ISSN: 1231-4005
e-ISSN: 2354-0133
DOI: 10.5604/12314005.1213597

THE ANALYSIS OF FRICTION IN THE BEARING OF ROCKER ARMS

Przemysław Kubiak, Krzysztof Siczek

Lodz University of Technology
Department of Vehicles and Fundamentals of Machine Design
Zeromskiego Street 116, 90-924 Lodz, Poland
tel.: +48 42 6312248, +48 42 6312250, fax: +48 42 6312255
e-mail: przemyslaw.kubiak@p.lodz.pl, ks670907@p.lodz.pl

Krystian Siczek

Lodz University of Technology,
Faculty of Electrical, Electronic, Computer and Control Engineering
Stefanowskiego Street 18/22, 90-924 Lodz, Poland
tel.: +48 42 6312250, fax: +48 42 6312255
e-mail:krystian.siczek@p.lodz.pl

Abstract

Rocker arms are important parts in many valve train systems of combustion engines. They are responsible for translating the profile of the camshaft into motion for opening and closing the inlet and outlet valves. The design, materials and lash adjuster elements for existing rocker arms were discussed. The modified rocker arm bearing was designed for the investigated Benzer 50cc four-stroke engine. The modification for the analysed engine includes the change of bearing for both inlet and outlet rocker arms. The original rocker arm bearing is of the slide type and mates with the axis fixed via the pin to the cylinder head. The modified bearing can be of the needle roller or the slide type and include two bearings mating with the axis, to which the rocker arm is fixed. The aim of the presented study was to obtain the courses of pressure in cylinder against the crankshaft angle, which was needed to calculate the loading force of rocker arm bearing and then friction torque therein. The models of valve train assembly and of the bearing operating under boundary friction were elaborated. Resulted values of the friction torque in rocker arm bearing for different operating conditions of engine were presented in the paper.

Keywords: *rocker arm, slide bearing, combustion engine, friction*

1. Introduction

Rocker arms are the important parts in the valve train system of combustion engine. The function, operation, and design of rocker arms and cam followers are presented in [12]. A rocker arm is used in the operation of an engine because it is responsible for translating the profile of the camshaft into motion for opening and closing the inlet and outlet valves [13].

Valve train stability depends on the rocker system design [14]. Failure of rocker arms is discussed in [1, 3, 4, 6, 7, 17, 21] and tribological properties of the roller follower – camshaft set in [5, 8, 22].

For the planned building of the tester for wear the valve train components, the Benzer 50cc four-stroke engine will be adapted. It has the max power of 1.17 kW at 6070 rpm and the max torque of 1.76 Nm at 2030 rpm, as it was measured in Department of Vehicles and Fundamentals of Machine Design at Lodz University of Technology. The compression ratio is equal 9:1. The valve train of such engine is presented in the Fig. 1 [23]. It contains one inlet valve with head diameter equal 18.5 mm and one outlet valve with head diameter equal 16 mm. The planned modification for the analysed engine includes the change of bearing for both inlet and outlet rocker arms. The original rocker arm bearing is of the slide type and mates with the axis 1 fixed via the pin 4 to the cylinder head 2. The modified bearing can be needle roller 5 or sliding type and include two bearings mating with the axis, to which the rocker arm 3 is fixed via the pin 4. The aim of the presented study was to obtain the characteristics $p(CA)$, which was needed to calculate the loading force F of rocker arm bearing and then friction torque M_T therein.

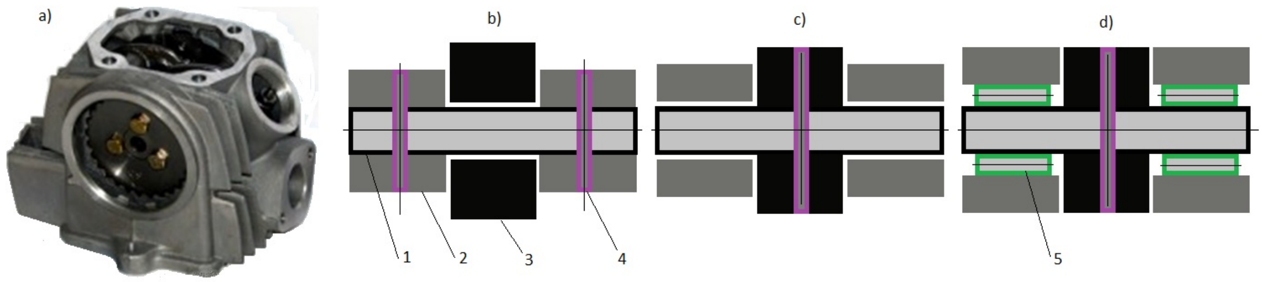

Fig. 1. a) valve train of Benzer 50cc four-stroke engine [23], b) scheme for the original rocker arm slide bearing, c) scheme for the modified rocker arm slide bearing, d) scheme for the modified rocker arm needle roller bearing; 1 – axis, 2 – cylinder head, 3 – rocker arm, 4 – fixing pin, 5 – needle roller. Source for a), b) and c) – own investigations

2. Types of Rocker Arms

Rocker arms are placed typically between the pushrod and the inlet and outlet valves. They are pushed up by the mating pushrods and therefore they push down on the valves tips. In case of Over Head Cam applications, the cams ride directly on the Rocker Arms.

In addition to just changing the direction of the motion from up on the pushrod side to down on the valve side, the Rocker Arm changes the amount of motion transferred. Typically, a Rocker Arm will "multiply" its motion by a Rocker Arm Ratio by a factor of 1.45 to 1.7, meaning that for each .100" of pushrod motion you would get .145" to .170" of valve motion.

In the Push Rod Engines, four types of rocker arms are used. They are namely: Stamped Steel Rocker Arms, Roller Tipped Rocker Arms, Full Roller Rocker Arms and Shaft Rocker Arms [23].

The Stamped Steel Rocker Arms are probably the most commonly produced. They are the easiest and cheapest to manufacture, as they are stamped from one piece of metal. They use a trunnion pivot that holds the rocker in position with a nut that has a rounded bottom. This is a very simple way of holding the rocker in place while allowing it to pivot up and down.

The Roller Tipped Rocker Arms are similar to the Stamped Steel Rocker and add a roller on the tip of the valve end of the rocker arm. This allows the less friction, for somewhat more power, and reduced wear on the valve tip. The Roller Tipped Rocker Arms still use the trunnion pivot nut and stud for simplicity. They can also be cast or machined steel or aluminium.

The Full Roller Rocker Arms are either machined steel or aluminium. They replace the trunnion pivot with bearings. They still use the stud from the trunnion pivot but they do not use the nut. They have a very short shaft with bearings on each end (inside the rocker), the shaft is bolted securely in place, and the bearings allow the rocker to pivot.

The Shaft Rocker Arms build off the Full Roller Rocker Arms. They have a shaft that goes through the rocker arms. Sometimes the shaft only goes through 2 rocker arms and sometimes the shaft will go through all of the rocker arms depending on how the head was manufactured. The reason for using a shaft is for rigidity. Putting a shaft through the rocker arms is much more rigid than just using a stud from the head. The more rigid the valve train, the less the valve train deflection and the less chance for uncontrolled valve train motion at higher RPM.

In the Over Head Cam Engines, there are two types of rocker arms used. They are namely: Center Pivot Rocker Arms and End Pivot (Finger Follower) Rocker Arms.

The Center Pivot Rocker Arms look like traditional rocker arms but they differ. Instead of the pushrod pushing up on the lifter, the Cam Shaft is moved into the head and the Cam Shaft pushes directly up on the lifter to force the valve down. The pivot point is in the centre of the rocker arm and the Cam Shaft is on one end of the rocker arm instead of the pushrod.

The End Pivots or Finger Followers put the pivot points at the ends of the Rocker Arms. In order for the Cam Shaft to push down on the Rocker Arm, such camshaft has to be located in the middle of the rocker arm.

3. Materials of Rocker Arms and Valve Lash Adjustment Elements

According to [3] the most common rocker materials are steel and aluminium.

The chrome-moly steel, although heavier than other materials, can offer some design advantages and have much thinner sections than aluminium due to its superior strength density. It takes at least two times the aluminium to approach the strength of steel. The moment of inertia, or performance mass, of properly engineered steel parts can actually be close to that of aluminium.

A. Steel is used in many automotive applications, as this material can provide a balance between weight and durability. Stamped steel was the OEM standard for Gen I and II, while cast steel was and is the standard for Gen III and IV. While these are suitable for OEM and basic performance, the aftermarket and racing groups can used also other options [21].
B. Anodized-aluminium roller rockers possess high performance.
C. High-strength alloy aluminium rockers arms are good, lightweight performers. Basic aluminium rocker arms are available with cast-alloy or extruded bodies, and high-end aluminium rocker arms are available machined from billet alloys [1].
D. Chrome-moly steel is a common material for high-performance rocker arms. The strength and rigidity of this material is hard to beat.
E. High-strength alloy steels are used in high-end, precision rocker arms, with rock-like rigidity for high-rpm race applications.

It was reported in [24], that High Density Polyethylene (HDPE) composite rocker arm possesses the lightweight, high strength and good frictional characteristics. A 3-D finite element analysis was carried out to find out the maximum stresses developed in the rocker arms made of steel and composite. From the results, it was noted that almost same stresses are developed for both the materials (steel and the composite). It was concluded that the stresses developed in the composite is well within the limits without failure. Such composite may be an alternate material for steel to be used as rocker arm.

The axis and the ball end of rocker arm can be coated by layer protecting mating surfaces against wear and for decrease the friction in the contact. The example can be the nc-WC/a-C:H coatings [10], elaborated by Makowka Marcin in Lodz University of Technology.

According to [9] in the case of the rocker arm, the hydraulic plug-in element can be used. Damage to the rocker arm most frequently occurs in the following: camshaft, contact between rocker arm and plug-in element and between rocker arm and valve, piston and housing of the hydraulic element, ball valve, and return spring. Failure can be caused by low oil quantity, oil foaming, impurities in the oil, or assembly error during installation.

4. Calculation of Friction Moment in Rocker Arm Bearing

Rocker arms can use both roller and slide bearing. The roller bearing is realized using the needle type of it, manufactured, i.e. by SKF.

According to [18] SKF provides integrated, high performance rocker arm bearings that generate less friction while providing more mileage and longer service life.

The friction torque in rocker arm needle bearing can be estimated from the equation (1) [25]:

$$M_T = \mu_m \cdot F \cdot R\,, \tag{1}$$

where:

M_T – friction torque in rocker arm bearing,

μ_m – friction coefficient in rocker arm bearing,
F – loading of rocker arm bearing,
R – radius of the shaft.

The friction coefficients μ_m of various needle bearings are presented in the Tab. 1 [11].

Tab. 1. Friction coefficients of various roller bearings [11]

Bearing classification	Friction coefficient μ_m
Needle roller and cage assembly bearings	0.0020 ~ 0.0030
Full-complement needle roller bearings	0.0040 ~ 0.0050
Thrust needle roller bearings	0.0030 ~ 0.0040

For the further analysis the friction coefficient value $\mu_m = 0.003$ has been assumed.

In the case of slide bearing applied in rocker arm, the obtaining of the friction moment is more difficult. The friction torque in slide bearing with full lubrication is given by the equation (2) [20]:

$$M_{Ti} = \pi\, d_i^2\, b_i\, \omega \eta_T \cdot \left(2 m_{Li} \sqrt{1 - \in_{Li}^2}\right)^{-1}, \quad i = 1, 2, ..., K, \tag{2}$$

where:
K – the number of camshaft bearings, ($K = 1$)
d_i – the diameter of the i-th bearing of the shaft, ($d_{i=1} = 0.007$ m)
b_i – the width of the i-th bearing, ($b_{i=1} = 0.02$ m)
m_{Li} – the relative clearance of the i-th bearing, ($0 < m_{Li=1} < 0.005$)
$\in_{Li}$ – the relative misalignment between shaft pivot and bushing of the i-th bearing,
ω – angular velocity of the shaft, ($0 < \omega < 250$ rad/s)
η_T – the dynamic viscosity of lubricating oil ($\eta_{T=293\mathrm{K}} = 0.38$ Pa·s, $\eta_{T=353\mathrm{K}} = 0.03$ Pa·s).

In multicylinder engines, the bearing load pulsation is small – a large number of cams reduce the amplitude of load changes. It can be assumed that $(\forall i) \in_{Li} = 0.7$ [20]. For simplicity of calculation, it can be also assumed in case of one-cylinder engine. Obtained values of friction torque M_T equal $1.6 \cdot 10^{-7}$ Nm for $T = 353$ K and $2.1 \cdot 10^{-6}$ Nm for $T = 293$ K seem underestimated.

According to [20] lubrication at the rocker arm pivot/shaft interfaces is mostly boundary lubrication. It is due to the fact, that there is very little lubricant supply to the surfaces. The boundary friction force at these interfaces is proportional to the constant friction coefficient and the applied contact load. Friction at the rocker arm pivots can reach 10% of the total friction in the valve train assembly at low engine speeds. Friction changes nonlinearly with the engine speed decrease (Fig. 2). Calculations carried out using data from [2]. When 5 camshaft bearings exist, the resulted values of friction torque M_T in the single bearing can be very high, up to 0.26 Nm.

To obtain loading of rocker arm bearing the engine power and torque were measured on the dyno VT-1, with the error of 0.1 %. In addition, pressure in the cylinder was measured using piezoelectric sensor for some engine speeds. Scaling the courses of such pressure to the values of measured torque and using CAD model based on geometry parameters (measured, from literature) of engine and its valve train, the forces loading axis of outlet rocker arm F_{outlet} and inlet rocker arm F_{inlet} were obtained. They allowed calculation of friction torque M_T with the error up to 10%.

For slide journal bearing of conformal geometry and lubricated with oil, the minimum film thickness h [m] is determined from equation (3) [19]:

$$h = c \cdot (1 - \varepsilon), \tag{3}$$

where:
c [m] – the clearance, that is, the difference between the radii of the bushing and the shaft,
ε [–] – the eccentricity ratio varying from 0 to 1.

The minimum film thickness h is a function (4) of load F, speed u, eccentricity ε [19]:

$$\Delta \cdot (d/b)^2 = \pi \cdot \varepsilon \cdot (1-\varepsilon^2)^{-2} \cdot (0.621 \cdot \varepsilon^2 + 1)^{0.5}, \tag{4}$$

where:
Δ [–] – the Sommerfeldt number, calculated from equation (5) [19]

$$\Delta = F \cdot (b \cdot u \cdot \eta_T)^{-1} \cdot (c/r_D)^2, \tag{5}$$

where:
$u = \omega \cdot r_D$ [m/s] – the relative speed between the surfaces.

According to [15] the dimensionless equation for the friction force F_f has the form (6):

$$\overline{F}_f = \mu_m \cdot \overline{\kappa}(\varepsilon) \cdot (\varepsilon - \varepsilon_{tr}) \cdot \operatorname{sgn}(\overline{u}) \cdot \Delta_1 + c \cdot r_D \cdot b^{-2} \cdot 2\pi \cdot (1-\varepsilon^2)^{-2} \cdot \overline{u}\,. \tag{6}$$

The dimensionless parameters [–] are following:

$$\overline{F}_f = c^2 \cdot (\eta_T \cdot u_{tr} \cdot b^2)^{-1} \cdot F\,, \quad \overline{\kappa}(\varepsilon) = c^2 \cdot (\eta_T \cdot u_{tr} \cdot b^2)^{-1} \cdot \kappa(\varepsilon)\,, \quad \overline{u} = u / u_{tr}\,,$$

$$\Delta_1 = \begin{cases} 1 & \text{if } (\varepsilon - \varepsilon_{tr}) > 0 \\ 0 & \text{if } (\varepsilon - \varepsilon_{tr}) < 0 \end{cases}, \quad \operatorname{sgn}(\overline{u}) = \begin{cases} 1 & \text{if } \overline{u} \ge 0 \\ -1 & \text{if } \overline{u} < 0 \end{cases}.$$

The stiffness $\kappa(\varepsilon)$ of asperities in contact shaft – bushing is estimated from the equation (7):

$$\overline{\kappa}(\varepsilon) = \overline{\kappa}_0 \cdot (\varepsilon - \varepsilon_{tr}) \cdot (\varepsilon_b - \varepsilon_{tr})^{-1}, \tag{7}$$

where:
$\overline{\kappa}_0$ [–] – dimensionless spring constant equivalent to the stiffness of the asperities, given by (8).

$$\overline{\kappa}_0 = \overline{F} \cdot (\varepsilon_b - \varepsilon_{tr})^{-1}, \tag{8}$$

where:
ε_b [m] – the eccentricity at the border between boundary and mixed lubrication in the Stribeck curve; it can be calculated from the journal velocity, u_b, at this borderline.

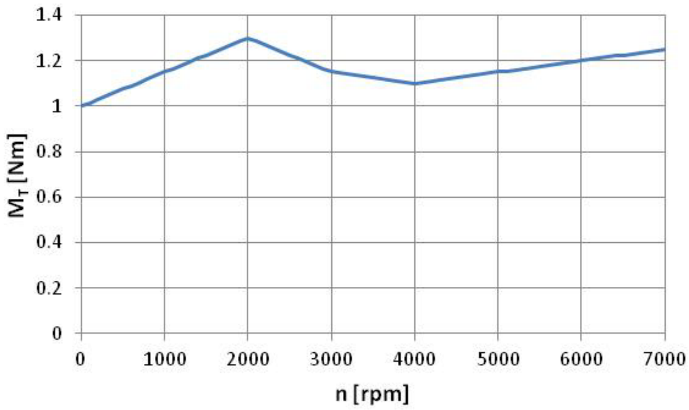

Fig. 2. Effect of engine speed n on the friction torque M_T in the bearings of rocker arms in the valve train for non-friction modified SAE oil at 373 K. Source: own calculations

The film thickness h in the hydrodynamic journal bearing can be modelled using a function of the peripheral angle θ, starting from the actual bearing symmetry plane, as given by (9) [15]:

$$h = c \cdot (1 + \cos\theta)\,, \tag{9}$$

The friction force F_f was obtained numerically, for: $\varepsilon_{tr} = 0.96$, $\varepsilon_b = 0.99$, $u_{tr} = 0.125$ m/s.

5. Results

The sample original characteristics for the engine measured were shown in the Fig. 3. The obtained courses of measured pressure in cylinder *p*, forces loading axis of outlet rocker arm F_{outlet} and inlet rocker arm F_{inlet} against crankshaft angle *CA* for the engine were shown in the Fig. 4. In the beginning of opening the outlet valve, the values of pressure *p* are up to 0.5 MPa and then they decrease quickly. The obtained values of the force F_{outlet} are clearly higher than of $F_{inlet.}$

Values of calculated friction torque for different case of bearing operation were shown in the Tab. 2 and were of an order smaller than those from [2]. They were obtained for two values of operating temperatures T = 293 K and T = 353 K. They were calculated for the case of inlet and outlet rocker arm. There were considered three configurations of bearing: sliding one as in original engine, sliding one after modification into set of two bearings and needle roller one after modification into set of two bearings.

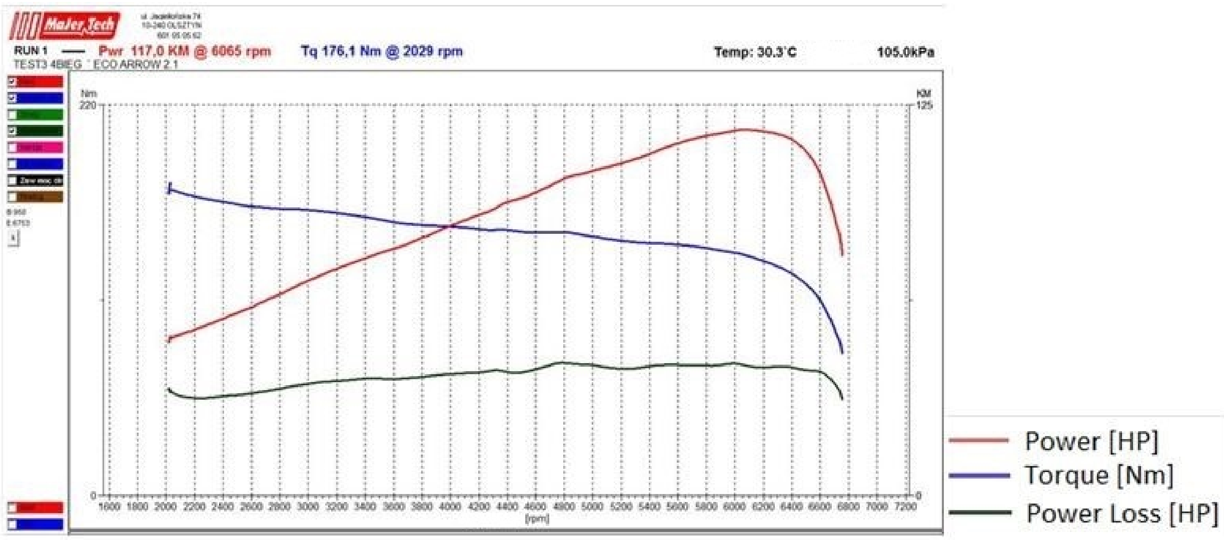

Fig. 3. Characteristics for the analysed engine. Values on the vertical axes divided by 100. Source: own studies

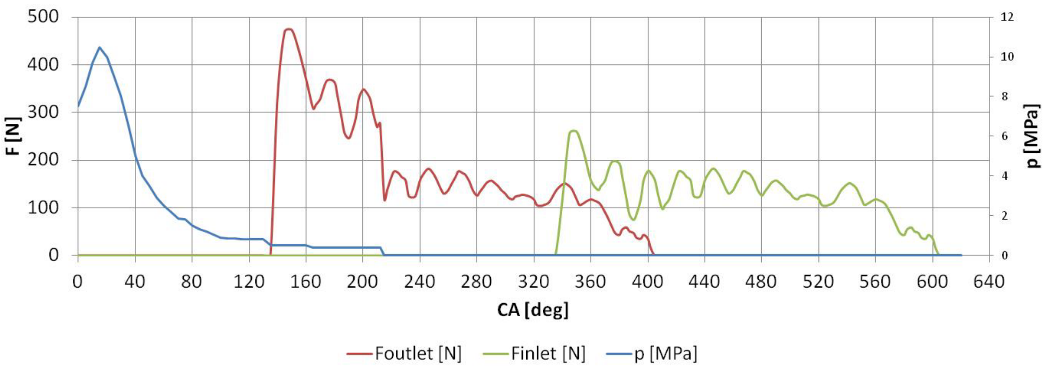

Fig. 4. Sample courses of measured pressure in cylinder p, loading forces loading axis of outlet rocker arm F_{outlet} and inlet rocker arm F_{inlet} against crankshaft angle CA. Source: own investigations

Tab. 2. Friction torque for different bearing of rocker arm

	T = 293 K		T = 353 K	
Bearing	Inlet	Outlet	Inlet	Outlet
Needle roller	0.004956	0.002058	0.004956	0.002058
Sliding original	0.03304	0.01372	0.02478	0.01029
Sliding modified	0.028084	0.011662	0.019824	0.008232

6. Summary

Obtained values of loading the analysed bearing in case of outlet were up 40% higher than in inlet case. After modification of original rocker arm bearing the maximum values of friction torque decreases with 15-17% for the T = 293 K and 21-24% for the T = 353 K. The decreasing was higher for the outlet bearing. Values of friction torque for needle roller bearing were smaller with an order than these were for sliding bearing. Changes in the temperature have had no effect for the friction torque in the case of the needle roller bearing.

Acknowledgement

This paper is partially supported by research project No DIR.5210.16.2016/1

References

[1] Chung, C.-S., Kim, H.-K., *Safety evaluation of the rocker arm of a diesel engine*, Mater. Des., Vol. 31, No. 2, pp. 940-945, 2010.
[2] Crolla, D., *Encyclopedia of Automotive Engineering*, John Wiley & Sons, 23.03.2015 – 4101.
[3] Husain, S.M., Sheikh, S., *Rocker arm: A review*. Int. J. Innovat. Res. Sci. Eng. Technol. Vol. 2, p. 4, 2013.
[4] Kano, M., Tanimoto, I., *Wear resistance properties of ceramic rocker arm pads*, Wear, Vol. 145, No. 1, pp. 153-165, 1991.
[5] Khurram, M., Mufti, R.A., Zahid, R., et al., *Effect of lubricant chemistry on the performance of end pivoted roller follower valve train*, Tribol. Int., Vol. 93, B, pp. 717-722, 2016.
[6] Lee, D.W., Cho, S.S., Joo, W.S. *An estimation of failure stress condition in rocker arm shaft through FEA and microscopic fractography*, J. Mech. Sci. Technol., Vol. 22, No. 11, pp. 2056-2061, 2008.
[7] Lee, D.W., Lee, S.J., Cho, S.S., et al., *Failure of rocker arm shaft for 4-cylinder SOHC engine*, Eng. Fail. Anal., Vol. 12, No. 3, pp. 405-412, 2005.
[8] Lubas, J., *Practical application of boron-modified sliding pairs in I.C. engine*, Tribol. Int., Vol. 43, No. 11, pp. 2046-2050. 2010,
[9] Makówka, M., Pawlak, W., Konarski, P., Wendler, B., *Hydrogen content influence on tribological properties of nc-WC/a-C:H coatings*, Diamond and Related Materials, Vol. 67, pp. 16-25, 2016.
[10] Mushiri, T., Mbohwa, C., *Finite element analysis of a car rocker arm*, Proceedings of the 2015 International Conference on Operations Excellence and Service Engineering Orlando, Florida, USA, September 10-11, 2015.
[11] NTN Needle Roller Bearing Handbook, CAT. No. 9013/E, http://www.ntn-snr.com/portal/fr/en-en/file.cfm/NB-HB_en.pdf?contentID=8899 (available in 2016.04.26).
[12] Nunney, M.J., *Light and heavy vehicle technology*. New York: Elsevier, 2013.
[13] Palomaki, E., Hoffman, K., *Stock Rocker vs. Roller Rocker*, New York: Rensselaer Polytechnic Institute, 2003.
[14] *Pick-a-part: Selecting valve train components for a performance build*, Engine Builder, February 2014.
[15] Rachoor, H., Investigation of Dynamic Friction in Lubricated *Surfaces*, PhD dissertation, New Jersey's Science & Technology University, 1996.
[16] Stachowiak, G., Batchelor, A., *Engineering Tribology*, 2nd ed., Butterworth-Heinemann 2001.
[17] Tuckart, W. *Failure analysis of pushrod valve train of stationary diesel engine*, International Workshop of Tribology TRIBaires 2013; May 7-9, 2013, pp. 40-41.
[18] *Valve lash adjustment elements*, INA product information. Retrieved from: http://www.wilmink.nl/wilmink/produkt/ina_klepstoter_info.pdf. (available in 2016.04.26).
[19] Wang, Y., *Introduction to Engine Valvetrains*, SAE, Warrandale, PA, 2007.

[20] Wituszynski, K., *Suboptymalna synteza i analiza rozrządu krzywkowego silników spalinowych*, Prace Instytutu Technologii i Eksploatacji Maszyn Politechniki Lubelskiej, Seria A, Nr 17, Lublin, Poland 1984.
[21] Yu, Z. W., Xu, X. L., *Failure analysis of diesel engine rocker arms*, Eng. Fail. Anal., Vol. 13, No. 4, pp. 598-605, 2006.
[22] Zhu, D., Cheng, H. S., *Tribological performance of ceramic roller followers/camshaft system in automobile valve*, Tribol. Ser., Vol. 18, pp. 149-156, 1991.
[23] http://performancetrends.com/Definitions/Rocker-Arms.htm (available in 2016.04.26).
[24] http://www.skf.com/group/industry-solutions/two-three-wheelers/applications/engine/valve-actuation/skf-rocker-arm-bearing/index.html (available in 2016.04.26).
[25] http://www.skf.com/us/products/bearings-units-housings/ball-bearings/principles/friction/estimating-frictional-moment/index.html (avail. in 2016.04.26).