damage or injury), but not rare situations to be only one of the above analysis (for example, on the basis of damage)can come to a conclusion or expect or confirm the occurrence of an accident in a certain way. The most common cases of this type are related to the determination of authenticity occurrence of traffic accidents.

In some cases of traffic accidents, based on the available documentation, it is possible to calculate the deceleration of the vehicle, and in some cases, so we can say that there are two possible directions of traffic accident analysis, depending on the available data. Therefore, an expert in the analysis of traffic accident, he must find the elements of the case file, which will enable him or determination, or more accurately calculate the deceleration of the vehicle.

For the analysis of traffic accidents, expert of traffic technical professions are obliged to apply the legal requirements regarding the safety of the braking system. Requirements braking system of vehicles can differ depending on the period of validity of the regulations. Thus, the Regulation of dimensions, total masses and axle loads of vehicles and the basic conditions that must be fulfil by devices and equipment on vehicles in traffic (hereinafter referred to as 'the old Regulations'), which was abolished by the 22.09.2010, the minimum value prescribed braking coefficient for passenger cars is 0.55. Applying the

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applicable Regulation of the division of motor vehicles and trailers, and technical requirements for vehicles in traffic (hereinafter 'the new Ordinance'), the minimum value prescribed braking coefficient for passenger cars is 0.50.

Another significant difference between the old and the new Regulations is that the difference braking force on the wheels of the same axle does not exceed 30%, under the new Regulations, or 20% under the old Regulation. The third significant difference in the brake system of passenger cars between old and new Regulations is related to the ratio of braking force the each axle and the braking coefficient. Under the old Regulations the value of brake force at each axle is at least 30% of the braking coefficient, and under the new Regulations such restrictions does not exist. To calculate the braking coefficient or percentile differences as a basis always use higher braking force. Differences in values of minimum correct braking systems old and new Regulations also exist for other categories of vehicles.

For correct brake system of a vehicle, the expert must verify the correctness of the operating brake and check the correctness of auxiliary brake. Condition to meet the auxiliary brake is that the braking coefficient auxiliary brake must not be less than 20%.

DETERMINATION OF DECELERATION

Deceleration of the vehicle in the particular case at this particular place can achieve depends on two main factors, namely the deceleration provided by roadway (bk) and deceleration can achieve vehicle (ba). Deceleration of the vehicle (b) in a particular case is determined by the 'law of the minimum' b = min (bk, ba) or deceleration of the vehicle (b) is the minimum of deceleration that can provide vehicle and roadway.

Deceleration braking system of the vehicle can achieve (ba) can be determined by calculating the braking coefficient if available, of a diagram of brake force, or assessment, taking into account all the important elements. Regardless, did the deceleration of the braking system of vehicle could achieve calculated or estimated, an expert to determine the deceleration of the vehicle is able to achieve (b), must take into account several factors that influence the definition and assessment of deceleration, which are: road condition (new, worn, smooth, rough, dark, bright, ...), road surface material (asphalt, concrete, cube, earth, dirt road, ...), the road surface (dry, moist, wet snow, frozen, ...), weather conditions (at the time of the accident was raining, just started raining, snowing, windy, ...), or as exist marks of braking and possibly some indirect indication of the correctness of the brake system of the vehicle.

The first and certainly the easiest way of determining the deceleration to the vehicle after a traffic accident sent for emergency inspection, where the measured braking force, and then based on measured brake force calculation of the theoretical (maximum) value of deceleration that the vehicle could achieve. Then it checks to see if the vehicle is in the specific conditions at a given site could achieve the calculated deceleration and check the condition of the carriageway, pavement materials, road surface, weather conditions, the existence of exist marks and the like. And determines the deceleration of the vehicle is able to achieve. If the deceleration of the vehicle is able to achieve less than the deceleration provided by road, or if you are not leaving traces of braking, then the expert to analyze accidents using calculated deceleration. Please note that this is a theoretical value that is rarely achieved in real terms. In real driving conditions, the value of the deceleration is generally lower. Namely, the technical review, when measuring the brake force on the device with rollers, and due to the characteristics of these devices, there is no lock braking, in real driving conditions, if any exist marks, the wheels were blocked. Also, the coefficient of adhesion of road is often less than the coefficient of grip the rollers on a technical review.

If braking forces are not measured, i.e. if the expert does not have the possibility of calculating the braking coefficient and deceleration, an expert will assess vehicle deceleration is able to achieve, also bearing in mind that the slowing vehicles (b) define the basis deceleration realise the vehicle and the deceleration provided by roadway.

The expert will assess the benefits braking coefficient corresponding to the minimum of the correct braking system for a given category of vehicle, or to evaluate the braking coefficient depending on the brand and type of car, the age and condition of the braking system, etc. Regardless of the method estimates the deceleration of the braking system of the vehicle could have been achieved, an expert to determine the deceleration of the vehicle is able to achieve (b), must take into account these factors influence the definition and assessment of deceleration, as follows: the road condition, road materials, road surface, weather conditions, the existence of skid marks and the like.

If the expert estimate that at the time of a traffic accident braking system was at least technically correct, it will be an expert to use the value of the coefficient of brake service brakes which corresponds to a minimum of proper brake system for a given vehicle category (see table 1). If the expert assessment of the vehicle braking coefficient varies from a minimum of proper braking system for a given category of vehicle, adopt a lower or higher value of the braking coefficient values corresponding to the minimum of the correct braking system for a given category of vehicle. After considering the circumstances of overall expert will assess the deceleration taking into account all the circumstances are fulfilled or evaluate slowing based on several factors, but will be in the form of so-called again. law of the minimum. This method of deceleration in some cases it may be accurate enough, whereas the expert must take into account whether the slight differences in the assessed value of the deceleration affects the output of the analysis of traffic accidents, and the definition of failure of participants of a traffic accident.

Table 1The minimum prescribed braking coefficients operating brake

	Operating brakes	*Auxiliary brakes*
Motorcycle	*40*	*20*
Passenger vehicle	*50*	*20*
Buses	*50*	*20*
Trucks	*45*	*20*
Trailers	*40*	-
Other vehicles (tractors, machinery, ...)	*25*	-

We emphasize that in practice it can happen that the roadway does not occur skidmarks brake, where the vehicle is forced braked. This situation occurs when the deceleration roadway provides greater brake than that achieved deceleration braking system of the vehicle and the vehicle brake 'only' a deceleration of the vehicle can achieve and will not be skidmarks brake on the roadway. If the vehicle deceleration greater than the deceleration provided by roadway, then the vehicle in that place brake provided the roadway and a rule will 'stay' skid marks brake on the roadway.

It is important to emphasize the frequent situation encountered in real driving conditions, and the forced braking, when a decline in performance achieved braking or retarding a decline in the vehicle, which must be taken into account under the right conditions. Namely, if the length of the skid marks brake 20 m to 30 m and if the vehicle speed at the beginning of skid marks greater than 60 km/h, a decline decelerating from 10%. If the skid marks brake longer than 30 m and if the vehicle speed at the beginning of the skid marks braking in excess of 60 km/h there is a decline deceleration of 15%. Example: If the vehicle deceleration is 5.4 m/s^2, and there by the speed of the vehicle at the beginning of the skid marks brake 80 km/h and the length of skid marks brake is 25 m, the further calculation leads to a deceleration of $5.4 \cdot 0.9 = 4.86$ m/s^2.

EXAMPLE OF DETERMINING ACCIDENT

This paper presents three examples of analysis of traffic accidents, with special emphasis on the definition of deceleration and impact deceleration to determine the failure of the participants of an accident.

The first example shows how the lack of data on the weight of the vehicle (which is a person who performs inspection was required to state, as is obtained by reading from the vehicle registration card or by measuring the vehicle), the exercise of extraordinary technical inspection and measurement of brake force leaves open the question of the failure participants in traffic accidents. Namely, with respect to the mass of the same brand and type of the vehicle may vary, and more than 500 kg, depending on the types of vehicles and engines, and that the maximum value of the coefficient of the brake, and therefore the maximum speed of the vehicle to brake the traces may vary, and more than 30 km/h. Depending on the use of the vehicle weight, will depend on the value of the calculated velocity to skid marks braking, and therefore the possible gaps participant accident, because failures of participants depends on the calculated speed at which the vehicle was on the scene of an accident.

The second example shows how the speed of the vehicle at the time of impact varies depending on whether at the time of the accident was located in the trunk of the vehicle load. Available documentation contained a diagram of brake force, but not the fact that you are in the trunk of the vehicle was located burden and whether the vehicle is on a technical review measured the potential burden that was in the trunk, and given the state of drivers that are located in the trunk load mass 300 kg. Depending on the vehicle's weight, varies the speed of vehicles at the time of the collision. On the other hand, given that the used vehicle weight of the vehicle registration to be, depending on whether the measurement of brake force in vehicle located burden or not, differentiate and deceleration of the vehicle. Using different deceleration and speed at the time of the collision, leads to different vehicle speed at the beginning of the skid marks brake.

The third example shows an analysis of traffic accident which was not known whether the cargo that is transported on the roof of the vehicle affected by the weight of the vehicle at the time of occurrence is greater than the maximum allowable weight of the vehicle. Specifically, the file was a fact that the braking system of the vehicle is technically correct, in which there were no data on measured forces on a technical review, nor in evidence existed diagram of brake force. The analysis is further complicated if one takes into account that the vehicle transporting cargo on the roof, where it is not known whether the vehicle is on a technical review was burdened with the same load as the time of the accident, nor is it known weight load. Since the weight load is not known, it is the mass of the vehicle at the time of an accident could be greater than the maximum allowable weight of the vehicle.

Example analysis of a traffic accident with the attached diagram of brake force which is not fixed vehicle weight

In a traffic accident in April 2008, there was a collision RENAULT KANGOO with pedestrian. In the scriptures were not given information on the type of engine, associated equipment and possible cargo in the hold and on the basis of an analysis of the investigating documents could not get to the data on the mass RENAULT. Consequences of traffic accidents were such that the pedestrian sustained injuries resulting in death.

Based on the attached diagram of brake force with extraordinary technical inspection performed on XXX, in the "XXX" in the vote, it has been found that the sum of the braking force RENAULT was 776 daN, as the braking force RENAULT were.

At front right wheel 246 daN — At front left wheel 248 daN
At back right wheel 144daN — At back left wheel 138 daN

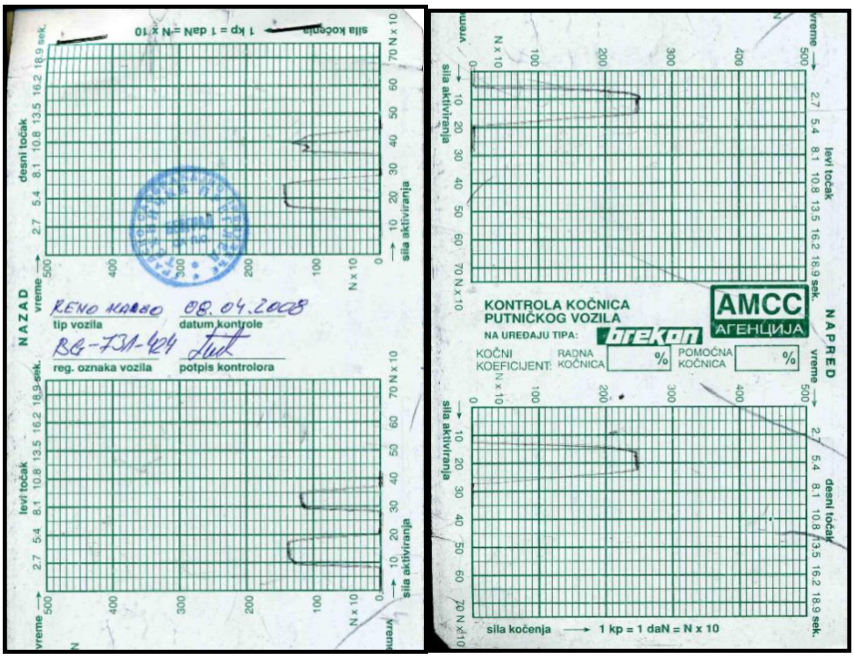
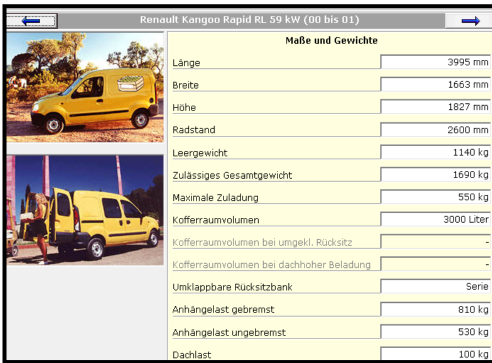


Figure 1 The diagram of braking force

However, the Commission of the Institute of Traffic Engineering in Belgrade, the file is not found information on weight RENAULT in the time of the extraordinary technical inspection. In fact, during the inspection, did not state that the mass-RENAULT, as well as whether RENAULT during the time of the accident and emergency technical inspection was loaded and how much.

At the database of vehicles Committee of the Institute of Transportation Faculty found that mass RENAULT KANGOO-may be 1050 kg to 1505 kg (see Figure 2), depending on the type of engine and ancillary equipment, and based on analysis of images could determine of the RENAULT mass.

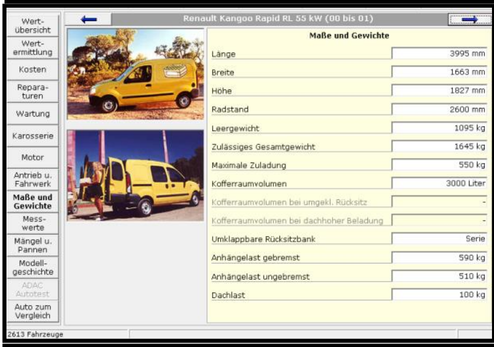


Renault Kangoo Rapid RL 59 kW (00 bis 01)

Maße und Gewichte	
Länge	3995 mm
Breite	1663 mm
Höhe	1827 mm
Radstand	2600 mm
Leergewicht	1140 kg
Zulässiges Gesamtgewicht	1690 kg
Maximale Zuladung	550 kg
Kofferraumvolumen	3000 Liter
Kofferraumvolumen bei umgekl. Rücksitz	-
Kofferraumvolumen bei dachhoher Beladung	-
Umklappbare Rücksitzbank	Serie
Anhängelast gebremst	810 kg
Anhängelast ungebremst	530 kg
Dachlast	100 kg

Renault Kangoo Rapid RL 43 kW (bis 01)

Maße und Gewichte	
Länge	3995 mm
Breite	1663 mm
Höhe	1827 mm
Radstand	2600 mm
Leergewicht	1050 kg
Zulässiges Gesamtgewicht	1600 kg
Maximale Zuladung	550 kg
Kofferraumvolumen	3000 Liter
Kofferraumvolumen bei umgekl. Rücksitz	-
Kofferraumvolumen bei dachhoher Beladung	-
Umklappbare Rücksitzbank	Serie
Anhängelast gebremst	780 kg
Anhängelast ungebremst	485 kg
Dachlast	100 kg

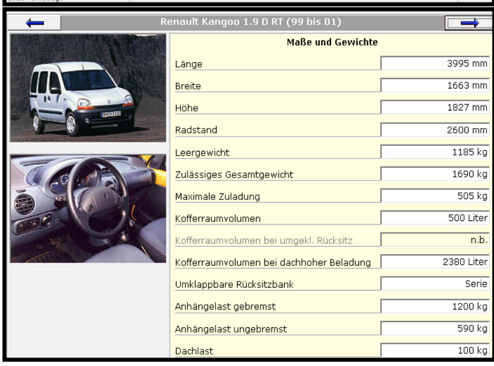


Renault Kangoo Rapid RL 55 kW (00 bis 01)

Maße und Gewichte	
Länge	3995 mm
Breite	1663 mm
Höhe	1827 mm
Radstand	2600 mm
Leergewicht	1095 kg
Zulässiges Gesamtgewicht	1645 kg
Maximale Zuladung	550 kg
Kofferraumvolumen	3000 Liter
Kofferraumvolumen bei umgekl. Rücksitz	-
Kofferraumvolumen bei dachhoher Beladung	-
Umklappbare Rücksitzbank	Serie
Anhängelast gebremst	590 kg
Anhängelast ungebremst	510 kg
Dachlast	100 kg

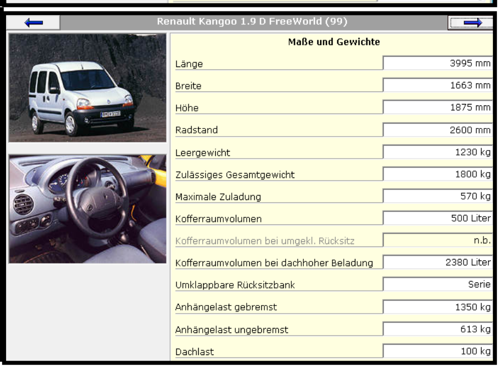
Renault Kangoo 1.9 dTi 4+2 (00)

Maße und Gewichte	
Länge	3995 mm
Breite	1663 mm
Höhe	1827 mm
Radstand	2600 mm
Leergewicht	1195 kg
Zulässiges Gesamtgewicht	1700 kg
Maximale Zuladung	505 kg
Kofferraumvolumen	500 Liter
Kofferraumvolumen bei umgekl. Rücksitz	n.b.
Kofferraumvolumen bei dachhoher Beladung	2380 Liter
Umklappbare Rücksitzbank	Serie
Anhängelast gebremst	1200 kg
Anhängelast ungebremst	595 kg
Dachlast	100 kg



Renault Kangoo 1.9 D RT (99 bis 01)

Maße und Gewichte	
Länge	3995 mm
Breite	1663 mm
Höhe	1827 mm
Radstand	2600 mm
Leergewicht	1185 kg
Zulässiges Gesamtgewicht	1690 kg
Maximale Zuladung	505 kg
Kofferraumvolumen	500 Liter
Kofferraumvolumen bei umgekl. Rücksitz	n.b.
Kofferraumvolumen bei dachhoher Beladung	2380 Liter
Umklappbare Rücksitzbank	Serie
Anhängelast gebremst	1200 kg
Anhängelast ungebremst	590 kg
Dachlast	100 kg



Renault Kangoo 1.9 D FreeWorld (99)

Maße und Gewichte	
Länge	3995 mm
Breite	1663 mm
Höhe	1875 mm
Radstand	2600 mm
Leergewicht	1230 kg
Zulässiges Gesamtgewicht	1800 kg
Maximale Zuladung	570 kg
Kofferraumvolumen	500 Liter
Kofferraumvolumen bei umgekl. Rücksitz	n.b.
Kofferraumvolumen bei dachhoher Beladung	2380 Liter
Umklappbare Rücksitzbank	Serie
Anhängelast gebremst	1350 kg
Anhängelast ungebremst	613 kg
Dachlast	100 kg

Renault Kangoo 1.9 dCi Privilège 4x4 (02 bis 03)

Maße und Gewichte	
Länge	3995 mm
Breite	1672 mm
Höhe	1894 mm
Radstand	2624 mm
Leergewicht	1405 kg
Zulässiges Gesamtgewicht	1830 kg
Maximale Zuladung	425 kg
Kofferraumvolumen	650 Liter
Kofferraumvolumen bei umgekl. Rücksitz	2600 Liter
Kofferraumvolumen bei dachhoher Beladung	n.b.
Umklappbare Rücksitzbank	Serie
Anhängelast gebremst	1150 kg
Anhängelast ungebremst	700 kg
Dachlast	100 kg

Renault Kangoo 1.5 dCi FAP Expression (08)

Maße und Gewichte	
Länge	4213 mm
Breite	1829 mm
Höhe	1839 mm
Radstand	2697 mm
Leergewicht	1505 kg
Zulässiges Gesamtgewicht	1982 kg
Maximale Zuladung	477 kg
Kofferraumvolumen	660 Liter
Kofferraumvolumen bei umgekl. Rücksitz	n.b.
Kofferraumvolumen bei dachhoher Beladung	2688 Liter
Umklappbare Rücksitzbank	Serie
Anhängelast gebremst	1050 kg
Anhängelast ungebremst	750 kg
Dachlast	100 kg

Figure 2Mass RENAULT KANGOO-a depending of type energy and ancillary equipment

If the mass of the RENAULT was 1055 kg and 1505 kg, then, with regard to the intensities of the brake force RENAULT, the braking ratio of RENAULT was:

$\mathrm{k} = (776 \cdot 10)/((1055 + 75) \cdot 9{,}81$, and $\mathrm{k} = (776 \cdot 10)/((1505 + 75) \cdot 9{,}81$

$\mathrm{k} = 0{,}7$, and $\mathrm{k} = 0{,}5$

and the deceleration would RENAULT able to achieve in real terms would be up to:

$b = 0{,}7 \cdot 9{,}81$, and $b = 0{,}5 \cdot 9{,}81$

$b = 6{,}9\ {}^{m}/_{s^2}$, and $b = 4{,}9\ {}^{m}/_{s^2}$

Bearing in mind that in the long skid marks brake from 20 to 30 m and speeds greater than 60 km/h, a decline in braking performance achieved up to 10%, it would RENAULT able to achieve deceleration to:

$b_1 = 6{,}87 \cdot 0{,}9$, and $b_1 = 4{,}91 \cdot 0{,}9$

$b = 6{,}2\ {}^{m}/_{s^2}$, and $b = 4{,}4\ {}^{m}/_{s^2}$

The RENAULT speed at the moment of a collision with a pedestrian would be 46.5 km/h (mass RENAULT times of 1055 kg), or 41.8 km/h (mass RENAULT times of 1505 kg) until the speed RENAULT and, at the beginning of skid marks brake were up to:

$V = \sqrt{\left(\frac{46{,}5}{3{,}6}\right)^2 + 2 \cdot 6{,}18 \cdot 20{,}5}$, and $V = \sqrt{\left(\frac{41{,}8}{3{,}6}\right)^2 + 2 \cdot 4{,}42 \cdot 20{,}5}$

$V = 20{,}5\ {}^{m}/_{s}\ or\ V = 73{,}8\ {}^{km}/_{h}$, and $V = 17{,}78\ {}^{m}/_{s}\ or\ V = 64\ {}^{km}/_{h}$

The RENAULT speed, at the time of response driver of RENAULT would be up to:

$V = \frac{73{,}8}{3{,}6} + \frac{6{,}18 \cdot 0{,}15}{2}$, and $V = \frac{64}{3{,}6} + \frac{4{,}42 \cdot 0{,}15}{2}$

$V = 20{,}96\ {}^{m}/_{s}\ or\ V = 75{,}5\ {}^{km}/_{h}$, and $V = 18{,}1\ {}^{m}/_{s}\ or\ V = 65{,}2\ {}^{km}/_{h}$

The RENAULT that the response of the driver RENAULT's braked to a place of collision with a pedestrian, crossed the path length:

$d = 20{,}96 + 20{,}5$,

$d = 41{,}5\ m$,

If the mass of RENAULT's was 1055 kg, then the speed RENAULT in response time of drivers, in which the driver had the opportunity to reacting in the same manner and from the same place stop RENAULT ago of a collision was up to:

$V = \sqrt{(6{,}18 \cdot 0{,}925)^2 + 2 \cdot 6{,}18 \cdot 41{,}46} - 6{,}18 \cdot 0{,}925$,

$V = 17{,}63\ {}^{m}/_{s}\ \ or\ V = 63{,}4\ {}^{km}/_{h}$, so the driver RENAULT's has a possibility of avoiding accidents when driving RENAULT times the speed limit to 60 km/h.

If the mass of RENAULT's was 1505 kg, then the speed RENAULT in response time of drivers, in which the driver had the opportunity to reacting in the same manner and from the same place stop RENAULT ago of a collision was up to:

$$V = \sqrt{(4{,}42 \cdot 0{,}925)^2 + 2 \cdot 4{,}42 \cdot 41{,}46} - 4{,}42 \cdot 0{,}925$$

$$V = 15{,}48\ {}^{m}/{}_{s}\ \ or\ V = 55{,}8\ {}^{km}/{}_{h}$$

so the driver of RENAULT’s would not have had the opportunity of avoiding accidents when driving RENAULT times the speed limit to 60 km/h.

If the mass of RENAULT's was 1055 kg, then the maximum speed RENAULT times could be up to 75.5 km/h, while the speed at which the driver RENAULT's a possibility of avoiding accidents, reacting to the same place and the same was up to 63.4 km/h. Given that the speed limit at the site of the accident is limited to 60 km/h, the driver would RENAULT's a possibility of avoiding accidents when driving RENAULT was up to the speed limit. Under these circumstances, the driver RENAULT times would stand deficiencies relating to the possibility of avoiding accidents.

If the mass of RENAULT's was 1505 kg, then the maximum speed RENAULT times could be up to 65.2 km/h, while the speed at which the driver RENAULT's a possibility of avoiding accidents, reacting to the same place and the same was up to 55.8 km/h. As to the speed of the scene of an accident is limited to 60 km/h, it RENAULT driver would not have had the opportunity of avoiding an accident or driving RENAULT's speed limits. Under these circumstances, the driver RENAULT would not have any failures related to the possibility of avoiding accidents and failures related to the contribution of an accident, but would eventually standing deficiencies relating to the weight of the consequences of this traffic accident.

Example analysis of a traffic accident for which there is attached a diagram of brake force participants vehicle accident

In a traffic accident in May 1998, there was a collision FIAT and OPEL, where the place of collision was located in the left lane, looking in the direction of FIAT-a. In the scriptures there were no data on the mass FIAT, while there were various statements about the cargo that was allegedly transported in the FIAT-in.

If it was located in the trunk FIAT’s load weight of 300 kg, and as mentioned driver FIAT, then, using the program PC Crash, speed FIAT`s the time of the collision with the OPEL was 61 km/h, and a speed OPEL would be 51 km/h.

If the trunk FIAT’s not located burden, and as mentioned witness, then, using the program PC Crash, speed FIAT’s the time of the collision with the OPEL was 64 km/h, and a speeds OPEL was 44.9 km/h.

As in the case file did not find information on whether the Mercedes was located burden and whether the brake force measured with or without cargo in the luggage compartment FIAT, given that the used mass FIAT of commercial licenses, it will be slowing FIAT vary. In fact, depending on whether, at the time of measurement of brake force in the FIAT was located a load of 300 kg or not, the braking coefficient FIAT would be:

$k = (720 \cdot 10)/((1230 + 75 + 300) \cdot 9{,}81)$, and $k = (720 \cdot 10)/((1230 +$

$75) \cdot 9{,}81)$

$k = 0{,}45$, and $k = 0{,}56$

the deceleration would FIAT able to achieve in real terms would be up to:

$b = 0{,}46 \cdot 9{,}91$, and $b = 0{,}56 \cdot 0{,}9$

$b = 4{,}5 \ {}^{m}/_{s^2}$, and $b = 5{,}5 \ {}^{m}/_{s^2}$

If you are located in a FIAT cargo weight of 300 kg, and as mentioned driver FIAT or if FIAT would not have located a burden, and as stated by the witness, then the speed FIAT at the beginning of skid marks brake were:

$V = \sqrt{\left(\frac{61}{3{,}6}\right)^2 + 2 \cdot 4{,}15 \cdot 15{,}8}$, and $V = \sqrt{\left(\frac{64}{3{,}6}\right)^2 + 2 \cdot 5{,}5 \cdot 15{,}8}$

$V = 20{,}72 \ {}^{m}/_{s} \ or \ V = 74{,}6 \ {}^{km}/_{h}$, and $V = 22{,}13 \ {}^{m}/_{s} \ or \ V = 79{,}7 \ {}^{km}/_{h}$

while the speed of the Mercedes in the time of the driver braking response was:

$V = 20{,}72 + \frac{4{,}5 \cdot 2}{2}$, and $V = 22{,}13 + \frac{5{,}5 \cdot 0{,}2}{2}$

$V = 21{,}17 \ {}^{m}/_{s} \ or \ V = 76{,}2 \ {}^{km}/_{h}$, and $V = 22{,}68 \ {}^{m}/_{s} \ or \ V = 81{,}6 \ {}^{km}/_{h}$

This example shows how incomplete data on the weight of cargo can affect the operation of the braking system, the vehicle speed at the time of the collision, but the definition of failure of participants of a traffic accident. Namely, if the FIAT during the technical inspection located the burden of a weight of 300 kg and used mass FIAT of vehicle licenses, then with respect to the measured braking force, braking system FIAT was inoperative of law, because the braking coefficient FIAT was 0.45, and the minimum value of the braking coefficient is 0.5. With the value of the braking coefficient of 0.45 FIAT by the time response of drivers FIAT braking had a speed of 76.2 km/h, and with regard to the place of accident speed was limited to 80 km/h, the driver's FIAT would any failures related to the occurrence of accidents and the severity of consequences.

On the other hand if the FIAT during the technical review would not be located burden if used mass FIAT of vehicle licenses, then with respect to the measured braking force, braking system FIAT was correct, because the braking coefficient FIAT was 0.56, a minimum value of the braking coefficient is 0.5. However, the value of the braking coefficient of 0.56 FIAT by the time response of drivers FIAT braking had a speed of 81.6 km/h, and with regard to the place of accident speed was limited to 80 km/h, it would be on the driver's FIAT stay failures relate to the weight of the consequences of traffic accidents.

Example analysis of a traffic accident for which there is not attached a diagram of brake force vehicles where the vehicle transporting cargo on the roof

In May 2007, a traffic accident that involved a passenger car, LADA and a pedestrian has occurred. On the roof LADA is transported cargo unknown mass, because on the basis of available data from the file, the mass load from the roof LADA could not be reliably determined. The accident occurred when pedestrian collision and the front left part of LADA and pedestrian head impact and load from the roof of LADA. At the scene of an accident fixed skid marks brake of LADA are found.

The question was whether the weight of the vehicle with a load higher than the maximum permissible weight of the vehicle, as in the writings there were no precise data on the weight of cargo that is transported LADA, as well as data on measuring the brake force LADA or deceleration that could achieve braking LADA system, and it is not possible to determine during the investigation of a traffic accident. But the fact that they are on the road the other skid marks brake was sufficient to establish that the weight of the vehicle at the time of an accident, including the weight load on the roof, was not greater than the maximum allowable weight of the vehicle.

In fact, bearing in mind that as a result of reaction LADA driver braked wheels LADA were blocked in the process of braking (with cargo on the roof), we find that the braking system LADA at the time of occurrence of traffic accidents could achieve deceleration to achieve locked wheels on asphalt. If the LADA was overloaded, then the brake system could cause wheels to brake LADA in the process of block on asphalt, because it would force the scope point was greater than the braking force at the wheels from locking, so that in these circumstances the possibility that the braking system LADA not realize deceleration provided by roadway in the area of traffic accidents.

An example presented shows that on the basis of a limited of data and the very specific circumstances of traffic accidents, possible to determine a fact which is of great importance for the elucidation of all the circumstances of the car accident.

CONCLUSION

Presented paper has emphasized the importance of precise calculation of deceleration and precise data collection in investigations. Paper presented as examples of deceleration depends on the budget of the other parameters, which are important for the analysis of accidents, such as speed, the characteristic position of the participant accidents in certain stages of the collision and determine the possibility of avoiding accidents, but also situations where precise identification speed is essential for accurate determination of the collision. It is extremely important that traffic expert know the rules of proper calculation of deceleration of the vehicle, as well as rules regarding the minimum legal requirements of proper braking system of the vehicle.

Traffic experts must take into account the conditions of the brake system, which prescribes the rules at the time of occurrence of traffic accidents. In addition, it can be concluded that no matter what the expert has the possibility of calculating the deceleration must take into account all the other circumstances of the accident, in order to accurately and precisely determine the deceleration of the vehicle in this case. Was presented and the way they are based on a limited number of data in the file in some cases, can determine the elements of importance for the analysis of traffic accidents. Deceleration of the vehicle can be determined without knowing the braking force, but again must take into account all the circumstances of the car accident.

If the expert does not have the necessary data, deceleration, as a rule, is defined by the so-called. 'law of the minimum', according to which the vehicle deceleration to achieve the minimum of the defined deceleration according to all the circumstances (braking force, road conditions, weather conditions, the existence of skid marks and the like.).

REFERENCES

[1] Vujanić, M., Problems in traffic safety with the practicum, Faculty of Traffic and Transport Engineering, Belgrade, 2001

[2] Vujanić, M., Antić, M., Problems in traffic safety with the practicum part II, Faculty of Traffic and Transport Engineering, Belgrade 2006

[3] Dragač, R., Vujanić M., Traffic Safety Part II, Faculty of Traffic and Transport Engineering, Belgrade 2002

[4] Vujanic, M. et al., Handbook for traffic-technical expertise and assessment of damage, MODULE, Banja Luka, 2000

[5] Regulation on the division of motor vehicles and trailers and the technical conditions of vehicles in traffic, Official Gazette Republic of Serbia, no. 64/10, 69/10, 2010th

[6] Regulations on the size, total masses and axle loads of vehicles and the basic conditions that must be met by devices and equipment on vehicles in traffic, Official Gazette of the SFRY no. 50/82, 11/83, 4/85, 65/85, 64/86, 22/90, 50/90 and 51/90.

[7] Vujanic, M. et al. Commentary on Road Traffic Safety, Official Gazette, Belgrade 2009

[8] Expertise traffic accident of Institute of Faculty of Traffic and Transport Engineering

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EFFECTIVE APPROACH TO ANALYTICAL, ANGLE RESOLVED SIMULATION OF PISTON–CYLINDER FRICTION IN IC ENGINES

Slobodan Popović[1], *Nenad Miljić, Marko Kitanović*

UDC:536.8

ABSTRACT: Most frequently used approach to describe, model and analyse engine effective performance and mechanical losses is largely based on global, time/angle averaged empirical friction models. Importance of detailed, systematic approach to simulation of motor car and its real-world behaviour in terms of combustion efficiency and overall fuel consumption is greater than ever. Therefore, commonly employed empirical or semi-empirical approaches seem insufficient in terms of precision and capabilities for global optimisation of power train system which poses a need to invest more attention into comprehensive angle-resolved, analytical models. Such models have been already presented elsewhere, mainly based on fundamental friction theory of Stribeck, however, simplifications in both slider mechanism dynamic and combustion simulation influenced insufficient accuracy and reliability. In this paper, detailed Stribeck's theory based analytical approach has been employed to model friction in contact of piston ring assembly to cylinder (PRAC) and piston skirt to cylinder (PSC), as well. Gas pressure trace was measured and used to predict instantaneous indicated and effective torque based on engine 1DoF dynamic model and friction models presented in paper.

KEY WORDS: engine, friction model, instantaneous crankshaft speed

EFEKTIVAN PRISTUP ANALITIČKIM, SIMULACIJA REŠENOG UGLA KLIPNOG TRENJA U IC MOTORIMA

REZIME: Najčešće korišćen pristup za opisivanje, model i analizu performansi efikasnosti motora i mehaničke gubitke je uglavnom zasnovan na globalnom, vremenskom uglu prosečnog empirijskog modela trenja. Značaj detaljnog, sistematskog pristupa simulaciji motora automobila i njegovog ponašanja u stvarnom svetu u smislu efikasnosti sagorevanja i ukupnoj potrošnji goriva, veći je nego ikada. Stoga se, obično korišćeni empirijski ili polu-empirijski pristupi čine nedovoljnim u smislu preciznosti i mogućnosti za globalnom optimizacijom pogonskog sistema koji ima potrebu ulaganja sa više pažnje u sveobuhvatne strane rešenih analitičkih modela. Takvi modeli su već prezentovani na drugim mestima, uglavnom zasnovani na osnovnoj Stribekovoj teoriji trenja, međutim, pojednostavljenja u oba klizna mehanizma dinamike i simulacije sagorevanja su uticala na nedovoljnu tačnost i pouzdanost. U ovom radu, detaljna Stribekova teorija zasnovana na analitičkom pristupu je uposlena da modelira trenje u kontaktu sklopa klipa prstenova i cilindra (PRAC) i klipnjače i cilindra (PSC) takođe. Trag pritiska gasa je izmeren i korišćen za predviđanje trenutno naznačenog i efikasnog obrtnog momenta na osnovu motora jednog stepena slobode dinamičkog modela i trenja modela prikazanog u radu.

KLJUČNE REČI: motor, trenje modela, trenutna brzina radilice

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EFFECTIVE APPROACH TO ANALYTICAL, ANGLE RESOLVED SIMULATION OF PISTON–CYLINDER FRICTION IN IC ENGINES

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UDC: 536.8

INTRODUCTION

Considering constantly growing importance of the analytical approach to the Internal Combustion Engine (ICE) design, virtual prototyping and combustion dynamics optimisation, highly sensitive and accurate simulation model of the engine is necessary to predict and evaluate fuel economy and overall performance in both stationary and transient operation. Cyclic nature of the combustion process taking place in engine cylinder(s) and crank slider mechanism dynamics are directly reflected in instantaneous engine torque, both indicated and effective one, which is further seen as an input to vehicle transmission, as a most common case, or to any other machine using an ICE as a prime mover. The information on highly dynamic nature of the ICE is, therefore, contained in the data obtained through angle based measurement of instantaneous engine torque and crank shaft angular speed. Being a highly dynamic and non-linear system, ICE could be presented and effectively simulated only by means of non-linear simulation models. Such a model can be applied as to provide a reliable prediction of crankshaft response to cycling nature of combustion process, both in terms of effective torque and angular velocity, or to evaluate and analyse combustion process through reverse approach based on measurement of instantaneous angular speed.

The ICE simulation model prediction capabilities largely depend on the correct and detailed approach to the modelling of mechanical losses. Such a model should cope effectively with complex and diverse phenomena related to energy dissipation in friction generated in contacts of adjacent engine components in relative motion (i.e. piston rings and skirt, bearings, cams, tappets, valve stems, gears) and to energy used to drive engine auxiliaries (i.e. coolant and lubricant circulation pumps, fuel supply, electrical generator).

The class of fully empirical, global, time-averaged models represent the simplest and most commonly used approach to this problem. Models proposed by Chen and Flynn [1], Yagi et al. [2] and Milington and Hartles [3] are good examples for this class of models, however, prediction capabilities are rather poor without proper model calibration. Component, semi-empirical, time-averaged models, such as that of Sandoval [4] or Arsie et al. [5], provide an engineer with simple, yet effective tool for evaluation of design parameters and their influences on mechanical losses. Obviously, the nature of mechanical losses differs largely and therefore, global, particularly fully empirical models provide

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satisfactory solution only in terms of modelling of BMEP and mean effective torque. Considered the modelling of engine dynamic response this approach seems insufficient, and different phenomena must be analysed and modelled using different approaches and techniques.

The friction processes represent a dynamic component of mechanical losses, which implies an angle-based approach to modelling. Quite opposite to that, the power used to drive engine auxiliaries is commonly regarded as a stady-state component and can be simulated by time averaged, global models. According to Taylor [6], the friction in Piston Ring Assembly–Cylinder (PRAC) contact dominates contributing the mechanical losses of an ICE by approximately 40–50%, following by Piston Skirt–Cylinder (PSC) contact friction with up to 25%. The distribution of other components is as follows: engine bearings 20–30%, valve train 7–15%, and auxiliaries with 20–25%. Considering these numbers, the friction originating in PRAC and PSC contacts are the most influential, and therefore, the special attention should be paid to this particular engineering problem. Investing a time and effort in development of fully analytical, dynamic, angle-resolved models based on fundamental lubrication theories is therefore, more than justified.

This work focuses mainly on the losses due to friction generated in piston–cylinder contact. The Piston Ring Assembly–Cylinder (PRAC) and Piston Skirt–Cylinder (PSC) contacts are analysed, modelled and presented separately in detail. Other phenomena, namely losses in crank and cam shaft bearings, valve train and auxiliaries are briefly presented as well in order to provide a full overview of the model structure. However, keen reader is highly advised to consult extensive literature for more detailed approach to these topics. Model verification based on comparison of simulated and measured engine crankshaft angular speed was presented in the final section.

ENGINE FRICTION MODEL STRUCTURE OVERVIEW

Piston Ring Assembly–Cylinder Friction sub model

The number of simulation models have been presented in literature in the past and used more or less effectively to evaluate diverse design and process parameters influences to dynamic components of mechanical losses in ICE. From theoretical standpoint of view, the most complete and comprehensive models covering the friction in PRAC is that based on solution of Reynolds equation. More details on this extensive numerical approach were overviewed by Taylor [6], Stanley [7] and Livanos [8,9]. In engineering applications, these models are hardly to be considered as a first choice, having in mind extensive and complex computational effort needed to provide an engineer with data necessary for engine design and overall performance evaluation. Far more interesting and promising seems the approach based on Stribeck's theory. This approach is widely used for a period of time and well documented by Ciulli [10], Guzzomi [11],[12], Taraza [13][14], Zweiri [15],[16], Rakopulos [17], Kouremenos [18], Thring [19] and Lin [19]. Although based on same theoretical principles, models referenced here differ in number of details such as engine class to which the model is adopted (ship propulsion [17][18]), interpretation of piston ring geometry [7],[13][14], or in number of empirical constants [10],[13][16]. Ciulli et al. [10] recognize mixed and hydrodynamic lubrication as predominant and suggested approach based on Stribeck's theory. Stribeck's number (duty parameter), in general form can be defined as follows [6,7,21,22]:

$$S = \frac{\eta \cdot |v|}{W}, \tag{1}$$

where η is lubricant dynamic viscosity, v is instantaneous relative velocity and W normal specific load per unit length. Friction coefficient μ relies on simple relations based on duty parameter S and coefficients C and m which are to be identified for each specific case:

$$\mu = C \cdot S^{m}, \tag{2}$$

The model incorporates a number of empirical constants necessary to calculate friction coefficient for two regimes of mixed and hydrodynamic lubrication which are supposed to be predominant in piston–cylinder contact:

$$\mu_{pr,i}(\alpha) = \begin{cases} \dfrac{C_1 - C_2}{C_3} \cdot S_{pr,i}(\alpha) + C_2, & S_{pr,i}(\alpha) \le C_3 \\ C_4 \cdot \left(\sqrt{S_{pr,i}(\alpha)} - \sqrt{C_3} \right) + C_1 & S_{pr,i}(\alpha) > C_3 \end{cases}, \tag{3}$$

Coefficient C_3 has a physical background representing critical value of duty parameter Scr. Other constants are engine design and geometry related and must be identified experimentally. According to Stanley [7] and Taraza [13], oil film formation and its thickness (OFT) are predominantly influenced by piston ring geometry, namely ring curvature, but almost independent of the ring axial thickness. The piston ring curvature parameter (PRCP) is defined as the ratio of the ring profile recess “c” at the ring edge to the height “a” of the parabolic profile. The PRCP is presented in Figure 1. The PRCP varies within the range of 0.03 to 0.15, with an optimal value of 0.06. It is also shown in [7] that lubrication conditions in PRAC contact, within the engine working cycle, change from boundary in the vicinity of both TDC and BDC, via mixed to fully developed hydrodynamic lubrication around each piston mid stroke. General behaviour of Oil Film Thickness (OFT) indicating such lubrication scheme is presented in Figure 2.

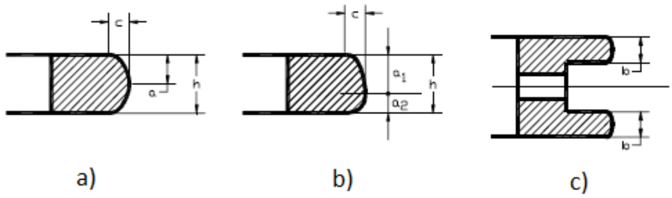

Figure 1 Piston ring curvature: a) top compression ring; b) second ring; c) oil ring

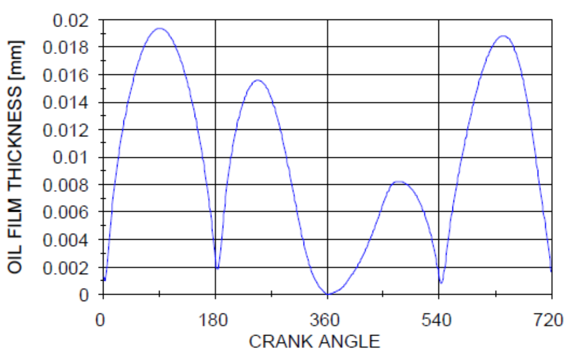

Figure 2 OFT variation during an engine cycle for the top ring [13]

According to these observations, separate angle-based sub-models can be set for each lubrication regime. Based on a Stribeck diagram presented in **Error! Reference source not found.**, for regime of boundary lubrication, dry friction coefficient of the two rubbing metal surfaces μ_o can be used. Assuming that the values of S_{cr}, S_o and μ_o are known in advance (accessible from literature), the mixed lubrication friction coefficient can be retrieved from equation:

$$\ln \mu_{mix} = \frac{\ln \mu_{cr} - \ln \mu_o}{\ln S_{cr} - \ln S_o} \cdot \left(\ln S - \ln S_{cr}\right) + \ln \mu_{cr}, \quad (4)$$

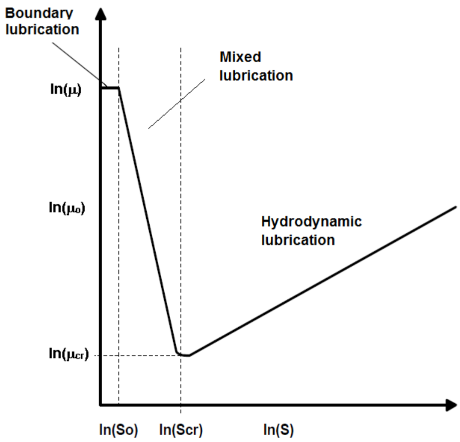

Figure 3 The friction coefficient dependance on duty parameter (simplified Stribec's curve)

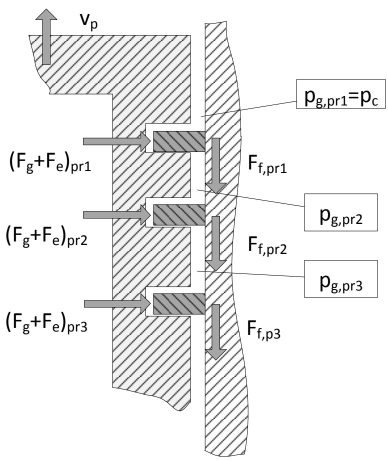

Figure 4 Piston ring curvature: a) top compression ring; b) second ring; c) oil ring

Summarizing previous equations, and applying general annotation to each piston ring of PRA, the model can be presented in angular domain as follows:

$$\mu_{pr,i}(\alpha)=\begin{cases}\mu_o & , \quad S_{pr,i}(\alpha)\le S_o \\ \mu_o\cdot\left(1-\dfrac{S_{pr,i}(\alpha)}{S_{pr,cr}}\right)+\mu_{cr}\cdot\dfrac{S_{pr,i}(\alpha)}{S_{pr,cr}} & , \quad S_{pr,i}(\alpha)\le S_{cr} \\ C_{pr}\cdot S_{pr,i}(\alpha)^{m_{pr}} & , \quad S_{pr,i}(\alpha)>S_{cr}\end{cases}, \tag{5}$$

Recommended numerical values for constants μ_o, μ_{cr}, S_o, S_{cr}, C_{pr} and m_{pr} can be found in literature [7,13,14].

Duty parameter must be accommodated as to incorporate dependences on relevant PRA design and process parameters. From equation (1), one can retrieve expression for duty parameter for a given crank shaft angular position introducing $\eta(T_l)$ as lubricant dynamic viscosity dependant on lubricant temperature T_l, instantaneous piston velocity $v_p(\alpha)$ and piston ring normal load per unit length $W(\alpha)$. The most convenient way to express piston ring load per unit length W is to introduce the pressure acting on the piston ring $p_{pr,i}(\alpha)$ and corresponding piston ring axial thickness $a_{pr,i}$.

$$S_{pr,i}(\alpha)=\frac{\eta(T_l)\cdot|v_p(\alpha)|}{W(\alpha)}=\frac{\eta(T_l)\cdot|v_p(\alpha)|}{p_{pr,i}(\alpha)\cdot a_{pr,i}}, \tag{6}$$

Film thickness OFT, according to [17], can be expressed as a simple function of duty parameter and piston ring height.

$$OFT_{pr,1}=S_{pr,i}\cdot a_{pr,i}, \tag{7}$$

The load acting on inner surface of each piston ring incorporates the effects of tangential force due to elastic nature of the piston ring and that of gas pressure coming from combustion chamber. The basic principle of piston ring friction mechanism is presented in **Error! Reference source not found.**. According to [10] and [16], tangential force $F_{Tpr,i}$ and corresponding pressure $p_{Epr,i}$ are defined as follows:

$$F_{Tpr,i}=\frac{E_{pr}\cdot g_{o,i}\cdot h_{pr,i}\cdot\left(\dfrac{w_{pr,i}}{D}\right)^3}{14.14\cdot\left(1-\dfrac{w_{pr,i}}{D}\right)^3}, \tag{8}$$

$$p_{Epr,i}=\frac{2\cdot F_{Tpr,i}}{D\cdot h_{pr,i}}=\frac{E_{pr}\cdot g_{o,i}\cdot\left(\dfrac{w_{pr,i}}{D}\right)^3}{7.07\cdot D\cdot\left(1-\dfrac{w_{pr,i}}{D}\right)^3}, \tag{9}$$

Parameters are defined as follows:

E_{pr} - Young's modulus of elasticity for piston ring material
$g_{o,i}$ - Piston ring gap in open condition
D - Nominal diameter (cylinder/piston)
$w_{pr,i}$ - Piston ring radial thickness (piston ring width)
$h_{pr,i}$ - Piston ring axial thickness (piston ring hight).

The resultant force acting on each piston ring incorporates a gas pressure which is particularly influential in the region of the compression piston ring. The expressions for resultant force $F_{pr,i}$, corresponding pressure $p_{pr,i}$ and friction force $F_{Fpr,i}$ are:

$$F_{pr,i}(\alpha)=\left[p_{Epr,i}+\left(p_{g,i}(\alpha)-p_o\right)\cdot\left(1-2\cdot\frac{w_{pr,i}}{D}\right)\right]\cdot\pi\cdot D\cdot h_{pr,i}\,, \tag{10}$$

$$p_{pr,i}(\alpha)=p_{Epr,i}+\left(p_{g,i}(\alpha)-p_o\right)\cdot\left(1-2\cdot\frac{w_{pr,i}}{D}\right), \tag{11}$$

$$F_{Fpr,i}(\alpha)=\mu_{pr,i}(\alpha)\cdot F_{pr,i}(\alpha), \tag{12}$$

Gas pressure acting upon inner surface of each piston ring $p_{g,i}(\alpha)$ (Figure 4) is an unknown variable and can be determined by means of separate nonlinear thermodynamic model which can include gas leakage through each piston ring chamber and must be supported alternatively by in-cylinder pressure measurement or combustion process model. Such a complex approach has been presented by Dowson [6] and Wannatong [24]. However, much simpler, yet effective solution can be based on assumption that the pressure in the first piston ring chamber is almost equal to that in combustion chamber, while in the second and third piston ring chambers pressure is further reduced by a rate of approximately 50% and 90% respectively. This approach introduces a small angular phase shift in pressure traces compared to solution proposed by Dowson, particularly in the third piston ring chamber. However it can be neglected, having in mind a considerably small influence of the friction on the third piston ring.

Piston skirt–Cylinder Friction sub-model

Lubrication in Piston Skirt–Cylinder (PSC) contact is commonly assumed hydrodynamic. On that assumption, sub-model proposed for modelling friction in PRAC contact where duty number is higher than critical value will be used. Normal load is introduced through normal force F_N and piston skirt width wps which, depending on piston design features can incorporate also piston pin recesses.

$$\mu_{ps}(\alpha)=C_{ps}\cdot S_{ps}(\alpha)^{m_{ps}}=C_{ps}\cdot\left[\frac{\eta(T_l)\cdot\left|v_p(\alpha)\right|}{p_{ps}(\alpha)\cdot h_{ps}}\right]^{m_{ps}}=C_{ps}\cdot\left[\frac{\eta(T_l)\cdot\left|v_p(\alpha)\right|\cdot w_{ps}}{F_N(\varphi)}\right]^{m_{ps}} \tag{13}$$

Lubricant viscosity variations with temperature

Friction force in PRAC and PSC contacts as well in radial HD bearings, according to expressions (1)-(13) depend upon a lubricant dynamic viscosity which, in also depends upon lubricant temperature. This is of secondary importance if steady-state operation with stabilized lubricant temperature assumed, and viscosity can be determined from lubricant specifications. For full operating range this is insufficient, particularly if one takes into consideration a lubricant temperature rise due to the load and friction in surface contacts. According to Klaus [26], local increase in lubricant temperature of 50-60 K in radial HD bearings are highly expected.

The viscosity temperature relationship is commonly approximated by the ASTM, or simply MacCoull-Walther's equation [23], or MacCoull-Walther-Wright's equation [27]. More convenient approach is proposed by Manning based on Wright's equation [27]:

$$\log_{10}(\log_{10}(Z)) = A - B \cdot \log_{10}(T_l), \tag{14}$$

accompanied by equation for direct calculation of kinematic viscosity ν(Tl)

$$\nu(T_l) = (Z - 0.7) - e^{f(Z)}, \tag{15}$$

Function argument Z and complex polynomial function f(Z) are well documented by Seeton [27], while coefficients A and B are determined by solving a system of equations given for two discrete values of viscosity and corresponding temperatures known from manufacturer's specification.

SIMULATION RESULTS

Piston Ring Assembly and Skirt Friction

The model components presented in previous section has been tested in order to check sensitivity and to provide basic conclusions necessary for further refinement and calibration. In-cylinder pressure data measured in firing engine were used to demonstrate models behaviour. The analysis was conducted for series-production PFI petrol engine DMB M202PB13 (technical data given in Table 1).

Table 1 Main engine specification

Description	Value
Engine manufacturer	DMB
Engine type	SI, MPI, M202PB13
Bore/Stroke	80.5/67.4 mm
No. of cylinders	4
Compression ratio	9.2 (+0.2/–0.1)
Max. power	52 kW @ 5800 min–1
Cooling system	liquid
Fuel system	Port Injection

Table 2 Operating points

Engine Speed	Load (BMEP)
cca. 1810 min–1	8.13 105 Pa
cca. 1810 min–1	3.09 105 Pa
cca. 2810 min–1	3.09 105 Pa

The first part of analysis was performed as to demonstrate basic influences of engine speed and load to OFT, friction coefficient and friction force in the piston ring–cylinder liner contact. The compression ring was chosen as a representative case, having in mind its dominant contribution in the overall friction in PRAC. Operating points chosen for this analysis were presented in Table 2. **Error! Reference source not found.** illustrates the dependency of the first piston ring (compression ring) OFT for 3 different operating points. OFT increases with an increase in engine speed and consequently piston speed. Provided the higher loads (higher BMEP) for a given constant engine speed, OFT decreases. This is expected because higher pressure in the first piston ring chamber (equal to the measured in-cylinder pressure) provides higher load to the back surface of the ring, and so, decreases the quantity of the lubricant in the contact. Both observations correspond with the results provided with more complex models based on Reynolds equation and experiments [6-9]. According to model, OFT riches minimum, i.e. zero values at each dead centre where piston changes direction and therefore, its speed is zero. This result cannot be considered entirely correct because it implies that lubricating oil vanishes from contact. The model, due to its simplicity, is not fully capable to predict minor phase shift due to transient hydrodynamic effects which is reported by Stenley [7]. However, model indicates correctly the regions where lubricating conditions are critical.

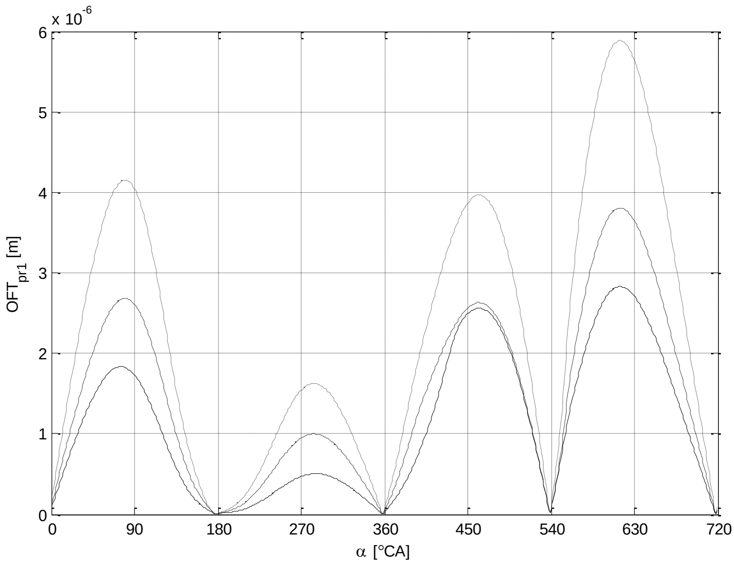

Figure 5 Oil Film Thickness (OFT) for compression piston ring: cca. 30% load @ 1800 min^{-1} – dashed line; cca. 100% load @ 1800 min^{-1} – solid line; cca. 30% load @ 2800 min^{-1} – dotted line;

The friction coefficient dependency on engine speed and load for the compression ring is presented in Figure 6. It follows the behaviour of OFT, and decrease in friction coefficient due to increased load, as well as lower values with increased speed are observed. Model indicates high gradients of friction coefficients in the regions close to each dead centre, which is expected because of the change in piston velocity.

The Figure 7 illustrates the friction force in PRAC. According to the observations presented for friction coefficient, intensive and sudden changes in friction force are indicated around each dead centre. However, the most dominant gradients are reported in the vicinity of the firing TDC, which is expected due to the effect of the rising pressure due to the combustion. The influence of piston speed is visible, but significantly smaller than that of the engine load. These results correspond well with basic findings reported by other authors [6-10,13,17-20]. Results are indicative, however further refinements for smooth transitions from mixed to boundary lubrication and vice versa, are needed.

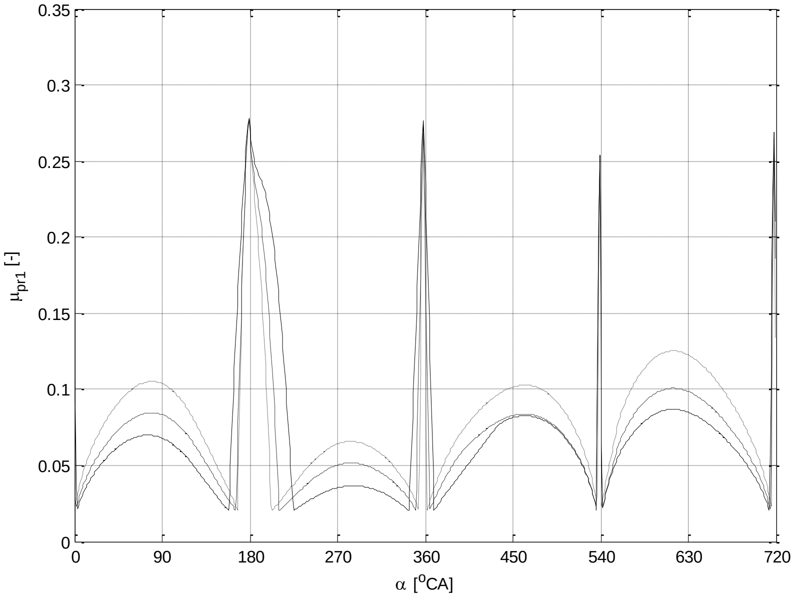

Figure 6 Friction coefficient in PRAC (compression piston ring): cca. 30% load @ 1800 min^{-1} – dashed line; cca. 100% load @ 1800 min^{-1} – solid line; cca. 30% load @ 2800 min^{-1} – dotted line;

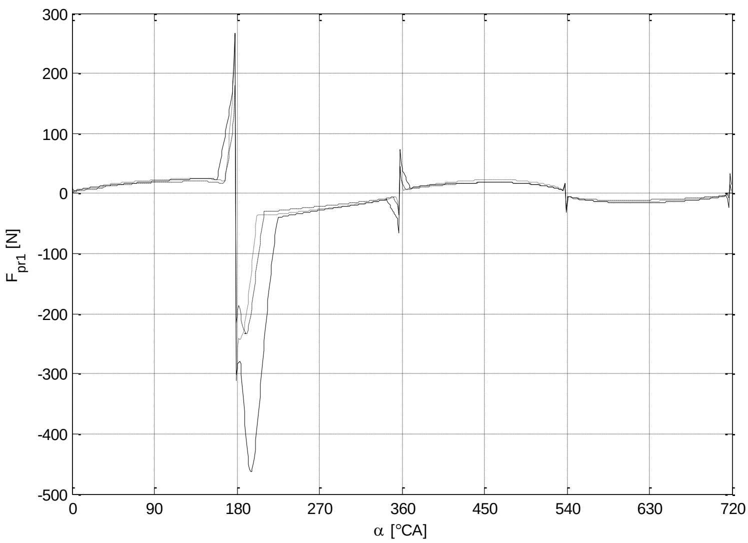

Figure 7 Friction force in PRAC (compression ring): cca. 30% load @ 1800 min^{-1} – dashed line; cca. 100% load @ 1800 min^{-1} – solid line; cca. 30% load @ 2800 min^{-1} – dotted line;

The influence of engine speed and load on friction force in PSC contact is presented in Figure 8. As expected, the increase in friction force is strongly related to the changes of normal force which pushes piston to the surface of the cylinder liner. The piston skirt friction force develops as normal force increases after the engine firing, therefore, angular phase shift in respect to the friction force in PRAC contact is clearly visible. The influence of engine load is visibly stronger than that of piston speed. The effect of piston skirt friction force is, however, of the marginal importance during exhaust and intake strokes, and for simplicity, can be neglected.

Total friction torque per cylinder, incorporating friction in PRAC and PSC contacts is presented in **Error! Reference source not found.**. Basically, the model indicates that the friction due to normal force in PSC contact dominates. The influence of combustion around TDC is smaller due to the effect of crank slider mechanism dynamics. This diagram indicates that model provides solid information on basic both geometric and process parameters which influences friction in piston–liner assembly, and which can be regarded as an optimisation objective.

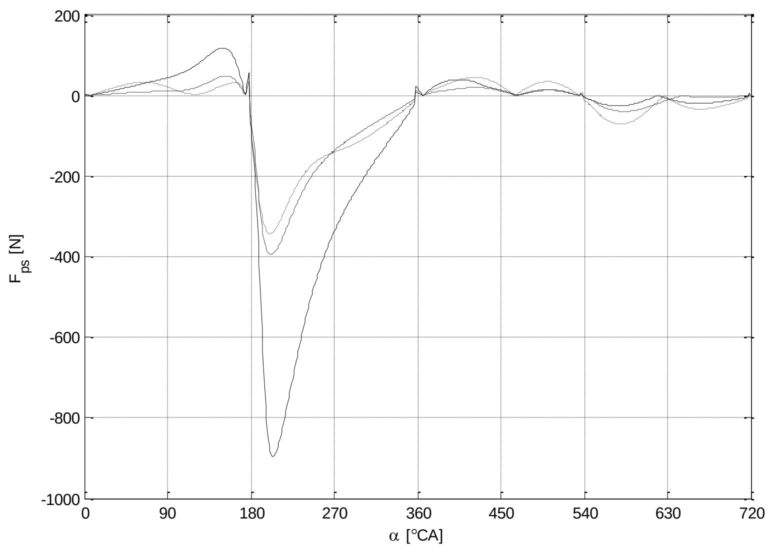

Figure 8 Friction force in PSC: cca. 30% load @ 1800 min^{-1} – dashed line; cca. 100% load @ 1800 min^{-1} – solid line; cca. 30% load @ 2800 min^{-1} – dotted line;

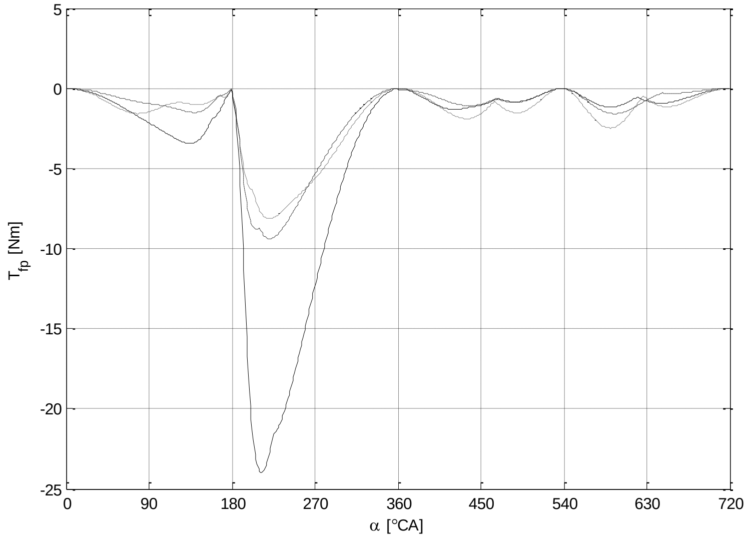

Figure 9 Total friction torque in PRAC and PSC contact per cylinder: cca. 30% load @ 1800 min^{-1} – dashed line; cca. 100% load @ 1800 min^{-1} – solid line; cca. 30% load @ 2800 min^{-1} – dotted line;

MODEL VERIFICATION

Piston-cylinder liner friction model was used to compare simulated and measured values of instantaneous crankshaft angular speed. As an input to simulation model, in-cylinder pressure was provided experimentally. However, in order to provide correct simulation, an extension to friction losses in piston–cylinder contact model must be provided as to encounter friction in bearings, valve train and power for engine auxiliaries.

Friction in bearings

Engine bearings (crank and cam shafts) are exposed to variable loading which reflects dynamic nature of the combustion process and geometric features of the crank slide and cam-tappet mechanisms. Being responsible for approximately 20% of total losses, special attention must be paid to the modelling of phenomena related to hydrodynamic lubrication in radial bearings. In order to provide effective and accurate calculation, model must incorporate gas pressure force as an input, friction force in piston–cylinder contact, inertia forces acting on both rotating and reciprocating masses which, through iterative approach can lead to a bearing loading force determination. Resultant force acting in connecting rod bearing is determined through clear approach. However, resultant force in main bearings of a multi-cylinder engine which is the most often case, poses a number of issues related to load distribution along the crankshaft which is generally considered as statically underdetermined structure. According to [25], reactions on bearings situated outside the crank on which a force is applied are low due to the shaft stiffness. Accordingly, crankshaft/camshaft can be assumed as a structure consisting of successive sections (cranks or cams, depending on the case) transmitting only torque, while the resulting radial load in main bearings can be approximated as a sum of load reactions in adjacent cranks/cams.

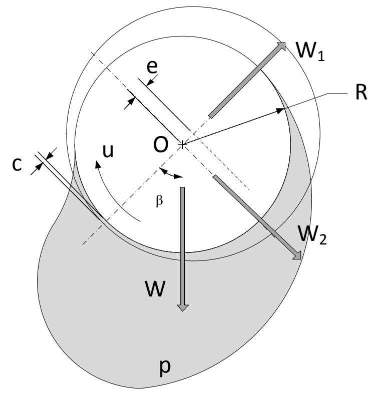

Figure 10 Pressure profile and load distribution in radial HD bearing

The mobility method proposed by Booker [6], which is based on OFT determination, provides quite satisfactory results in terms of computational effort and accuracy, and will be deployed here as well. Complete, general, theoretical approach is well documented by Stachowiak [23]. For the purpose of current study, this method will be shortly reviewed with the emphasis on basic components and necessary corrections which must be applied for short dynamically loaded bearing. In order to establish the relationship between total bearing load, which comes from engine crank slide mechanism dynamic calculation model, and basic geometry of a bearing, expression for instantaneous load capacity of the bearing W will be used based on general analytical approach [23].

$$W = \sqrt{W_1^2 + W_2^2} = \frac{u \cdot \eta(T_L) \cdot \varepsilon \cdot L^3}{c^2 \cdot (1-\varepsilon^2)^2} \cdot \frac{\pi}{4} \cdot \sqrt{\left(\frac{16}{\pi^2} - 1\right) \cdot \varepsilon^2 + 1}, \tag{16}$$

The expression corresponds to the bearing disposition presented in Figure 10. Parameters are defined as follows: u represents the instantaneous journal velocity relative to bearing, c is the bearing clearance, ε is the relative eccentricity of the journal in the bearing (e/c). The attitude angle β between the load line and the line of centres (Figure 10) can be determined directly from the load components W_1 and W_2 from the following relation:

$$\beta = \tan^{-1}\left(-\frac{W_2}{W_1}\right) = \tan^{-1}\left(\frac{\pi}{4} \cdot \frac{\sqrt{1-\varepsilon^2}}{\varepsilon}\right), \tag{17}$$

Considering engine shaft radial bearing a short one, friction force F_{Fb} can be determined from model proposed by Ocvirk [13][14][23].

$$F_{F,b} = \left[\frac{2 \cdot \pi \cdot \eta(T_l) \cdot u \cdot L \cdot R}{c \cdot \sqrt{1-\varepsilon^2}}\right]_1 + \left[\frac{e \cdot W \cdot}{2 \cdot R} \sin\beta\right]_2 + \left[\frac{v \cdot W}{u} \cdot \cos\gamma\right]_3, \tag{18}$$

The friction force equation introduces three components, the first of which is dominant and represents the influence of shearing force of the lubricant film, while the second one simulates the contribution of the torque due to the eccentric load. The third term represents a correction for dissipation due to the journal movement which can be neglected for quasi-static conditions. The model considered at this point is a quasi-steady-state model assuming established equilibrium values for the unknown relative eccentricity ε and attitude angle β for every direction and magnitude of the load. These values are determined iteratively using formulation for Sommerfeld Number. More details on this approach can be found in selected literature [21] and [23].

Friction in valve train

Mechanical losses in valve train were predicted by means of simplified time averaged model presented by Sandoval et al. [4]. The model incorporates exclusively those terms relevant for design of valve train system of engine used in experiment. The model provides FMEP, which can be effortlessly converted in torque and power. The model is presented as a single line equation:

$$FMEP_{vts} \approx C_{ff} \cdot \left(1 + \frac{500}{n}\right) \cdot \frac{i_v}{s \cdot i_c} + C_{oh} \cdot \sqrt{\frac{\eta(T_l)}{\eta(T_{l,o})}} \cdot \left(\frac{h_{v,\max}{}^{1.5} \cdot n^{0.5} i_v}{D \cdot s \cdot i_c}\right) + C_b \cdot \sqrt{\frac{\eta(T_l)}{\eta(T_{l,o})}} \cdot \frac{n \cdot i_b}{D^2 \cdot s \cdot i_c} \quad (19)$$

Where iv is a number of valves, i_c number of cylinders, i_b number of bearings s and D piston stroke and bore respectively, and $h_{v,max}$ valve lift. The first term includes the effect of mixed lubrication for flat follower and the second one evaluates hydrodynamic friction in tappet–liner and valve–guide contacts. The last term evaluates friction in camshaft bearings. Other terms related to cam rollers and seals are neglected. Constants C_{ff}, C_{oh} and C_b can be found in literature [4].

Mechanical losses in engine auxiliaries

Energy used to provide coolant circulation depends on coolant pressure, coolant flow rate and hydraulic resistance in radiator and passages. The necessary flow rate depends on engine operation point, however can be estimated based on circulation pump outlet pressure, its speed and efficiency. The outlet pressure is proportional to the square of the flow rate, which is also proportional to the speed of the pump. The power is approximated as follows:

$$P_{pc} \approx n_{pc}{}^3 = (k_{pc} \cdot n)^3, \quad (20)$$

and, assuming that coolant circulation pump absorbs approximately 1% of engine nominal power Pe,max, relation can be recomposed as to incorporate ratio of instant engine speed to its nominal value nmax:

$$P_{pc}(n) = P_{pc,\max} \cdot \left(\frac{n}{n_{\max}}\right)^3 = 0.01 \cdot P_{e,\max} \cdot \left(\frac{n}{n_{\max}}\right)^3, \quad (21)$$

Engine lubrication system absorbs mechanical power from crankshaft which can be for common gear type pump determined from lubrication system pressure pl and known pump design parameters namely, nominal diameter D_{po}, teeth height h_t, gear height h_p, pump to engine speed ratio k_{pl} and pump volumetric and mechanical efficiency $\eta_{pl,v}$ and $\eta_{pl,v}$, respectively:

$$P_{pl}(n, p_L) = \frac{Q_l(n) \cdot p_l}{\eta_{pl,m}} = D_{po} \cdot h_t \cdot h_p \cdot k_{pl} \cdot \frac{\eta_{pl,v}}{\eta_{pl,m}}, \quad (22)$$

Assuming equal values for both volumetric and mechanical efficiencies $\eta_{pl,v}$ and $\eta_{pl,v}$, previous expression becomes dependant on known parameters, retaining uncertainty of less than 0.2%.

Experimental verification

In this work, simplified 1-DoF dynamic engine model is applied to simulate crankshaft instantaneous angular speed. Engine-dynamometer dynamic system is presented in Figure 11. Engine mass moment of inertia J_E is assumed as a function of both mass and position of slider mechanism components in respect to shaft angle position. Engine inertia and its first derivative in respect of crank angle are calculated by means of dynamically equivalent model, while inertia of flywheel J_{FW}, connecting shaft J_S and dynamometer J_D are known constants obtained from manufacturing specification. Assuming crank and connecting shafts rigid [1][3], torque balance equation for engine-dynamometer system arises from kinetic energy equation (Newton's principle):

$$\left[J_E(\alpha)+J_{FW}+JS+J_D\right]\cdot\ddot{\alpha}(\alpha)+\frac{1}{2}\cdot\frac{dJ_E(\alpha)}{d\alpha}\cdot\dot{\alpha}(\alpha)^2=T_G(\alpha)-T_F(\alpha)-T_L\,, \qquad (23)$$

Gas-pressure torque contributions from individual cylinders are denoted by term T_G while T_L is the measured load torque. The term T_F denotes the sum of torques from friction and mechanical losses in engine moving components and auxiliaries. In this analysis, in-cylinder pressure measured in 2nd cylinder, which is regarded as a master cylinder, was copied and phase shifted as to provide gas pressure torque T_G for an engine as a system. The term T_F was predicted by means of hybrid model (combination of angle-based and time-averaged components) presented in this work.

The instantaneous crankshaft angular speed was measured at power take-off (PTO) by means of flywheel gear as an incremental disc. The experimental results are presented in Figure 12 for three loads, BMEP 8.2 10^5 Pa, 6.04 10^5Pa and 3.16 10^5Pa at engine speed of approx. n=2300 min^{-1}. The measured signal was filtered, averaged on an ensemble of 50 consecutive cycles, and smoothed using cubic approximation spline in order to retain smooth second order derivative (dash-dot line, Figure 12). The signal was corrected for flywheel radial run-out (dashed line, Figure 12). The simulation results were presented in solid line.

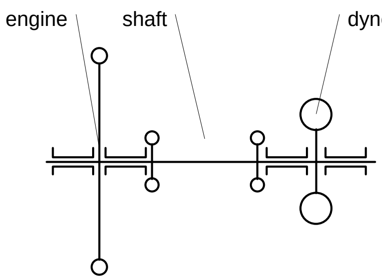

Figure 11 Reduced, equivalent 1-DoF Engine dynamic model

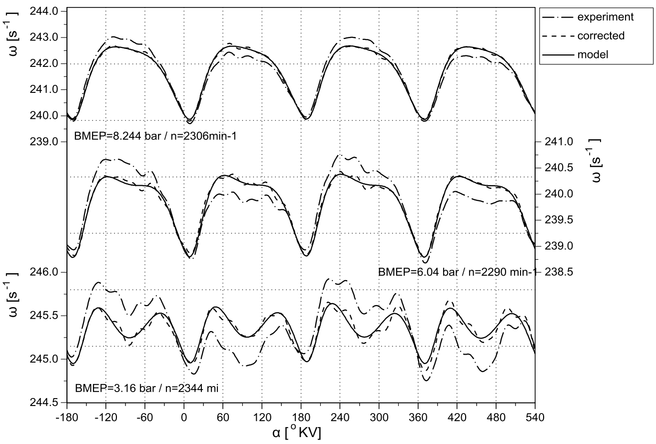

Figure 12 Instantaneous crankshaft angular speed n=2300 min^{-1}

The model for simulation of friction losses in PRAC and PSC contacts provided solid values for prediction of instantaneous crankshaft angular speed. The influence of mass inertia torques dominates over gas pressure torques at the low BMEP operating point. Here, uncertainties in engine friction and dynamic model become significant. The results improve at higher loads where gas pressure torque increases. Although the basic approach in friction modelling presented in this work is analytical and based on fundamental theories, it is still strongly affected by a number of design and empirical constants which must be carefully identified. In spite of that the simulation was performed using model set up based on numerical values accessible from literature, and having in mind simplifications implemented in model of engine crank slider mechanism dynamics and friction losses, the results are correct, follow the trends and provide solid prediction of engine dynamic behaviour in angle based presentations. Based on analysis in previous section, model can provide valuable information for process and design optimisation.

CONCLUSIONS

Simplified, angle-based approach to modelling of friction in Piston Ring Assembly – Cylinder contact and Piston Skirt – Cylinder contact were presented. The models were used to analyse the influence of engine speed and load to friction phenomena inpiston – cylinder liner contact. The model provides generally good results in terms of global trends which were reported by other authors and based on both complex modelling and experimental verification.

The model was tested against instantaneous angular crankshaft speed and compared to values obtained experimentally. The model was supplemented by angle-based friction model in HD bearings, and time-averaged models for friction in valve train system and for power consumed by engine auxiliaries.

The model verification indicates solid prediction capabilities. Uncertainties in friction models exist, and empirical constants which cannot be avoided require careful identification. Procedure presented in this work, proved sufficiently robust and sensitive and provided improved in terms of friction coefficient identification could be used effectively for prediction of engine performance in dynamic operation.

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REFERENCES

[1] Chen, S. X., Moskwa, J. J.: “Application of Nonlinear Sliding-Mode Observers for Cylinder Pressure Reconstruction”, Control Engineering Practice. 5 (1997).

[2] Yagi, S et al.: “Estimate of Total Engine Loss and Engine Output in Four Stroke S.I. Engines”, SAE Technical Papers. (1991) SAE-910347.

[3] Filipi, Z. S., Assanis, D. N.: “A Nonlinear, Transient, Single-Cylinder Diesel Engine Simulation for Predictions of Instantaneous Engine Speed and Torque”, Journal of Engineering for Gas Turbines and Power. 123 (2001) 951-959.

[4] Sandoval, D., Heywood, J.: “An Improved Friction Model for Spark-Ignition Engines”, SAE Technical Papers. (2003) SAE 2003-01-0725.

[5] Arsie, I. et al.: “Development and Validation of a Model for Mechanical Efficiency in a Spark Ignition Engine”, SAE Technical Papers. (1999) SAE 1999-01-0905.

[6] Taylor, C. M.: “Engine Tribology”, Elsvier Science Publishers, 1993.

[7] Stanley, R. et al.: “A Simplified Friction Model of the Piston Ring Assembly”, SAE Technical Papers. (1999) SAE 1999-01-0974.

[8] Livanos, G. A., Kyrtatos, N. P.: “Friction model of a marine diesel engine piston assembly”, Tribology International. 40 (2007) 1441-1453.

[9] Livanos, G. A., Kyrtatos, N. P.: “Friction model of a marine diesel engine piston assembly”, Tribology International. 40 (2007) 1441-1453.

[10] Ciulli, E., Rizzoni, G., Dawson, J.: “Numerical and Experimental Study of Friction on a Single Cylinder CFR Engine”, SAE Technical Papers. (1996) SAE 960357.

[11] Guzzomi, A. L. Hesterman, D. C. Stone, B. J.: “The effect of piston friction on engine block dynamics”, Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-body Dynamics. 221 (2007) 277-289.

[12] Guzzomi, A. L. Hesterman, D. C. Stone, B. J.: “Variable inertia effects of an engine including piston friction and a crank or gudgeon pin offset”, Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering. 222 (2008) 397-414.

[13] Taraza, D., Henein, N., Bryzik, W.: “Friction Losses in Multi-Cylinder Diesel Engines”, SAE Technical Papers. SAE-2000-0 (2000) SAE 2000-01-0921.

[14] Taraza, D., Henein, N., Ceausu, R., Bryzik, W.: “Complex Diesel Engine Simulation with Focus on Transient Operation”, Energy & Fuels. 22 (2008) 1411-1417.

[15] Zweiri, Z. H., Whidborne, J. F., Seneviratne, L. D.: “Detailed analytical model of a single-cylinder diesel engine in the crank angle domain”, Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering. 215 (2001) 1197-1216.

[16] Zweiri, Z. H., Whidborne, J. F., Seneviratne, L. D.: “Instantaneous friction components model for transient engine operation”, Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering. 214 (2000) 809-824.

[17] Rakopoulos, C., et al.: “Application and Evaluation of a Detailed Friction Model on a DI Diesel Engine with Extremely High Peak Combustion Pressures”, SAE Technical Papers. (2002) SAE 2002-01-0068.

[18] Kouremenos, D. et al.: “Development of a Detailed Friction Model to Predict Mechanical Losses at Elevated Maximum Combustion Pressures”, SAE Technical Papers. (2001) SAE 2001-01-0333.

[19] Thring, R.: Engine Friction Modeling, SAE Technical Papers. (1992) SAE 920482.

[20] Lin, S.-S., Patterson, D. J.: “Piston-Ring Assembly Friction Modeling by Similarity Analysis”, SAE Technical Papers. (1993) SAE 930749.

[21] Popović, S. J.: “Research and Development of instantaneous crankshaft angular speed based method for engine combustion analysis”, PhD Thessis, University of Belgrade, 2013.

[22] Stolarski, T. A.: “Tribology in Machine Design”, Butterworth-Heinemann, Oxford, England, 2000.

[23] Stachowiak, G. W., Batchelor, A. W. : “Engineering Tribology”, Butterworth-Heinemann, Elsevier Inc., 2006.

[24] Wannatong, K.: “Simulation algorithm for piston ring dynamics”, Simulation Modelling Practice and Theory. 16 (2008) 127-146.

[25] Hafner, K. E., Maass, H.: “Theorie der Triebwerksschwingungen der Verbrennungskraftmaschine” Band 1-4, Springer-Verlag Wien/VEB Verlag Technik, Berlin, 1984.

[26] Klaus, E. E. , Tewksbury, E. J.: “Lubricants and Their Application”, CRC Handbook of Lubrication Theory and Practice of Tribology, Volume II: Theory and Design,, CRC Press, 1988.

[27] Seeton, C. J.: “Viscosity–temperature correlation for liquids, Tribology Letters. 22 (2006) 67-78.

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MEASUREMENT THE FUEL CONSUMPTION OF BUSES FOR PUBLIC TRANSPORT BY THE METHODOLOGY ''SORT'' (STANDARDISED ON-ROAD TESTS CYCLES)

Slobodan Mišanović[1]*, Vladimir Spasojević*

UDC:629.1.07

ABSTRACT: The consumption of fuel is important indicator and aspect in operation of buses for public transportation. During the tender procedure for the procurement of new buses, fuel consumption is presenting one of the evaluation criteria in order to optimize the operating costs of the vehicle during its life cycle. Since 2009, in the EU as official methodology for measuring the fuel consumption of buses for urban - suburban transport, is used SORT methodology. The methodology is a simulation of real traffic conditions: acceleration, deceleration, stopping, vehicle load, measuring the amount of fuel consumed and processing results on the specially prepared test site for city, intercity and combined cycle. This paper shows the methodology SORT 1 (urban cycle) and the results of tests on five buses of GSP "Beograd", brand "Solaris Urbino 18", which were conducted in February and March of 2014, on the proving ground "Santa Oliva" (Spain) by certification body "Applus Idiada". Also, will be presented a comparative view of fuel consumption for buses "Solaris Urbino 18" in real operating conditions on the routes are served by GSP "Beograd".

KEY WORDS: fuel consumption, bus, urban cycle, public transport

MERENJE POTROŠNJE AUTOBUSA JAVNOG PREVOZA METODOLOGIJOM „SORT“ (STANDARDOZOVANI TEST CIKLUSI NA PUTU)

REZIME: Potrošnja goriva je važan indikator i aspekt u radu autobusa javnog prevoza. Tokom tendera nabavke novih autobusa, potrošnja goriva predstavlja jedan od kriterijuma u cilju optimizacije operativnih troškova vozila tokom njegovog životnog ciklusa. Od 2009 godine, u Evropskoj Uniji se kao zvanična metodologija za merenje potrošnje goriva u autobusima gradskog-prigradskog prevoza koristi SORT metodologija. Metodologija je simulacija realnih uslova u saobraćaju: ubrzavanje, usporavanje, zaustavljanje, opterećenje vozila, merenje utrošenog goriva i procesiranje rezultata na posebno pripremljenom poligonu za gradsku, prigradsku i kombinovanu vožnju. Ovaj rad pokazuje SORT 1 metodologiju (urbani ciklus) i rezultate testova na pet autobusa GSP “Beograd”, proizvođača autobusa “Solaris Urbino 18”, koji su izvršeni u februaru i martu 2014. godine, na poligonu “Santa Oliva” (Španija) od strane sertifikacionog tela “Applus Idiada”. Takođe, biće predstavljen uporedni prikaz potrošnje goriva za autobuse “Solaris Urbino 18” u realnim uslovima rada na trasama koje opslužuju autobusi GSP “Beograd”.

KLJUČNE REČI: potrošnja goriva, autobus, urbani ciklus, javni prevoz

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INTRODUCTION

The fuel consumption is presenting an important aspect of operation the buses for public transportation specifically in terms of optimizing the total cost of the life cycle. Also in the last time it is one of the mandatory criteria in tenders for the purchase of new buses. Until 2009, was not standardized methodology that is related to this issue, so the manufacturers of buses declared consumption that were based on individual factory tests.

Also many bus manufacturers declared that the factory consumption based on testing performed on engine test stands. However, nor the ESC cycle (European Steady Cycle) prescribing 13 fixed points of measuring nor the ETC cycle (European Transient Cycle), with more dynamic characteristics, take into account the whole vehicle. Furthermore these engines test cycles in no way adequately reflect the stop-and-go operation of a scheduled service bus.

For these reasons, classic normative tests are not sufficient to simulate the operation of a public transport vehicle. Therefore it seemed indispensable to design on-road test cycles for the whole vehicle in the framework of the SORT project, based on statistically generated data from several European transport companies (commercial speed, average time spent at stops, average distance between them, the load, etc.).

The goal of SORT methodology is standardized cycle test for bus, which will measure the fuel consumption and be applied to all bus manufacturers and operators in the public transportation system.

INFLUENTIAL FACTORS ON FUEL CONSUMPTION OF BUSES FOR PUBLIC TRANSPORT

In order to define the most important influencing factors(Figure 1), it is necessary to take into account a number of typical parameters such as traffic density, numbers of stops (either for the boarding and alighting of passengers or simply required by the environment), route topography, vehicle loads, and commercial speed. Although integrating so many variables is a difficult process, these parameters can all be seen to have a direct influence on commercial speed, which therefore becomes a kind of common denominator for the different variables [1].

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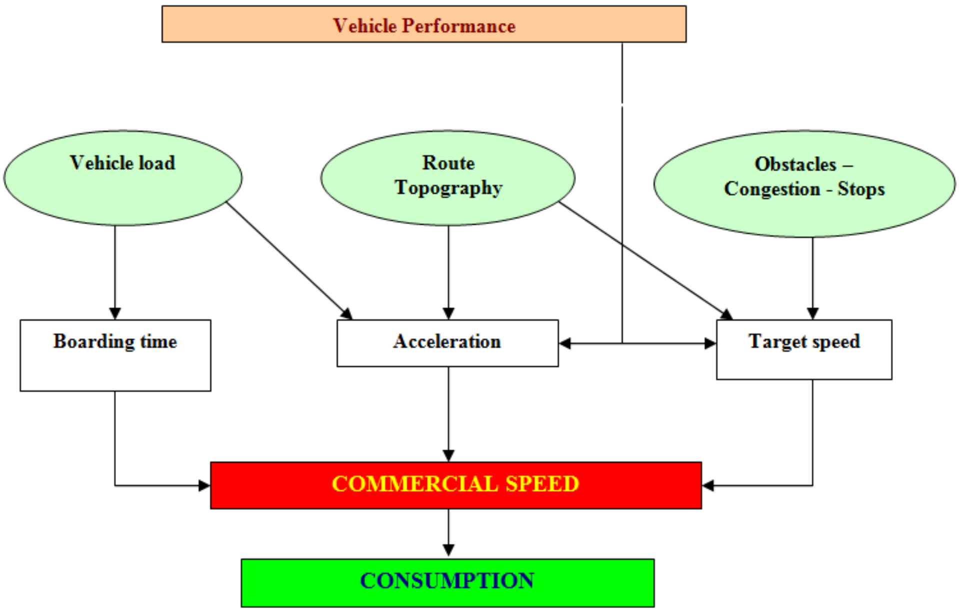


Figure1 Influential factors on fuel consumption

Commercial speed can be seen as the key parameter differentiating distinct operation patterns. Indeed, the graph above shows that any change of route severity impacts on commercial speed and thus on consumption (which is inversely proportional). Consumption reduction following increased commercial speed (Figure 2) is a paradox well-known to operators: commercial speed can only be influenced by structural measures such as dedicated bus lanes, and consequently a reduction of congestion-related stops, which has a favourable impact on consumption [2].

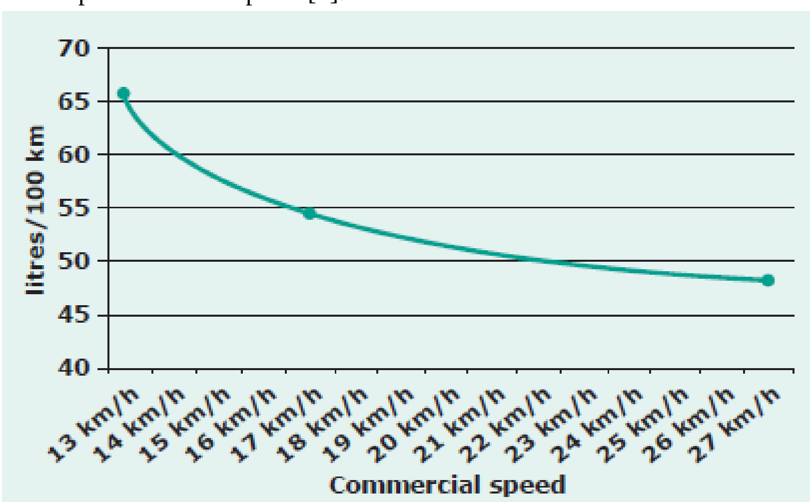


Figure 2 The consumption of an articulated bus according to its commercial speed

CYCLE DESIGN

Each company would like to have a cycle that reflects its operation pattern. This is clearly not feasible. As many cycles as bus routes would be needed. Moreover, in some cities, the same vehicles run on different routes, with their own features. Out of this situation, the basic SORT philosophy was born: designing a given number of cycles in such a way that companies can assess their operation as a combination of several base cycles. The basic cycle (Figure 3): SORT 1 (urban cycle), SORT 2 (combined urban-suburban) and SORT 3 (suburban cycle).

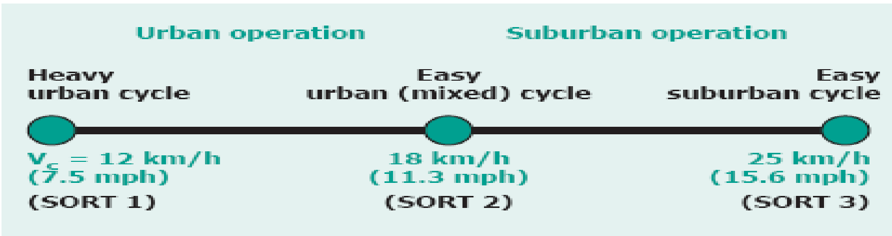
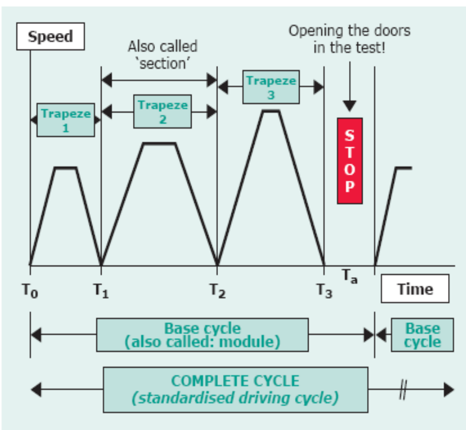


Figure 3 SORT base cycles

A proposed cycle is thus made up of the repetition of the identical base cycle featured by an average speed and a length, and driven with a simulated load . The need for result confidence could require increasing the number of repetitions in a complete cycle until the overall results lie within a given standard deviation. The sequence of trapezes (sections) in one module is intended to reflect the driving conditions of a public transport vehicle: frequent stops (either with opening of doors for boarding/alighting, or stops due to traffic conditions (traffic lights, congestion) [3]. Total idle time within a trapeze will be defined, as well as total duration of the base cycle module, in order to reach a pre-established mean speed, which will be dealt with later. These stops punctuate the trapezes, each of which defined is by an acceleration, a constant speed stretch (section target speed) and a braking phase (Figure 4).



	SORT 1	SORT 2	SORT 3
Rated average speed (km/h)	12.1	18	25.3
Stops/km	5.8	3.3	2.1
Stop time (%)	39.7	33.4	20.1
Trapeze 1 v-const. (km/h)/ length (m)	20/100	20/100	30/200
Acceleration (m/s²)	1.03	1.03	0.77
Trapeze 2 v-const. (km/h)/ length (m)	30/200	40/220	50/600
Acceleration (m/s²)	0.77	0.62	0.57
Trapeze 3 v-const. (km/h)/ length (m)	40/220	50/600	60/650
Acceleration (m/s²)	0.62	0.57	0.46
Length of stops (s)	20/20/20	20/20/20	20/10/10
Total length (m)	520	920	1,450
Deceleration (m/s²)	0.8	0.8	0.8

Figure 4 Structure of the bases cycles

For example SORT 1, city cycle consists of three trapezoids with a constant target speed of 20 km/h, 30km/h, 40 km/h. After each trapezoid provided a retention time of

20seconds, so this cycle is characterized by the total time idling than 60 seconds. The average speed of exploitation of this test is about 12 km/h. Length of the test track is 520 m (Figure 5). For each variety, the cycle is defined by a lump load of 3,200 kg for the solo bus, and 4,978 kg for the articulated bus [1].

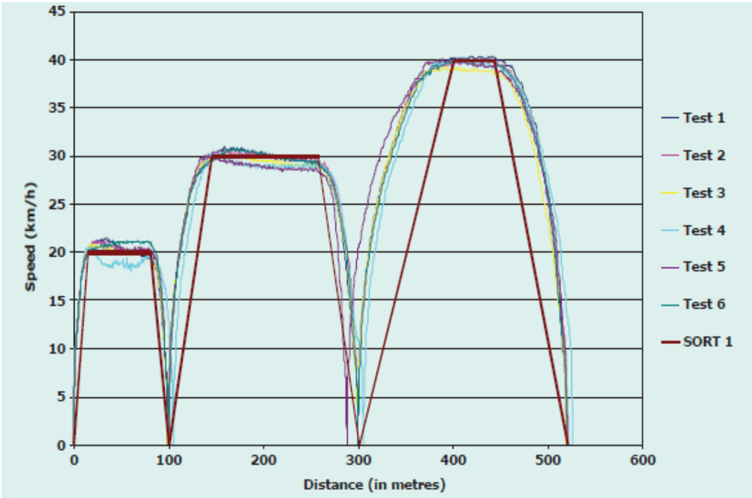


Figure 5 A good example of the conducted test SORT 1

MEASURING METHOD

Measuring instruments

In order to achieve comparable and reproducible measuring results, the instruments must fulfil the minimum requirements listed below (Table 1):

Table 1 Measuring instruments-minimum requirements

Meas. Instrument	Accuracy
Fuel-flow meter or a Gravimetric fuel meter	+ / - 2%
Speed	+ / - 0.5%
Distance measuring device (for positioning of traffic cones)	+ / - 0.2%

The measuring instruments calibration and maintenance must comply with the requirements as under DIN EN ISO 9000.

MEASURED VARIABLES

Speed

Vehicle speed is recorded during the measuring run. Here, the recording serves to monitor adherence to the trapezoidal speed. The correctness of the measurement must be checked during the breaks between two measurements. In the event of any deviations the measurement is to be repeated.

Time

The time required for the SORT cycles (including each stop period) is documented. The time and the resulting average speed are entered into the report.

Fuel consumption

The fuel consumption of the individual measuring runs is documented. For volumetric measurement of the fuel consumption, the measured values have to be corrected for a temperature of 20 °C.

Fuel temperature

The fuel temperature is measured (for volumetric measurement) at the start and end of each cycle in the area of the measuring chamber. The mean value is used for converting the fuel consumption to standard conditions

Fuel density

The density of the fuel used is measured at a temperature of 20 °C and entered into the report.

MEASURING PROCESS

The following procedure is recommended for the measuring process: [1]

- Measure out route points (1, 2, 3, 4) for the individual SORT cycles and mark them using traffic cones
  - The SORT 1 cycle, which is only 520 m, must be run through at least twice in succession (= 1040 m)
  - The SORT 2 (920 m) and SORT 3 (1450 m) cycles need to be run through, at least once
  - Should the cycle be run through more than once, without pause between two runs, the results of the overall distance shall be taken into account.
- The consumption measurements for each respective cycle are to be repeated until 3 measurements lie within an accuracy requirement of 2%. To calculate the accuracy, the difference between the maximum and minimum consumption value of the three measurements is divided by the minimum value ((Cmax-Cmin)/Cmin x 100). (Figure 6)
- The tolerance for the trapezoidal target speed is ± 1 km/h. During the transition from acceleration to steady-state driving, a maximum deviation of +3 km/h is permitted for a brief period.
- No more than 10 minutes should elapse between each measurement

SORT 1 / Direction North				
Measured values (X)	Values ordered by increasing order	Numbers of the considered values	Gap	Mean value
1 49.3	1 49.3	1, 7, 11	3.5	
3 52.0	7 50.6	7, 11, 9	2.3	
5 53.0	11 51.1	**11, 9, 3**	**1.7**	**51.63**
7 50.6	9 51.8	9, 3, 5	2.3	
9 51.8	3 52.0			
11 51.1	5 53.0			

SORT 1 / Direction South				
Measured values (X)	Values ordered by increasing order	Numbers of the considered values	Gap	Mean value
2 51.8	6 50.3	6, 12, 4	1.9	50.70
4 51.3	12 50.5	12, 4, 2	2.5	
6 50.3	4 51.3	4, 2, 8	1.5	
8 52.1	2 51.8	2, 8, 10	0.8	
10 52.2	8 52.1			
12 50.5	10 52.2			

Value SORT 1 = (51.63 + 50.70) / 2 = 51.17 litres

SORT 1 / Direction North				
Measured values (X)	Values ordered by increasing order	Numbers of the considered values	Gap	Mean value
1 49.3	1 49.3	1, 7, 11	3.5	
3 52.0	7 50.6	7, 11, 9	2.3	
5 53.0	11 51.1	**11, 9, 3**	**1.7**	**51.63**
7 50.6	9 51.8	9, 3, 5	2.3	
9 51.8	3 52.0			
11 51.1	5 53.0			

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8 52.1	2 51.8	2, 8, 10	0.8	
10 52.2	8 52.1			
12 50.5	10 52.2			

Value SORT 1 = (51.63 + 50.70) / 2 = 51.17 litres

Figure 6 An example of a calculation of the consumption SORT 1[4]

PRACTICAL EXPERIENCE ON THE CONDUCTED TEST SORT 1

In October of 2012. "Bus manufacturer Solaris Bus & Coach SA" won on the tender for the purchase 200 articulated buses, which was announced by the City of Belgrade. One of the criteria for selecting was fuel consumption. Articulated bus Solaris Urbino 18, was declared an average fuel consumption of 56 l/100 km in city mode of exploitation according to the methodology SORT 1. According to the obligation of contracts "Solaris Bus & Coach SA" after delivery, was obliged to carry out a control test fuel according to the methodology SORT 1 and prove that the declared fuel consumption , which was on the tender prevail. Delivery of 200 articulated buses were conducted in the period August-November 2013, within five contingents. For each contingent was determined a bus, on which will be measured the fuel consumption by the methodology SORT 1. The difference in fuel consumption on the control test, in relation to the declared value of the tender can be up to 5 % . In the case of consumption more than 5 % of declared " Solaris Bus & Coach SA" is required to pay penalties amounting to 2 % of the value of the each contingent [6].

Buses "Solaris Urbino 18", which have been tested: Vehicle No. 3002 , registration number: BG 728 –GC; Vehicle No. 3030, registration number: BG 753- WJ; Vehicle No. 3099, registration number: BG - 762 PZ; Vehicle No. 3110, registration number: BG 737 – HS; Vehicle number 3166, registration number: BG - 753 YC.

Tehnical data:

Length	18.000 mm
Width	2.550 mm
Height	3.050 mm
No. of passengers	159
No. of seats	40
Engine	DAF PR 228 (Euro 5)
Volume (cm3)	9.186
Power (KW) min-1	231/1900
Max torque (Nm) min-1	1275 /710-1.100
Gear	ZF-6AP
Front axle	ZF RL 75 EC
Drive axle	ZF AV 132
Centre axle	ZF AVN 132
Fuel tank	350 L
Ad blue tank	40 L

Figure 7Solaris Urbino 18, articulated city bus

Testing according methodology SORT 1 (urban cycle), was performed by accredited certification company

"APPLUS+IDIADA" Barcelona on specially prepared polygon "Santa Oliva" in the period from February to April of 2014. Test ground on which testing was done, has specially prepared, length of track 2x520 meters, which simulates the exploitation of buses in typical urban conditions, which means little exploitation speed (12-13 km/h) , frequent acceleration to 20, 30, 40 km/h, deceleration, braking, time on stop.

Before the test, vehicle was loaded with sacks of sand and 3 vessel with water to a simulated load of 50 % of the number of passengers in the vehicle.

Table 2 Results of measurements of fuel consumption, according methodology SORT 1, on the polygon'' Santa Oliva'' [7]

Garage number	Consumption on the test (L/100Km)	Declared consum. on the tender (L/100Km)	Difference (L/100Km)	%	Period	Mileage before test (Km)	Wind speed (m/s)	Temperature (C)
3002	57,2	56	1,2	2,14	18.02.2014	56193	0,6-1,0	12,1
3030	56,5	56	0,5	0,89	05.03.2014	46456	0,4-1,9	10,5
3110	55,30	56	-0,7	1,25	11.03.2014	42287	0,2-1,4	11,4
3099	58,2	56	2,2	3,93	31.03.2014	34819	1,4-1,5	13,1
3166	58,7	56	2,7	4,82	02.04.2014	32004	1,2-3,0	17,1

Based on the results of measurements of fuel consumption per Methodology SORT 1 (Table 1), it can be concluded that for every vehicle that was the subject of testing, were within the permissible limits of up to 5% compared to the stated consumption, which was at the tender (56 L/100 km). The average fuel consumption measured for all five vehicles that were tested by the methodology SORT1 is 57.18 l/100 km, which is 2.11% higher than the consumption on the tender.

The following table 3, presents a comparative view of the results of fuel consumption with a polygon test, results and consumption in real operating conditions, where vehicles operate for a period 1.3-15.4.2014.

Table 3 Comparison of fuel consumption on the test and real conditions of exploitation[5]

Garage number	Consumption on the test (L/100Km)	Consumption in real exploitation (L/100Km)	Difference (L/100Km)	%	Line
3002	57,2	52,12	-5,08	-8,88	17
3030	56,5	52,11	-4,39	-7,77	88
3110	55,30	57,66	2,36	4,27	23
3099	58,2	51,86	-6,34	-10,89	88
3166	58,7	55,85	-2,85	-4,86	65

Table 4 Average fuel consumption for the Solaris Urbino 18 (all vehicles) per month [5]

Month	Average fuel consumption, (L/100 Km)
September 2013	59,39
October 2013	56,51
November 2013	54,45
December 2013	54,62
January 2014	55,41
February 2014	54,62
March 2014	54,45
April 2014	53,88
May 2014	55,75
Average	55,45

From table 4, conclude that the average fuel consumption for a group of vehicles Solaris Urbino 18 in the period September 2013 to May of 2014. amount of 55.45 l/100 km.

CONCLUSIONS

Presented SORT methodology is an important tool in the preparation of tenders for the purchase of new buses and the choice of optimal drive configuration in terms of fuel economy. Results of the test of fuel consumption by SORT 1 (57.18 l/100 km), declared of consumption on the tender (56 L/100 km) and the consumption in the real operations (55.45 l/100 km) are very close value with small deviations .

The selected configuration of the engine and transmission on vehicles Solaris Urbino 18, fully meets the high weight factors of using the buses in Belgrade and demand for economic fuel consumption.

REFERENCES

[1] "UITP Project SORT- Standardised On-Road Tests cycles",

[2] (secend edition),UITP,D/2009/0105/31,Brussels,2009

[3] "Conditions de fonctionnement des autobus et cycles de conduite",RATP-Departement Environnement, Paris (1999)

[4] Dubrowski,D.:"Trapezzyklus fur Stadtbus’’,’’Streckenverbrauch eines Sdtlinienbusses", letter to the UITP Secretariat (August 1999)

[5] Misanovic,S.:"Statistical method of comparasion results of fuel consumption",UITP–Bus Committee,86th meeting,Praha, 1-3 April 2009

[6] Misanovic,S.:"Norms of fuel consumption along the lines for group buses Solaris Urbino 18" in GSP"Beograd", Belgrade, April 2014

[7] "Tender documentation for purchase 200 buses Solaris Urbino 18", GSP"Beograd", October 2012

[8] "Reports of Fuel consumption test according to the SORT 2009 procedure for 5 buses Solaris Urbino 18" of GSP ’’Beograd’’, Applus IDIADA, Project No.LM1402001,Santa Oliva (Spain), February-April 2014

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