RADICAL REDUCTION OF CRANKSHAFT TORSIONAL VIBRATIONS

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The reduction of torsional vibrations in rotating shafts, specifically in engine's crankshafts will increase the engine's lifetime. The study of dynamic performance of new Torsional Vibration Damper based on US Patent 7,438,165 having the control system with instantaneous frequencies tuner is presented. The analysis of dynamics supported by test results obtained from experiments and mathematical simulations illustrate the effectiveness of the proposed tuned Torsional Vibration Damper.

РАДИКАЛЬНОЕ СНИЖЕНИЕ КРУТИЛЬНЫХ КОЛЕБАНИЙ КОЛЕНЧАТОГО ВАЛА

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Сокращение крутильных колебаний валов, в частности, коленчатых валов двигателей увеличивает продолжительность его работоспособности. Исследование динамических характеристик новых гасителей крутильных колебаний, основанных на патенте США 7,438,165, имеющих систему управления с мгновенной частотой тюнера представлена в работе. Анализ динамики находится в согласии с результатами тестов, полученных экспериментально и в при помощи математического моделирования, иллюстрирует эффективность предлагаемой настройки гасителя крутильных колебаний.

Operating engine crankshaft is loaded by dynamic forces, generating the various types of vibrations. One of the types of crankshaft vibration is torsional vibration. Reduction of torsional vibrations could be reached by imcorporation of Torsional Vibration Damper (TVD) in crankshaft structure. Every vehicle engine is equipped with TVD. Today TVD are using the rubber elements or viscous systems for dampening. All of them have the own disadvantages. New type of TVD is proposed in US Patent 7,438,165 [1] for avoiding disadvantages of rubber and viscous TVD, by presenting better performance dynamic characteristics. The structural components description of invented TVD is presented.

Torsional vibrations of engine crankshaft (see Figure 1) have two parameters: frequency of vibrations and amplitude of twist angle α(see Figure2). It is desired to obtain minimal (ideally zero) twist angle α of shaft in steady-speed operation of an engine. For this purpose the engine is equipped by Torsional Vibration Damper which is installed in front part of crankshaft.

Figure 1. Engine Crankshaft

There is an industry regulation for allowable magnitude of twist angle α of crankshaft – (a peak-to-peak twist angle) must be below 0.4 degree. The measured values of a peak-to-peak twist angle practically are in range 0.36 – 0.60 degrees for different types of engines equipped with Torsional Vibration Damper.
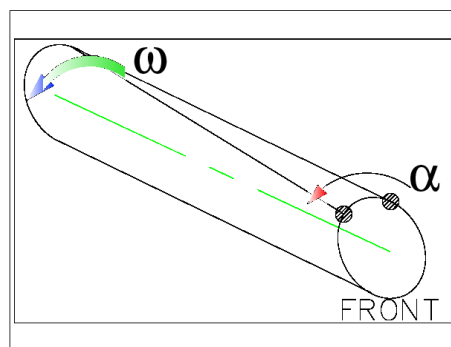

Figure 2. Graphical presentation of Crankshaft twist angle

For obtaining the lower values of a peak-to-peak twist angle the new type of torsional vibration damper was proposed in [1]. In Figure 3, there is shown the core part of proposed device 300. Hub 305 corresponds to the engine shaft. The hub is connected to

damper outer ring 314. A cantilever spring 316 is connected to ring 314, but is free at the near hub end, so between hub 305 and spring 316 there is a gap. Masses (or selectors) 318 are mounted freely on spring 316 and coupled to the hub 305 by springs 320. Selectors 318 are sliding along beam 316. Hence, selector 318 could move in radial direction using beam 316 and oscillating across clamp point of beam 316 to rim 314.
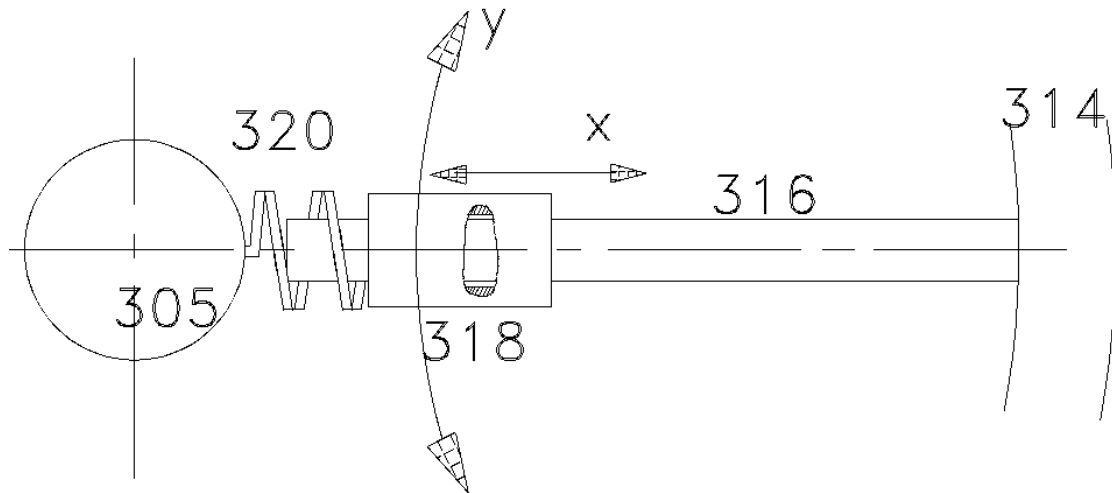

Figure 3. Graphical presentation of core elements of proposed TVD

Masses 318 are free to move along springs 316 except for the counter-forces due to the compression or extension of helical springs 320. Further, masses 318 are free to move perpendicular to the radius, except for the counter-forces due to displacement of springs 316. The entire apparatus 300 is symmetrical, in order to ensure the moment of inertia is centered about hub 305. As the radial position of the mass 318 is extended, the length of cantilever spring 316 for providing bending beam resistance is effectively shortened. The shortening of the beam, in turn, implies a greater perpendicular force, for a perpendicular displacement i.e. as the length of the bending beam is shortened, the spring constant of proportionality relating displacement to force 316 increases. These effects, the outward displacement of masses 318 under angular velocity and the corresponding increase of the spring constant of proportionality 316 as the masses 318 move outward, combined with select dimensioning of the spring 316, result in reducing of oscillations over the operating range of ω .The simplified dynamic graphical model of engine equipped by proposed Tuned Torsional Vibration Damper is presented in Figure 4. J is a mass moment of inertia of main system; "φ" is the angle of crankshaft rotation of main system. Main system with flywheel is connected to external load representing all resistance forces by shaft that has stiffness coefficient "k" and viscous friction coefficient of loss "b". Assume that "J_F" is a mass moment of inertia of conventional flywheel. Main system is the simple model of all moving and rotating parts of engine. Assume that to main system the following forces (torque) are applied: T_M is the driving torque, R_A is a resistance spent on making the positive useful work, R_L is the lost torque spent on friction, and other phenomena accompanied to negative resistance.
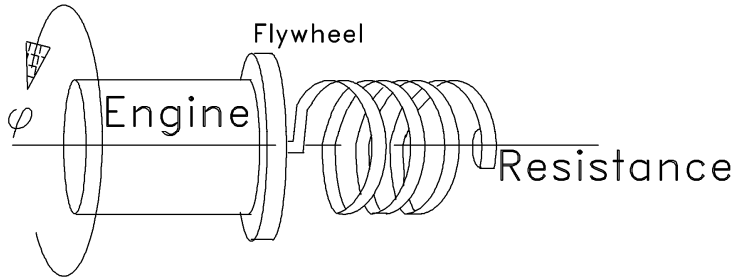
Figure 4. Simplified dynamic model.

It is selected that x is coordinate of moving selector 318 along springing beam 316 (as per Figure 3), y is the dynamic deflection of selector 318 across beam 316 (as per Figure 3).

Assume that **m** is the mass of selector 318, **r** is the constant distance from center of hub 305 to center of gravity of selector 318, **Z** is the constant deflection of spring beam from axis, c_L is the stiffness coefficient of spring 320, and c_Y is the stiffness coefficient of springing beam 316.

The differential equations for described system having installed proposed TVD will look like:

$$[(J + J_F) + mr^2(r^2 + 2rx + 2Zy)]\ddot{\varphi} + \ldots$$
$$+ 2mr^2\dot{\varphi}(r\dot{x} + Z\dot{y}) + k\varphi + b\dot{\varphi} = T_M - R_A - R_L. \tag{1}$$

$$mr^2\ddot{x} + c_L x = mr^3\dot{\varphi}^2 - mrZ\ddot{y} \tag{2}$$

$$mZ^2\ddot{y} + c_Y y = mr^2 Z\dot{\varphi}^2 + mrZ\ddot{x} \tag{3}$$

where
$` = d/dt$, and
$T_M = T_0 + T_1(\varphi_0, \dot{\varphi}_0),$
$R_A = R_{0A} + R_{1A}(\varphi_0, \dot{\varphi}_0),$
$R_L = R_{0L} + R_{1L}(\varphi_0, \dot{\varphi}_0).$
$$\tag{4}$$

From theory point of view the considered dynamic system has three degrees of freedom. The test parameters for torque and resistance for example for engine LS 7 are shown in Figure 5 as solid lines.
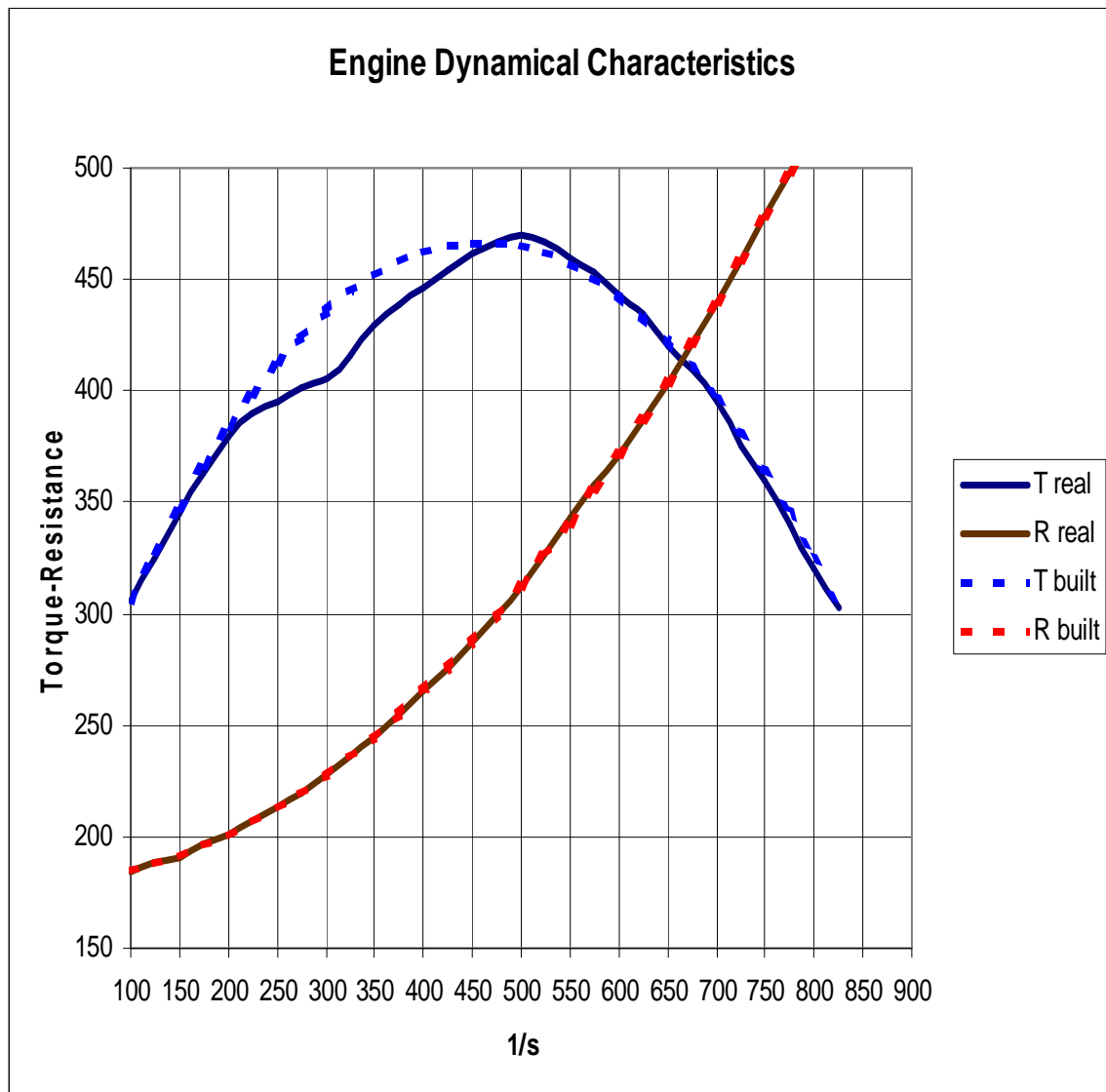
Engine Dynamical Characteristics

Figure 5. Parameter Curves for Engine LS 7

The approximate equation for driving torque (blue dashed line) would be
$$T_M = 204.4248 + 1.141593\varphi` - 0.001239\varphi`^2$$

$$(5\ A)$$

and for total resistance (red dashed line) is
$$R = 180 + 0.00053\ \varphi`^2$$

$$(5\ B)$$

Speed of rotation $\boldsymbol{\omega}$ is $\omega = \varphi` = d\varphi/dt$.
Stability condition in system is when
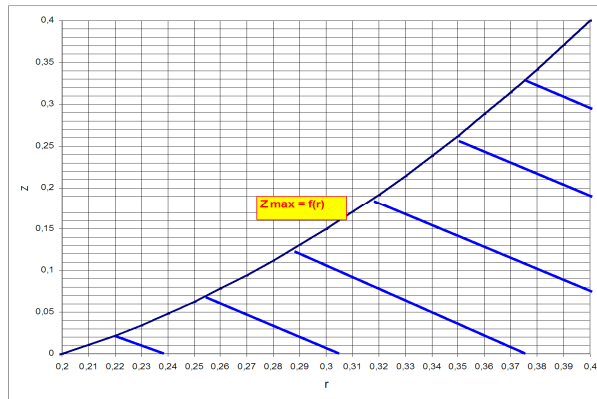$$(J + J_F) + mr^2(r^2 + 2rx + 2Zy) > 0$$

$$(6)$$

Figure 6. Selection of the constant deflection of a spring beam of TVD

Taking into consideration the last stability conditions the rules for proper parameter selection could be graphically represented. It is graphical chart shown in Figure 6 for selection of Z - the constant deflection of a spring beam. It is graphical chart shown in Figure 7 for proper selection of value of mass of selector. Dashed zones under curves indicate the area of parameters for stable regimes.
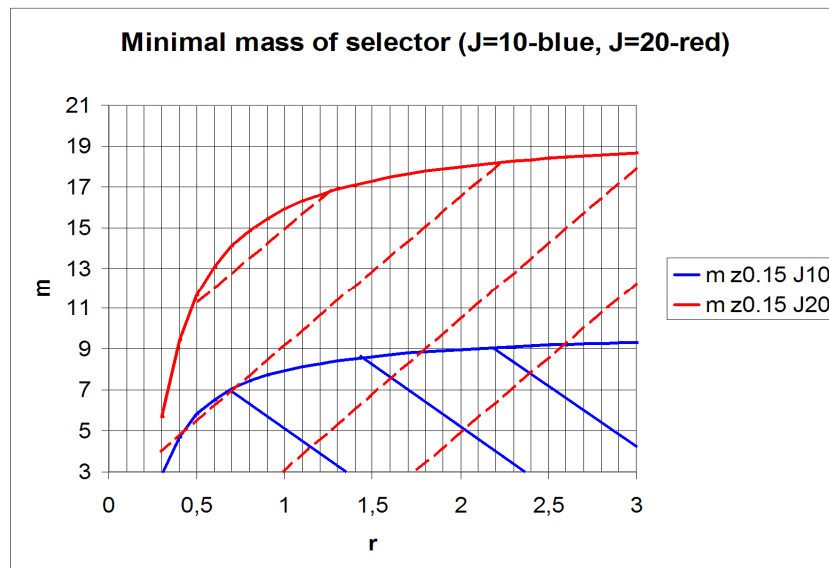

Figure 7. Selection of the mass of selector in TVD

The particularities of Torsional Vibration Damper usage could be visualized with help of Amplitude – Frequency Diagrams. The engine rotating system (crankshaft and all involving moving parts) is not totally stiff and there is some flexibility in shaft, so we could consider the system having flexible shaft as a dynamic system with one degree of freedom. Torsional Vibration Damper is assembled on the front of crankshaft through flexible joint. Dynamically it means that second degree of freedom is added.

The analysis of dynamical behavior is presented using the AFD – "amplitude – frequency diagram" as shown in Figure 8. In the chart on abscissa axis the dimensionless frequency numbers is shown and on ordinate axis the dimensionless amplitude of torsional angle vibration values is shown.
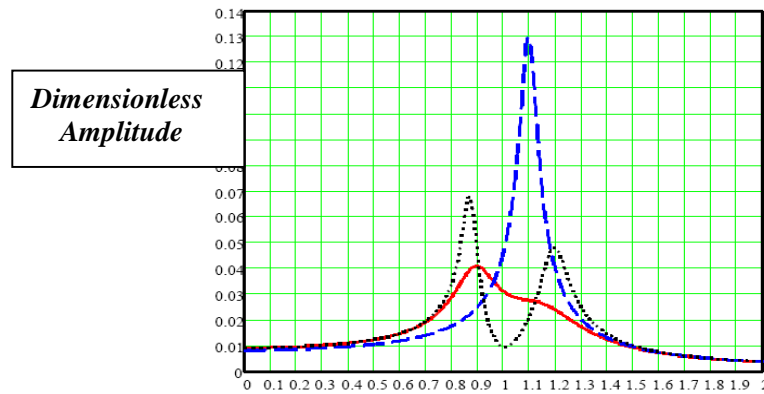

Figure 8. Amplitude - Frequency Diagram

Blue dashed curve represents AFD of torsion vibration for system without Torsional Vibration Damper. There is one peak of amplitude of 0.13. If Torsional Vibration Damper (J_{TVD} equal to 0.1J and neglected small dissipation) is installed, so the AFD should be like black dotted line having two peaks at frequencies 0.87 and 1.2, and one minimal value at frequency 1. The magnitude of this minimum is 0.01. From one point of view it is a good result, because the amplitude in resonance zone is reduced dramatically. But reduction is achieved in resonance zone only. From other point of view it is not good, because the system gets two resonance peaks, which are not treated. When Torsional Vibration Damper (J_{TVD} equal to 0.1J and adjusted dissipation) is installed, then the AFD should be like red line having also two peaks (like for previous case) at frequencies 0.9 and 1.15, and no minimal value in resonance zone. It is better situation in comparison to previous case, because peaks are lower, but worst situation is in resonance zone.

Comparison of dynamical effectiveness of traditional Torsional Vibration Damper and proposed Tuned Torsional Vibration Damper is presented in Figure 9. Similar to Figure 8 there is Dimensionless Frequency at abscissa axis and Dimensionless Displacement – Peak-to-Peak Twist Angle at ordinate axis in Figure 9. In this chart the brown dashed curve is equivalent to AFD in Figure 8 for adjusted dissipation case of Torsional Vibration Damper installation. The red curve is AFD built using system of equations (1) – (3). All values of amplitudes of torsion vibrations of system equipped with proposed Tuned Double Control Torsional Vibration Damper are lower (and in some areas radically lower) for all frequencies in comparison to system equipped with traditional regular (rubber or viscous) Torsional Vibration Damper.
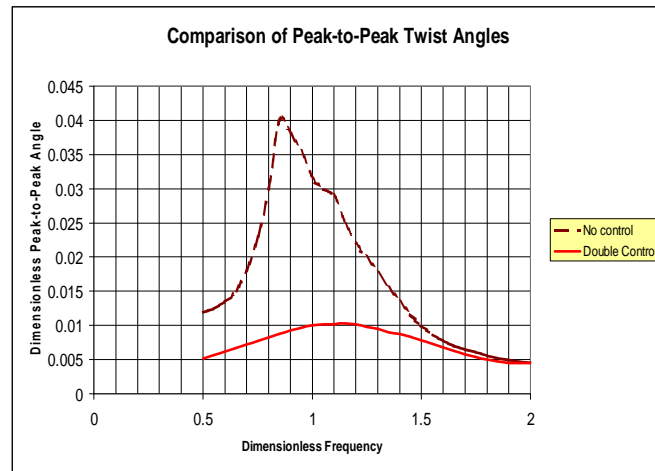
Figure 9. Comparison of values of a Peak-to-Peak Twist Angles for Tuned TVD. No Control vs. Double Controlled.

From point of view of dynamical efficiency it is much better to use Tuned Double Control Torsional Vibration Damper because it allows keeping the amplitude of torsion vibrations at very low levels in total range of operating frequencies. In other words there would be no problems with torsion vibration in all ranges of engine operational **rpm** when proposed Tuned Double Control Torsional Vibration Damper is installed**.** The name "Double Control" is reflecting that fact that Torsional Vibration Damper is controlled by two types of subsystems [1] – a mechanical control subsystem as shown in Figure 3, and an electronic subsystem as described in [1] on Figures 8 and 9.

If to use the mechanical control subsystem only (excluding an electronic subsystem) it should be AFD as presented in Figure 10. The thin dash-dotted curve is presented the dynamical behavior of Tuned Double Control Torsional Vibration Damper similar to shown in Figure 9, the solid red curve is presented Semi Controlled Torsional Vibration Damper. The curve for Semi Controlled Torsional Vibration Damper consists of three parts – part in the range of frequencies 0.5 – 1.25 is equal to the part for Tuned Double Control Torsional Vibration Damper (a mechanical control is predominant here), part in the range of frequencies 1.25 – 1.35 is for transient vibrations, and part in the range over frequency 1.35 is reflecting the dynamic of rotating shaft with firmly attached mass of Torsional Vibration Damper. The Semi Controlled Torsional Vibration Damper has two peaks: the small value peak at frequency of 1.15 and big value peak at frequency of 1.35.
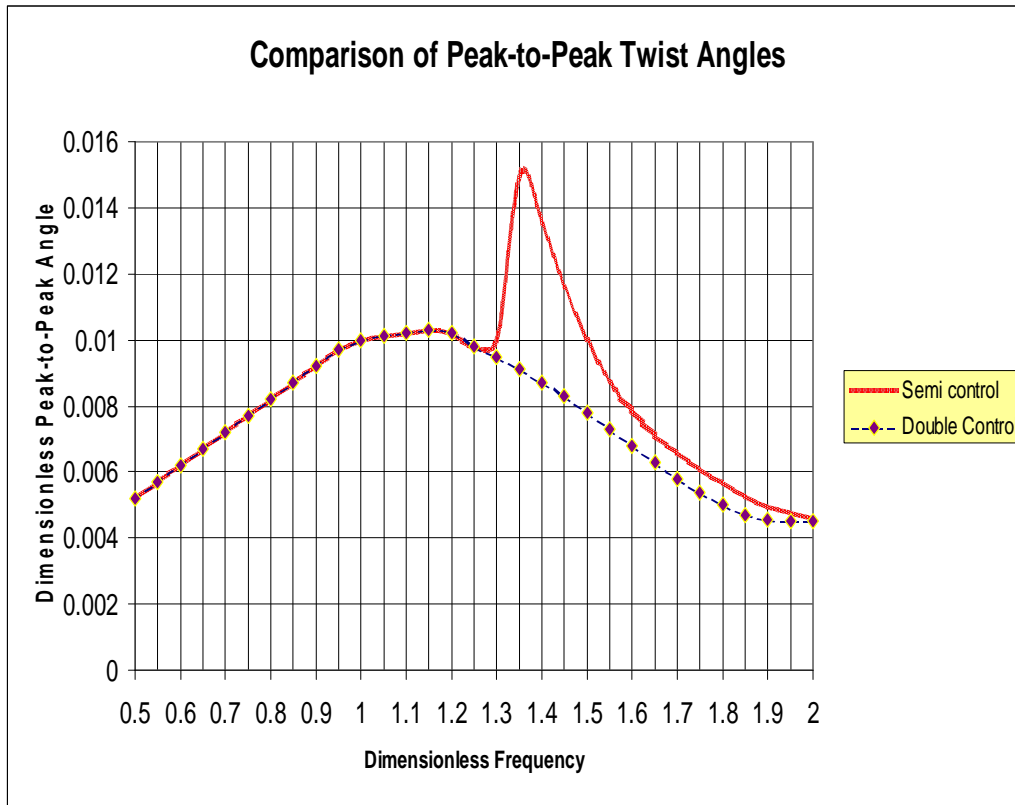
Figure 10. Comparison of values of a Peak-to-Peak Twist Angles for Tuned TVD. Semi Control vs. Double Controlled.

Comparison of a dynamical behavior of Tuned Double Control Torsional Vibration Damper and Semi Controlled Torsional Vibration Damper shows that a Tuned Double Control Torsional Vibration Damper has better performance. But when the electronic electromagnetic subsystem is excluded in Semi Controlled Torsional Vibration Damper, then that leads to simple and cheap manufacturing of Torsional Vibration Damper. Also it could be taken into consideration that in a big variety of engines the resonance peak is at the high magnitude of frequency sitting near the border of **rpm** range. So the second peak cannot be achieved in real situation. For instance, in VW Jetta the resonance problem exists for 6th order rpm (approximately at 4900 – 5000 rpm), and restricted rpm boarder number is 5000 rpm, so the engineer is dealing with lower frequency peak at 4500 – 4800 rpm, and second peak which could occur over 5000 rpm should be ignored.

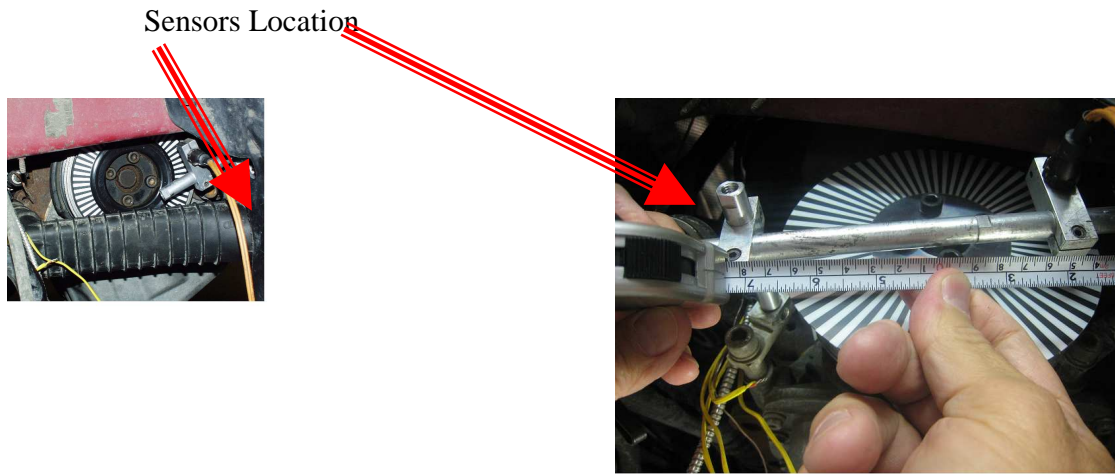Accepted sensors location for tested engine is illustrated in Figure 11.

Sensors Location

Figure 11. Sensors location during tests.

For experimental purposes the Diesel Engine 1.9L installed in VW Jetta was selected. Preliminary analysis and recorded experiments demonstrate that the torsional vibration resonance occurs in crankshaft of Diesel Engine 1.9L at 490 – 500 Hz. This is the lowest frequency where resonance happened. It is related to 6th order torsional vibrations of this engine. Hence the design goal was to appoint the parameters of the proposed TVD which could radically reduce the resonances at frequency range of 490 – 500 Hz.

Experimental part consisted of 3 phases: 1st phase is balancing of TVD on Balancing Machine, 2nd phase is determination of own natural frequencies of TVD on special test machines, and during 3rd phase the determination of Peak-to-Peak Twist Angles vs. running frequencies on real engine is analyzed. First 2 phases are routine only. The balancing of new TVD is needed to adjust the unbalanced forces within industrial requirements. The 2nd phase is used for adjusting the proper lowest natural frequency of TVD.

Herein the results of 3rd phase are presented which were mostly done in Federal Budget-Funded Research Institute for Machine Science RAS (IMASH RAS).
First of all the performance of regular TVD was analyzed. The measured experimentally Peak-to-Peak Twist Angles are presented in Figure 12 (regular Torsional Vibration Damper is installed). In the Figure 12 on abscissa axis the frequencies in **rpm** are shown, and on ordinate axis the of torsional peak-to-peak angle vibration values are shown. The operation range of **rpm** for this engine is 700 – 5,000.
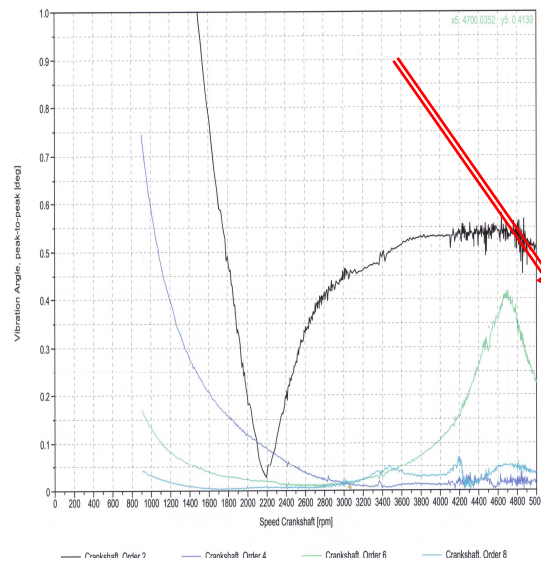
Figure 12. Recorded Peak-to-Peak Twist Angles. Regular TVD

The concerned resonance is recorded on this graph for 6th order torsional vibrations @ 4700 **rpm** with peak – to – peak angle of 0.41 degrees. It means that actual crankshaft torsional vibrations with 0.41 degrees (despite of presence of regular Torsional Vibration Damper) are at ω = 4700 (rpm) x 6 (order) x π/30 = 2951.6 1/sec = 470 Hz. Taking into an account that 4700 **rpm** is locating close to border number of 5000 **rpm** the problem with resonance at higher frequency could be not taking into account.

A variety of samples of Tuned Torsional Vibration Damper prototypes were fabricated and tested. Photos of three prototypes involved at present study are shown in Figure 13. The idea of different design for each prototype was to understand the impact of invented components on dynamic and effectiveness (separately or in combination).

Figure 13. Prototypes of different versions of Tuned TVD

The results of tests are presented in graphical form on Figures 14 – 16. On these Figures on abscissa axis is standing for the frequencies in **rpm**, and on ordinate axis is for the of torsional peak-to-peak angle vibration values.

There is a dynamic of proposed Torsional Vibration Damper having simplified structure focusing on the increased friction factor presented in Figure 14. The photo of this Torsional Vibration Damper is shown in the left side in Figure 13. There are 6th order torsional vibrations @ 4675 RPM with peak – to – peak angle of 0.30 degrees. It is lower than industrial requirement of 0.4 degrees with strong safety factor (75% of required level).
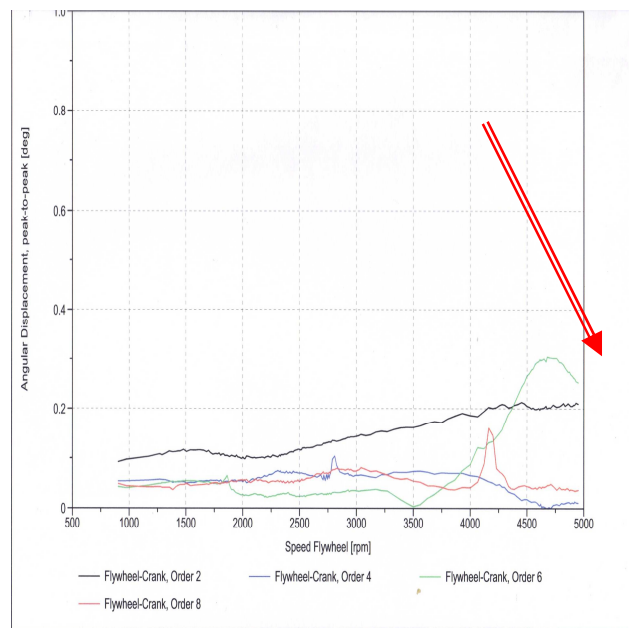
Figure 14. Recorded Peak-to-Peak Twist Angles. New TVD, Basic Model.

The weight of this Torsional Vibration Damper is less than a regular one, and the cost of manufacturing is equal to regular TVD.
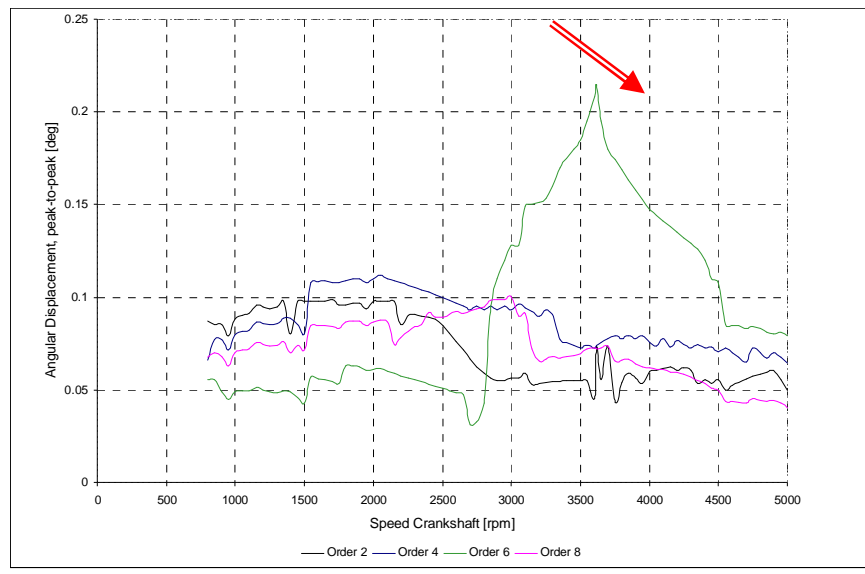

Figure 15. Recorded Peak-to-Peak Twist Angles. Semi Controlled TVD.

There is a dynamical performance of Semi Controlled Torsional Vibration Damper having mechanical control loop structure presented in Figure 15. The photo of this Torsional Vibration Damper is shown in the middle of Figure 13. There are 6th order torsional vibrations @ 3615 RPM with peak – to – peak angle of 0.214 degrees. It is significantly lower than industrial requirement of 0.4 degrees with very strong safety factor (53.5% of required level). The weight of this Torsional Vibration Damper is equivalent to a regular TVD, and the cost of manufacturing is on 5% more.

There is a dynamical behavior of Double Controlled Torsional Vibration Damper (completed with combined mechanical control loop and electronically controlled electromagnetic subsystem) presented in Figure 16. The photo of this Torsional Vibration Damper is shown in the right side of Figure 13. There are 6th order torsional vibrations @ 3150 RPM with peak – to – peak angle of 0.15 degrees. It is radically lower than industrial requirement of 0.4 degrees with very significant safety

factor (37.5% of required level). The weight of this Torsional Vibration Damper is equivalent to a regular used one, and the cost of manufacturing is on 10 % more. Actually this prototype has a design which includes a combination of Torsional Vibration Damper with Pulley (it is for some industrial applications).
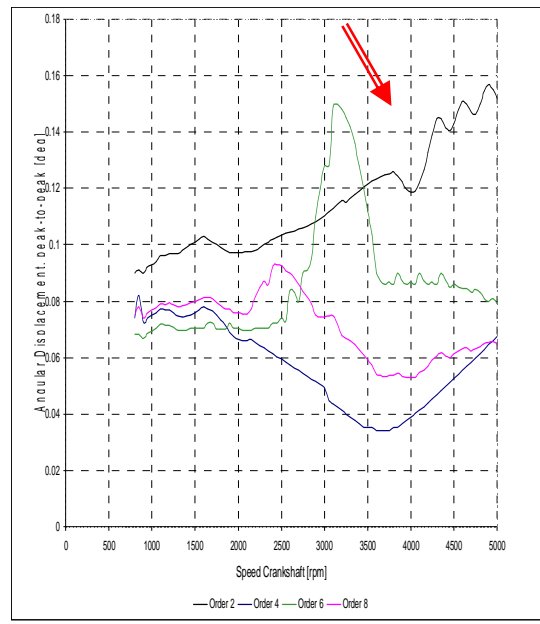

Figure 16. Recorded Peak-to-Peak Twist Angles. Double Controlled TVD.

Particularities of all tested prototypes are described in detail in [1]. Presented results of experimental research could be interpreted in the following way. In ordinary situation (when no special dynamic requirements and routine engine performance are needed) it would be recommended to replace the existing Torsional Dampers by Simplified Torsional Vibration Damper shown in the left side in Figure 13. It would be recommended to replace the existing Torsional Dampers by Semi Controlled Torsional Vibration Damper shown in the middle of Figure 13 if exploitation required the reduced torsional vibrations and improved dynamic engine performance. It is necessary highlight the additional advantage of proposed Torsional Vibration Damper in comparison to Viscous Torsional Vibration Damper – there is no fluid (so it is no need to arrange special protection from leakage) in proposed Torsional Vibration Damper. When exploitation required the lowest values of torsional vibrations and perfect dynamic engine performance (e.g. for VIP luxury vehicles), it would be recommended to replace the existing Torsional Dampers by Double Controlled Torsional Vibration Damper shown in the right side of Figure 13.

CONCLUSIONS

1. Usage of proposed Tuned Torsional Vibration Damper with instantaneous frequencies tuner gives the effective solution to get damping optimization for operating engines. Implementation of new TVD allows increasing effectiveness of using TVD by improving radically the dynamical performance of crankshaft operation, and obtaining the significantly minimal amplitudes (peak-to-peak angles 0.15 degrees) of torsional vibrations.
2. Proposed Torsional Vibration Damper with instantaneous frequencies tuner has practically the same weight and dimensional parameters as the regular Torsional Vibration Damper.

3. The less values of peak-to-peak torsional vibration angles lead to less loads on crankshaft and bearings, reduce the wearing and fatigue problems, reduce the fuel consumption, and increase the lifetime of engine.

REFERENCES

1. George Nerubenko. Torsional Vibration Damper of a Rotating Shaft. US Patent 7,438,165, Oct 21, 2008
2. Babitsky V.I., Krupenin V.L. Vibration of Strongly Nonlinear Discontinuous Systems. *//* Springer 2001, 330 p.