THE STRUCTURE, GEOMETRY, AND KINEMATICS OF A UNIVERSAL JOINT

Florian Ion Tiberiu Petrescu  
IFToMM, Romania  
E-mail: fitpetrescu@gmail.com

Relly Victoria Virgil Petrescu  
IFToMM, Romania  
E-mail: rvvpetrescu@gmail.com

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ABSTRACT

The paper briefly presents the geometry, structure, and kinematics of a universal joint, very commonly used in machine building, especially today for heavy and engine-driven vehicles and transmissions located in different areas as well as for all-wheel-drive vehicles. The universal joint, or the cardan cross, conveys the rotation movement from one bridge to the other (when the rotary shaft suffers both movements, upward and downward). The kinematic scheme of a cardan transmission is composed of two cardan shafts (one input and one output), both of which are equipped with a cardan cross (universal joint or universal joint). Between the two universal couplings, a further (additional) cardan shaft (axle) is mounted. This mechanism is designed to transmit the mechanical movement (within a vehicle) from one bridge to the other. If the vehicle's motor is on the front and with on the rear axle transmission, or vice versa when the vehicle's engine is on the rear and the transmission is on the front axle, or when we have multiple (multi-axle) transmission on heavy vehicles or 4x4 cars.

Keywords: Cardan transmission; Universal joint; Heavy vehicles; Angular speed variation; Rotation movement; Geometry; Structure; kinematics.
1. INTRODUCTION

The kinematic scheme of a cardan transmission can be seen in figure (1). It is composed of two cardan shafts (one input and one output), both of which are equipped with a cardan cross (universal joint or universal joint). Between the two universal couplings, a further (additional) cardan shaft (axle) is mounted. This mechanism is designed to transmit the mechanical movement (within a vehicle) from one bridge to the other. If the vehicle's motor is on the front and with on the rear axle transmission, or vice versa when the vehicle's engine is on the rear and the transmission is on the front axle, or when we have multiple (multi-axle) transmission on heavy vehicles or 4x4 cars (PETRESCU, 2012b).

![Figure 1: The cardan transmission](image)

We most often encounter this transmission on buses, trucks, trolleybuses, trains, heavy cars, trucks, and all-wheel drive cars. It usually transmits the rotation movement from the front axle to the rear axle. It is necessary because there are big games in the transmission both left-right and up and down. It is the only mechanism that can transmit a rotation motion on a long axis that moves at the same time laterally and up and down. Generally, the yield of such a transmission is quite high, even if the rotation speed varies within the mechanism, but it is reconstituted at the end.

In 2010, more than 800 million vehicles circulate across the planet (ANTONESCU, 2000; ANTONESCU; PETRESCU, 1985; ANTONESCU; PETRESCU, 2012b).
2. UNIVERSAL JOINT (CARDAN COUPLING)

The cardan coupling mechanism is a spherical mechanism (see figure 2) due to the spherical motion imposed by any universal coupler. In a cardan coupling the four rotation axes are competing in a point S (Figure 2); (PETRESCU, 2012b).

The spherical mechanism is crank-type if the angle $\delta_1$ is the smallest in relation to the angles $\delta_2$, $\delta_3$, $\delta_0$ (figure 2a). In the case of the cardan mechanism, the element 2 (element representing a ball moving on a spherical surface with the center...
in S) is materialized by the two moving axes $\Delta_{12}$ and $\Delta_{23}$ (figure 2b). The specificity of the cardan mechanism is that the angles $\delta_1$, $\delta_2$, $\delta_3$ are all 90° and the angle $\delta_0$ between the two fixed axes is obtuse, the angle $\delta=\alpha$ (the attachment of $\delta_0$) being sharpened (Figure 2c).

The analytical calculation of the input-output transmission function is done by means of the kinematic diagram of the cardanic mechanism (Figure 2c) and the schematic diagram of the $u_1$ and $u_2$ versions of the axes $\Delta_{12}$ and $\Delta_{23}$, where (figure 3a) is denoted by $\Delta_1$ axis of the drive shaft and with the $\Delta_2$ axis of the driven shaft. With these notations (Figure 3a), the transmission function between the input and output shaft (driven) is of the form $\varphi_2(\varphi_1)$ or $\psi_2(\varphi_1)$.

![Figure 3: Cardan transmission: Versors of the transmit function](image)

For this purpose one considers the mobile versor $\vec{u}_1$ and $\vec{u}_2$ (Fig. 3a), which are orthogonal and oriented in the directions of the moving axes $\Delta_{12}$ and $\Delta_{32}$ competing in the center $O = S$ (Figure 2).

The versor $\vec{u}_1$ rotates in the plane $[y_1z_1]$ and is positioned with the angle $\varphi_1$ from the $y_1$ axis (Figure 3). The vector $\vec{u}_2$ rotates with the angle $\varphi_2=\psi_2$ in the plane $[y_2z_2]$, being positioned relative to the common axis $z_2 = z_1$ (Figure 3b).

The two versors are analyzed analytically by their components on axes $x_1$, $y_1$ and $z_{1,2}$ (Figure 3a), according to the relations of the system (1).

\[
\begin{align*}
\vec{u}_1 &= (\cos \varphi_1) \cdot \vec{j}_1 + (\sin \varphi_1) \cdot \vec{k}_1 \\
\vec{u}_2 &= -(\sin \psi_2)(\sin \alpha) \cdot \vec{j}_1 + (\sin \psi_2)(\cos \alpha) \cdot \vec{j}_1 + (\cos \psi_2) \cdot \vec{k}_1
\end{align*}
\]

(1)

From the $\vec{u}_1 \cdot \vec{u}_2 = 0$ (perpendicularity of the versors $\vec{u}_1$ and $\vec{u}_2$) condition, the relationship between their projections on the fixed axes (2) is deduced.
Expression (2) is written in the default form (3) in which the rotation angle of the output shaft 2 is based on the angle of rotation of the input shaft 1, but also according to the sharp angle $\alpha=\delta$.

$$\tan \psi_2 = \frac{1}{\cos \alpha} \cdot \tan \varphi_1$$

Then the 0 transmission function gets the expression (4).

$$\psi_2 = \arctan \left( \frac{1}{\cos \alpha} \cdot \tan \varphi_1 \right)$$

By derivation the expression of reduced angular velocity (5) is obtained. Initially the expression is also depending on $\psi_2$, and at the end it is expressed only according to $\varphi_1$ (and of course by $\alpha$).

$$U \equiv \psi_2' = \frac{\omega_2}{\varphi_1} = \frac{\omega_2}{\omega_1} \cdot \frac{\cos^2 \psi_2}{\cos^2 \varphi_1} = \frac{\cos \alpha}{\cos^2 \varphi_1 + \sin^2 \varphi_1}$$

By a new derivation is obtained also the order of transmission 2, respectively the reduced angular acceleration (relation 6).

$$W \equiv \psi_2'' = \frac{\varepsilon_2}{\varphi_1} = \frac{\varepsilon_2}{\omega_1^2} = \frac{-\cos \alpha \cdot \sin^2 \alpha \cdot \sin(2\varphi_1)}{\left(\cos^2 \alpha \cdot \cos^2 \varphi_1 + \sin^2 \varphi_1\right)^2}$$

The extreme values of the reduced angular velocity of the driven shaft 2 are obtained from its analytical expression with the limit conditions for the input angle $\varphi_1$.

$$\begin{align*}
\varphi_1 &= 0, \pi, 2\pi \Rightarrow \psi_2'_{\text{max}} = \frac{1}{\cos \alpha} \\
\varphi_1 &= \frac{\pi}{2}, \frac{3\pi}{2} \Rightarrow \psi_2'_{\text{min}} = \cos \alpha
\end{align*}$$

The dual cardan mechanism is obtained by mounting two simple cardan shafts in series, so that the two intermediate shaft forks are coplanar (see Figure 4).
3. DOUBLE CARDAN MECHANISM

The double cardan mechanism has the advantage of performing the synchronous movement between the input shafts $\Delta_1$ and the output $\Delta_3$ (Figure 4a). The intermediate shaft with the axle $\Delta_2$ has two fixed points $O_1 \in \Delta_1$ and $O_2 \in \Delta_3$ and so that it no longer needs a material bonded to the fixed bed (Figure 4a).

For the output shaft there are two positions (see Figure 4b): one $\Delta_3 \parallel \Delta_1$, and another $\Delta'_3$ symmetrical with the axis $\Delta_3$ in relation to the axis $\Delta_2$. The synchronization of the two movements (input-output) can be proved by means of the 0 transmission function which is written for the two simple cardan couplings (figure 4a), (relation 8); (Petrescu, 2012b).

$$\tan \psi_2 = \frac{1}{\cos \alpha} \cdot \tan \varphi_1 = \frac{1}{\cos \alpha} \cdot \tan \psi_3 \Rightarrow \tan \varphi_1 = \tan \psi_3 \Rightarrow \psi_3 = \varphi_1$$ (8)

The dual cardan coupling variant with the parallel axle output and input is generally used on heavy goods vehicles (lorries) so that the distance between the two axes may vary within certain limits (see Figure 5); In this situation a variable length of the intermediate shaft 2 is required, constructively made by a telescopic intermediate shaft.

Figure 4: Cardan transmission (double cardan mechanism); the intermediate shaft is observed
The two pieces of the axle $\Delta_2$ și $\Delta'_2$, have the coplanar forks (Figure 5) and are connected by a transverse groove coupling, allowing the variation of the length $O_10_2$ and the angle dimension $\alpha = \delta$ when the distance between the axes $h_{13}$ varies.

If the two forks of the intermediate shaft $\Delta_2$ are not coplanar (see figure 6), the movement of the driven shaft $\Delta_3$ is no longer synchronous with that of the drive shaft $\Delta_1$.

4. CONCLUSIONS

Although apparently the dynamic loading of the double (dual) transmission increases with the mechanical (mechanical) inertial moment, the effect itself is negligible under actual operating conditions (for normal cardan transmissions, built and properly mounted).

The elasticity of the intermediate shaft influences the homokinetic deviation of the transmission as follows: a) in the usual cases the bicardan transmission becomes quasi-homo-kinetic and therefore the deviation from homokinetic can be virtually neglected; b) In special cases with long (or very long) intermediate shafts and high (or very high) mechanical moments of inertia it is necessary to offset the homokinetic deviation by designing the transmission so that the deviation from homokinetic becomes null, or negligible. c) Under normal operating conditions, the
influence of the elasticity of the intermediate shaft on torsion moments may be neglected.

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7. AUTHORS’ CONTRIBUTION

All the authors have contributed equally to carry out this work.

REFERENCES

ANGELES, J.; LOPEZ-CAJUN, C. (1988) Optimal synthesis of cam mechanisms with oscillating flat-face followers. *Mechanism Mach. Theory*, v. 23, n. 1, p. 1-6. doi: 10.1016/0094-114X(88)90002-X

ANTONESCU, P. (2000) *Mechanisms and Handlers*, Printech Publishing House. Bucharest.

ANTONESCU, P.; PETRESCU, F. I. T. (1985) Analytical method of synthesis of cam mechanism and flat stick. *Proceedings of the 4th International Symposium on Mechanism Theory and Practice (TPM' 85)*, Bucharest.

ANTONESCU, P.; PETRESCU, F. I. T. (1989) Contributions to cinetoelasodynamic analysis of distribution mechanisms. *SYROM'89*, Bucharest.

ANTONESCU, P.; OPREAN, M.; PETRESCU, F. I. T. (1985a) Contributions to the synthesis of oscillating cam mechanism and oscillating flat stick. *Proceedings of the 4th International Symposium on Theory and Practice of Mechanisms, (TPM' 85)*, Bucharest.

ANTONESCU, P.; OPREAN, M.; PETRESCU, F. I. T. (1985b) At the projection of the oscillante cams, there are mechanisms and distribution variables. *Proceedings of the V-Conference for Engines, Automobiles, Tractors and Agricultural Machines, I-Engines and Automobiles, (AMA’ 85)*, Brasov.

ANTONESCU, P.; OPREAN, M.; PETRESCU, F. I. T. (1986) Projection of the profile of the rotating camshaft acting on the oscillating plate with disengagement. *Proceedings of the 3rd National Computer Assisted Designing Symposium in Mechanisms and Machine Bodies*, (MOM’ 86), Brasov.
ANTONESCU, P.; OPREAN, M.; PETRESCU, F. I. T. (1987) Dynamic analysis of the cam distribution mechanisms. *Proceedings of the Seventh National Symposium of Industrial Robots and Spatial Mechanisms (IMS' 87)*, Bucharest.

ANTONESCU, P.; OPREAN, M.; PETRESCU, F. I. T. (1988) Analytical synthesis of Kurz profile, rotating flat cam cam. *Machine Build. Rev.* Bucharest.

ANTONESCU, P.; PETRESCU, F. I. T.; ANTONESCU, O. (1994) Contributions to the synthesis of the rotating cam mechanism and the tip of the balancing tip. Brasov.

ANTONESCU, P.; PETRESCU, F. I. T.; ANTONESCU, O. (1997) Geometrical synthesis of the rotary cam and balance tappet mechanism. Bucharest.

ANTONESCU, P.; PETRESCU, F. I. T.; ANTONESCU, O. (2000a) Contributions to the synthesis of the rotary disc-cam profile. *Proceedings of the 8th International Conference on Theory of Machines and Mechanisms, (TMM' 00)*, Liberec, Czech Republic, p. 51-56.

ANTONESCU, P.; PETRESCU, F. I. T.; ANTONESCU, O. (2000b) Synthesis of the rotary cam profile with balance follower. *Proceedings of the 8th Symposium on Mechanisms and Mechanical Transmissions (MMT' 00)*, Timișoara, p. 39-44.

ANTONESCU, P.; PETRESCU, F. I. T.; ANTONESCU, O. (2001) Contributions to the synthesis of mechanisms with rotary disc-cam. *Proceedings of the 8th IFToMM International Symposium on Theory of Machines and Mechanisms, (TMM' 01)*, Bucharest, ROMANIA, p. 31-36.

AVERSA, R.; PETRESCU, R. V.; APICELLA, A.; PETRESCU, F. I. T.; CALAUTIT, J. K.; MIRSAYAR, M. M.; BUCINELL, R.; BERTO, F.; AKASH, B. (2017a) Something about the V Engines Design. *Am. J. Appl. Sci.*, v. 14, n. 1, p. 34-52. doi: 10.3844/ajassp.2017.34.52

AVERSA, R.; PETRESCU, R. V.; AKASH, B.; BUCINELL, R.; CORCHADO, J.; BERTO, F.; MIRSAYAR, M. M.; CHEN, G.; LI, S.; APICELLA, A.; PETRESCU, F. I. T. (2017) Something about the Balancing of Thermal Motors. *Am. J. Eng. Appl. Sci.*, v. 10, n. 1, p. 200-217. doi: 10.3844/ajeassp.2017.200.217

AVERSA, R.; PETRESCU, R. V.; APICELLA, A.; PETRESCU, F. I. T. (2017) A Dynamic Model for Gears. *Am. J. Eng. Appl. Sci.*, v. 10, n. 2, p. 484-490. doi: 10.3844/ajeassp.2017.484.490

AVERSA, R.; PETRESCU, R. V.; PETRESCU, F. I. T.; APICELLA, A. (2017d) Smart-Factory: Optimization and Process Control of Composite Centrifuged Pipes. *Am. J. Appl. Sci.*, v. 13, n. 11, p. 1330-1341. doi: 10.3844/ajassp.2016.1330.1341

AVERSA, R.; TAMBURRINO, F.; PETRESCU, R. V.; PETRESCU, F. I. T.; ARTUR, M.; CHEN, G.; APICELLA, A. (2017e) Biomechanically Inspired Shape Memory Effect Machines Driven by Muscle like Acting NiTi Alloys. *Am. J. Appl. Sci.*, v. 13, n. 11, p. 1264-1271. doi: 10.3844/ajassp.2016.1264.1271

FAWCETT, G. F.; FAWCETT, J. N. (1974) *Comparison of polydyne and non polydyne cams*. In: Cams and cam mechanisms. Ed. J. Rees Jones, MEP, London and Birmingham, Alabama, 1974.

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GIORDANA, F.; ROGNONI, V.; RUGGIERI, G. (1979) On the influence of measurement errors in the Kinematic analysis of cam. Mechanism Mach. Theory, v. 14, n. 5, p. 327-340. doi: 10.1016/0094-114X(79)90019-3

HAIN, K. (1971) Optimization of a cam mechanism to give good transmissibility maximal output angle of swing and minimal acceleration. J. Mechanisms, v. 6, n. 4, p. 419-434. doi: 10.1016/0022-2569(71)90044-9

JONES, J. R.; REEVE, J. E. (1974) Dynamic response of cam curves based on sinusoidal segments. In: Cams and cam mechanisms. Ed. J. Rees Jones, MEP, London and Birmingham, Alabama.

KOSTER, M. P. (1974) The effects of backlash and shaft flexibility on the dynamic behavior of a cam mechanism. In: Cams and cam mechanisms, Ed. J. Rees Jones, MEP, London and Birmingham, Alabama, 1974.

MIRSAYAR, M. M.; JONEIDI, V. A.; PETRESCU, R. V.; PETRESCU, F. I. T.; BERTO, F. (2017) Extended MTSN criterion for fracture analysis of soda lime glass. Eng. Fract. Mech., v. 178, p. 50–59. doi: 10.1016/j.engfracmech.2017.04.018

PETRESCU, R. V.; AVERSA, R.; AKASH, B.; BUCINELL, R.; CORCHADO, J.; BERTO, F.; MIRSAYAR, M. M.; CALAUTIT, J. K.; APICELA, A.; PETRESCU, F. I. T. (2017a) Yield at Thermal Engines Internal Combustion. Am. J. Eng. Appl. Sci., v. 10, n. 1, p. 243-251. doi: 10.3844/ajeassp.2017.243.251

PETRESCU, R. V.; AVERSA, R.; AKASH, B.; BUCINELL, R.; CORCHADO, J.; BERTO, F.; MIRSAYAR, M. M.; CALAUTIT, J. K.; APICELA, A.; PETRESCU, F. I. T. (2017b) Forces at Internal Combustion Engines. Am. J. Eng. Appl. Sci., v. 10, n. 2, p. 382-393. doi: 10.3844/ajeassp.2017.382.393

PETRESCU, R. V.; AVERSA, R.; AKASH, B.; BUCINELL, R.; CORCHADO, J.; BERTO, F.; MIRSAYAR, M. M.; APICELA, A.; PETRESCU, F. I. T. (2017c) Gears-Part I. Am. J. Eng. Appl. Sci., v. 10, n. 2, p. 457-472. doi: 10.3844/ajeassp.2017.457.472

PETRESCU, R. V.; AVERSA, R.; AKASH, B.; BUCINELL, R.; CORCHADO, J.; BERTO, F.; MIRSAYAR, M. M.; APICELA, A.; PETRESCU, F. I. T. (2017d) Gears-Part II. Am. J. Eng. Appl. Sci., v. 10, n. 2, p. 473-483. doi: 10.3844/ajeassp.2017.473.483

PETRESCU, R. V.; AVERSA, R.; AKASH, B.; BUCINELL, R.; CORCHADO, J.; BERTO, F.; MIRSAYAR, M. M.; APICELA, A.; PETRESCU, F. I. T. (2017e) Cam-Gears Forces, Velocities, Powers and Efficiency. Am. J. Eng. Appl. Sci. v. 10, n. 2, p. 491-505. doi: 10.3844/ajeassp.2017.491.505

PETRESCU, R. V.; AVERSA, R.; AKASH, B.; BUCINELL, R.; CORCHADO, J.; BERTO, F.; MIRSAYAR, M. M.; KOSAITIS, S.; ABU-LEBDEH, T.; APICELA, A.; PETRESCU, F. I. T. (2017f) Dynamics of Mechanisms with Cams Illustrated in the Classical Distribution. Am. J. Eng. Appl. Sci., v. 10, n. 2, p. 551-567. doi: 10.3844/ajeassp.2017.551.567

PETRESCU, R. V.; AVERSA, R.; AKASH, B.; BUCINELL, R.; CORCHADO, J.; BERTO, F.; MIRSAYAR, M. M.; KOSAITIS, S.; ABU-LEBDEH, T.; APICELA, A.; PETRESCU, F. I. T. (2017g) Testing by Non-Destructive Control, Am. J. Eng. Appl. Sci., v. 10, n. 2, p. 568-583. doi: 10.3844/ajeassp.2017.568.583
PETRESCU, R. V.; AVERSA, R.; APICELA, A.; PETRESCU, F. I. T. (2017h) Transportation Engineering. Am. J. Eng. Appl. Sci., v. 10, n. 3, p. 685-702. doi: 10.3844/ajeassp.2017.685.702

PETRESCU, R. V.; AVERSA, R.; KOZAITIS, S.; APICELA, A.; PETRESCU, F. I. T. (2017i) The Quality of Transport and Environmental Protection, Part I. Am. J. Eng. Appl. Sci., v. 10, n. 3, p. 738-755. doi: 10.3844/ajeassp.2017.738.755

PETRESCU, F. I. T.; CALAUTIT, J. K.; MIRSAVAR, M. M.; MARINKOVIC, D. (2015) Structural Dynamics of the Distribution Mechanism with Rocking Tappet with Roll. Am. J. Eng. Appl. Sci., v. 8, n. 4, p. 589-601. doi: 10.3844/ajeassp.2015.589.601

PETRESCU, F. I. T.; PETRESCU, R. V. (2016) Otto Motor Dynamics. Geintec-Gestao Inovacao e Tecnologias, v. 6, n. 3, p. 3392-3406. doi: 10.7198/geintec.v6i3.373

PETRESCU, F. I. T.; PETRESCU, R. V. (2014) Cam Gears Dynamics in the Classic Distribution. Ind. J. Manag. Prod., v. 5, n. 1, p. 166-185. doi: 10.14807/ijmp.v5i1.133

PETRESCU, F. I. T.; PETRESCU, R. V. (2013a) An Algorithm for Setting the Dynamic Parameters of the Classic Distribution Mechanism. Int. Rev. Model. Simul., v. 6, n. 5, p. 1637-1641.

PETRESCU, F. I. T.; PETRESCU, R. V. (2013b) Dynamic Synthesis of the Rotary Cam and Translated Tappet with Roll. Int. Rev. Model. Simul., v. 6, n. 2, p. 600-607.

PETRESCU, F. I. T.; PETRESCU, R. V. (2013c) Cams with High Efficiency. Int. Rev. Mech. Eng., v. 7, n. 4, p. 599-606.

PETRESCU, F. I. T.; PETRESCU, R. V. (2013d) Forces and Efficiency of Cams. Int. Rev. Mech. Eng., v. 7, n. 3, p. 507-511.

PETRESCU, F. I. T.; PETRESCU, R. V. (2011) Dinamica mecanismelor de distributie. Create Space publisher, USA, (Romanian version). ISBN 978-1-4680-5265-7

PETRESCU, F. I. T.; PETRESCU, R. V. (2005a) Contributions at the dynamics of cams. In: Proceedings of the Ninth IFToMM International Symposium on Theory of Machines and Mechanisms, Bucharest, Romania, v. I, p. 123-128.

PETRESCU, F. I. T.; PETRESCU, R. V. (2005b) Determining the dynamic efficiency of cam. In: Proceedings of the Ninth IFToMM International Symposium on Theory of Machines and Mechanisms, Bucharest, Romania, v. I, p. 129-134.

PETRESCU, F. I. T. (2015a) Geometrical Synthesis of the Distribution Mechanisms. Am. J. Eng. Appl. Sci., v. 8, n. 1, p. 63-81. doi: 10.3844/ajeassp.2015.63.81

PETRESCU, F. I. T. (2015b) Machine Motion Equations at the Internal Combustion Heat Engines. Am. J. Eng. Appl. Sci., v. 8, n. 1, p. 127-137. doi: 10.3844/ajeassp.2015.127.137

PETRESCU, F. I. T. (2012a) Bazele analizei și optimizării sistemelor cu memorie rigidă – curs și aplicații. Create Space publisher, USA, (Romanian edition), 2012. ISBN 978-1-4700-2436-9
PETRESCU, F. I. T. (2012b) Teoria mecanismelor – Curs si aplicatii (editia a doua). Create Space publisher, USA, (Romanian version), 2012. ISBN 978-1-4792-9362-9

SAVA, I. (1970) Contributions to Dynamics and Optimization of Income Mechanism Synthesis. Ph.D. Thesis, I.P.B., 1970.

TARAZA, D.; HENEIN, N. A.; BRYZIK, W. (2001) The Frequency Analysis of the Crankshaft's Speed Variation: A Reliable Tool for Diesel Engine Diagnosis. J. Eng. Gas Turbines Power, v. 123, n. 2, p. 428-432. doi: 10.1115/1.1359479

TESAR, D.; MATTHEW, G. K. (1974) The design of modeled cam systems. In: Cams and cam mechanisms, Ed. J. Rees Jones, MEP, London and Birmingham, Alabama.

Wiederrich, J. L.; Roth, B. (1974) Design of low vibration cam profiles. In: Cams and Cam Mechanisms. Ed. J. Rees Jones, MEP, London and Birmingham, Alabama.