

Critical speed of propshaft and effect of forced synchroniztion in transmission of straight-four engine vehicles

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Abstract. The article shows the role of precession of a propshaft in increasing its vibration ativity. An auto inline four-cylinder engine has been shown to trigger a resonance mode of its drive shaft vibrations, i.e. intense flexural vibrations, leading to an emergency situation. This is due to the effect of forced synchronization (frequency capture) when the frequency of the second order unbalanced inertia forces is approaching the critical frequency of drive shaft rotations.

1.Introduction

Unbalance degree reduction of the different machine rotors is the primary task for solving the problem of their increased vibration activity, for which reason the most accurate possible definition of their unbalance parameters is necessary. This concerns both rigid rotors operating at the speeds, which are lower than the critical ones, and flexible rotors, operating at the overcritical speeds [1, 2]. However, when solving the problem of unbalance parameters identification, calculation results are often inconsistent with the experimental data and 'slight errors in rotor vibration measurements lead to big errors in unbalance parameters definition' [3]. Unbalance of a rotating propshaft is also one of the main causes of the emergence of vibrations, affecting its supports and being transferred to the units (aggregates) connected by it and the vehicle design as a whole. Upon that, it is important to keep in mind that the rotating propshaft carries out complicated motion, which can be represented as a superposition of axial rotation and precession. Ways of solving the problem of reduction of the different machine rotors unbalance are well-known. Theory and physics of the phenomenon related to it, including those related to the precession motion, are sufficiently completely considered in the rotor dynamics literature [4 - 9]. In existing literature, quite insufficient attention is paid to the precession motion problem in relation to the propshafts of transmissions of automobiles and other vehicles. In this literature, the consideration of the rotation motion dynamics is limited by the critical speed calculation. Critical speed is an important parameter characterizing the behavior of the propshaft under the operating conditions, but its vibration activity performance is largely defined by the precession motion, rather than self-rotation, which is usually left out of discussion of calculation methods of the resonant mode parameters. What is important is not only defining the critical speed of the propshaft, but also exploring the nature of the problem and showing a movement from the phenomenon to the crux of the problem. Speed corresponding to the sharp increase in the amplitude of vibrations excited by the propshaft unbalance is taken as a critical speed. This phenomenon is usually explained as a resonant increase in the vibration amplitude at the disturbing forces action frequency caused by the rotating shaft unbalance, which coincides with the own speed of the flat (planar) bending vibrations of the non-rotating shaft, which is logically inconsistent, whereas effects detected at the rotation motion are deduced from the conditions obtained for the non-rotating system. An approach to the explanation of the rotating shaft behavior from the perspective of the consideration of the time behavior of the small disturbances of its established



motion, which is a rotation with a constant angular speed, seems more logical than from the perspective of the analysis of its bending vibrations in relation to static balance position. This more precisely reflects the phenomenon and is especially important in the analysis of the behavior of the considered object during the rotation at overcritical speeds [5]. For the effects analysis, that may appear at the propshaft rotation, let us consider two cases: a) an ideal case, which is the rotation of a balanced propshaft; b) rotation of an unbalanced propshaft.

2. Rotation of a balanced propshaft

The rotating rotor will not cause the alternating disturbing influence on its supports, if the rotation axis coincides with one of its main axes of inertia [1, 2]. Upon that, a mass fixed on a rigid simple (two-point) beam is traditionally taken as a calculation model of such structures, i.e. the rotor rotation axis deflection caused by weight load is ignored. In order to ignore this phenomenon, in certain studies the rotor rotation is considered at its vertical configuration. The rotor model with a vertical configuration of the rotation axis is applied in order to simplify the solution of its dynamics problems [8, 10]. However, the propshaft is not usually vertically installed in vehicle transmission and has a deflection caused by the action of weight load. Upon that, the propshaft rotates about the obtained bending line. As the propshaft mass center and the centers of sections of its elementary weights are on the bending line, the balanced two-point propshaft with a ring cross-section pipe shall not cause vibrations at self-rotation.

3. Rotation of an unbalanced propshaft

During rotation of an unbalanced propshaft, there is a centrifugal force $R = m\omega_{sq}^2 r$, where m is the unbalanced mass; r is the radius of the location of the unbalanced mass; ω_{sq} is the propshaft speed. The centrifugal force and weight load during rotation of an elastic propshaft, i.e. a structure having the finite stiffness in a plane of bending, result in change of its kinematic behavior. And upon that, rotation of the propshaft around the bending line caused by the weight load with angular speed ω_{sq} is supplemented by precessional motion – rotation of the propshaft bending plane with angular speed Ω , which is called precession rate. Such conclusion is consistent with the results of analysis of precession motion of a rotor in a form of massless shaft with a disk fitted on it, while 'the precession axis turns out to be not the bearing axis, but a curve defining the shaft position under static (in this case – weight) load' [9]. Thus, the weight load and unbalance of the propshaft cause precession of its axis and, consequently, vibrations transmitted to the attached units (aggregates) through the supports of the rotating shaft. The first works (1869) on rotor dynamics connected with the 'rotor precession' phenomenon refer to the name of W. Rankine, who called it 'centrifugal whirling' [8]. A pin-supported rotating propshaft is some kind of gyro system that reminds a rate gyroscope. One of the degrees of freedom of such gyro is formed by its own rotation around the curved axis with angular speed ω_{gyr} , and the other – by rotation of the bending plane with angular speed Ω . In mechanics, a gyroscope is a rotating solid body of any shape; upon that, its speed is not a characteristic feature [11]. The problem of motion of such body 'in general is still not solved and, in a certain sense, unsolvable' [6]. To predict the behavior of gyro systems the conservation of angular (impulse) momentum law – the fundamental law of physics – is used, according to which

$$\frac{dL}{dt} = M, \quad (1)$$

where $L = \omega_{gyr} I$ is the impulse moment (kinetic moment or angular moment), I is the moment of inertia of a rotating body. If moment M of external forces applied to the gyroscope equals zero, then angular moment $L = const$, and the gyroscope rotation axis will maintain certain position in space. If the moment of external forces is not zero, then the gyroscope will move while one of its points will remain fixed. In this case, the gyroscope motion can be considered as rotation around an instantaneous

axis that goes through a fixed point. For the rotating propshaft, such points can be its pin-bearing supports. The interrelation of angular speeds of precession Ω , rotation about own (figure) axis ω_{gyr} , external moment M and moment of inertia of a rotating body I according to the approximate theory of gyro systems is expressed as follows:

$$\Omega = \frac{M}{\omega_{gyr} I} . \quad (2)$$

It should be noticed, that the precession of a rotating shaft can be both direct and reverse depending on the direction of angular speeds of the shaft itself and its precessional motion, which are defined, for example, by spacial characteristics of elasticity of the shaft supports, including their anisotropy or isotropy [13, 14]. Upon that, in general, precession frequency of a rotating shaft Ω depending on speed ω_{gyr} of the shaft itself is defined through coefficient s which can be an integer or a fraction, positive or negative, i.e. $\Omega = s\omega_{gyr}$ [7]. However, inertia forces of unbalanced masses of the most common rotor structures, including vehicle propshafts, generate forced vibrations of these structures in a form of synchronous precession with a frequency equal to their speed [2, 7] and for them $s = 1$ and $\Omega = \omega_{gyr}$. These forces are proportional to the square of the rotor speed. The source of disturbance can be the rotor weight and any other unidirectional load as well. In the educational literature, motion analysis of statically unbalanced rotors is often limited to the consideration of direct synchronous precession. In [9] the physics and conditions of excitation of direct (forward) and backward rotor precession are considered. The backward recession of unsupported rotating propshaft in certain conditions of its excitation has been registered during tests by means of photodiode sensors [15].

4. Synchronization of the frequency of action of unbalanced second order inertia forces of the engine and the critical speed of the propshaft

The area of dangerous self-excitation of bending vibrations of the propshafts can be rather precisely defined within the procedure, whose description is provided in educational literature [16, 17], in standard [18]. However, solution of this problem is not exhaustive. For example, such a solution cannot be called sufficient for the case of in-vehicle application of widespread in-line four-cylinder engines. It is connected with the fact that, as known [19], crankshaft rotation in operation of the in-line four-cylinder four-cycle engine with angular speed ω is followed by the action of unbalanced second order inertia forces with frequency 2ω . These forces are transferred to the unit (aggregate) supports (mounts) and the elements of dynamic system 'engine crankshaft – clutch – gearbox – propshaft' elastically attached to the flexible frame, i.e. with the common base. At coincidence of frequency 2ω of action of unbalanced second order inertia forces of the engine and the critical speed of the propshaft ω_{cr} , the latter enters the mode of resonance of the bending vibrations characterized by the sharp increase in amplitudes of the bending vibrations. Thus, if frequency 2ω of action of unbalanced second order inertia forces on the engine mounts approaches the critical speed of bending vibrations of the propshaft ω_{cr} connected via the gearbox and the clutch with the crankshaft rotating with frequency ω , there is forced synchronization of vibrations, frequency capture. Upon that, the propshaft is exposed to intensive bending vibrations with frequency 2ω of the synchronizing system, as it begins to operate in the resonance area that can result in the destruction of the propshaft [20-22]. Therefore, the critical frequency (speed) ω^{sync} corresponding to the forced synchronization of vibrations can be determined as follows:

$$\omega = \omega^{sync} = 0,5\omega_{cr} . \quad (3)$$

Here ω_{cr} - the propshaft critical speed (according to the first form).

5. Figure

Figure 1 shows the scheme of action of unbalanced second order inertia forces of to-and-fro moving masses of the engine on its mounts. This figure shows as follows: 1 – engine, 2 – an elastic element of engine mounting

(suspension), 3 – basis of the engine mount, 4 – a rotation axis of unbalanced mass, 5 – the vibrator housing, 6 – a core, 7 – vibrator unbalanced mass. The vibrator is schematically depicted on the figure in the form of two equal unbalanced masses symmetrically located in relation to the vertical axis and rotating in opposite directions (in antiphase) with frequency 2ω . Upon that, the horizontal component of centrifugal forces at every instant is equal to zero, and only the vertical component of centrifugal forces is transferred to the mounts [23].

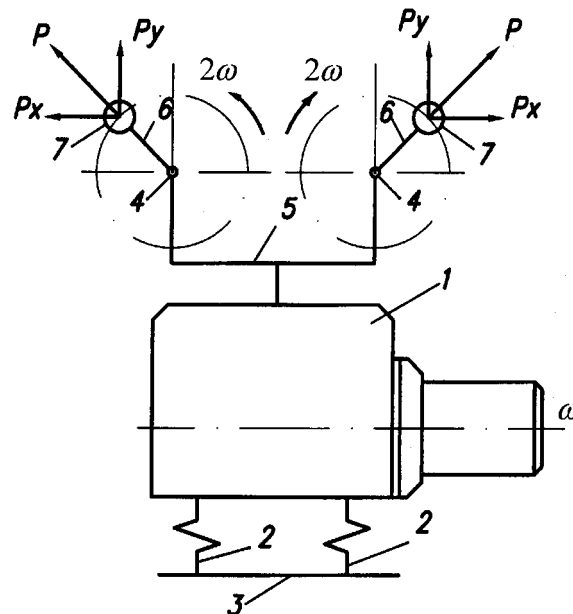


Figure 1 The scheme of effect of unbalanced second order inertia forces of to-and-fro moving masses of the engine on its mounts

6. Conclusion

6.1 The propshaft is one of the significant sources of the vibrations transmitted from the transmission to the vehicle design as a whole. In the process of its operation, it carries out complicated motion, which can be represented as a superposition of its axial rotation and precession.

6.2 The current state of the rotor dynamic processes theory allows making a conclusion that a rather precise reflection of the physics of increase in vibration activity of the propshaft rotating in the vehicle transmission is possible from the perspective of its precession motion analysis.

6.3 In case of in-vehicle installation of in-line four-cylinder engines it is evident that the effect of forced synchronization (frequency capture) may lead to the following: if the frequency of action of the unbalanced second order inertia forces of the engine approaches the critical propshaft speed, these propshafts will enter the resonant vibration mode and will be exposed to the intensive bending vibrations making an emergency possible. Upon that, to ensure safety during operation under the operating conditions, determination of the critical propshaft speed margin is necessary but insufficient. In this case, the additional condition is taking into account the oscillating system of the unbalanced second order inertia forces of the engine.

7. References

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