Impact of Counterbalance Mass on Torsional Vibrations of Crankshaft

Asllan Hajderi*, Drakuli Lumi

Department of Mechanic and Transport , 'Aleksander Moisiu' University, Durres, District Nr. 1, Street : 'Rruga e Currilave' Durres, Albania; *ashajderi@yahoo.com

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Abstract. In the present study it is shown how to change the main indicators of vibrations of crankshaft, in case of changing its masses. In the study it was taken the crankshaft of diesel engine with 4 cylinders in line with five main journals. To study the torsional vibration of the crankshaft system, the first it is carried the construction of equivalent reduced scheme with five discs connected to four flexible shafts. Results show that the reduced inertia moments of disks for the case with counterbalances grow to 24%. Free frequency and vibration forms, is calculated by using Holxer-Tole method. Results show that the increase of counterbalance mass leads in the reduction of vibration frequency. For the crankshaft with counterbalance the vibration form does not change, but the forced vibration amplitudes increase over 11%. Most charge part in torsion, remains the fifth shaft equally as the crankshaft without counterbalance.

Introduction

Torsional vibrations of the crankshaft are connected with the lateral system in the first set and at the end with the transmission elements of the power system. From studies it results that for the crankshaft system of tractor engines, the most dangerous form of vibration is vibration forms with four nodes, with the first node near flywheel, which is equal to the free vibration of the crankshaft with open clutch with a node near flywheel [9].

The study was obtained crankshaft of tractor diesel engine with four cylinders in line produced cast iron (Figure 1), which consists from five main journals and eight pages, forming a crankshaft with cranks in a plan.

In this crankshaft due to high loads on the main journals, it occurs a large and irregular consumption of main bearings and journals, which causes a large increase of flexible moments on pages four and five.

For a space 0,15 mm of the first main bearing and journal, which is almost the same as the consumption of middle main bearings and journals, flexible moments in fourth and fifth pages increase over 5 times until they cause his breaking in these pages, which is confirmed by the practice of using engines 75D [4]. To eliminate this phenomenon, on the basis of relevant dynamic study, it has resulted that the introduction of four counterbalances (each with 3.3 kg), its main journals gain a greater reduction of acting forces on this. In the middle main bearings and journals we have a reduction 40% of average radial load, or 13540 N [4].

By placing a counterbalances it is achieved a monotonous consumption of main journals and bearings crankshaft, which leads to reduction of flexible moments and in increasing of crankshaft life.

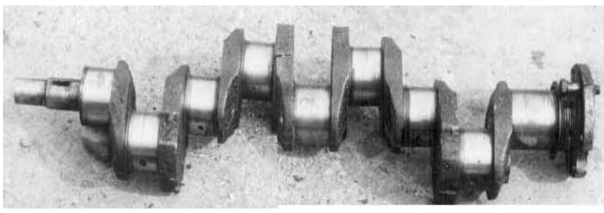

Figure 1: The crankshaft of tractor 75 D.

Setting of counterbalances brings a change of rotational mass, which affects in the loading state of the crankshaft from torsional vibration [1].

Vibration study of crankshaft system is associated with the movement of the piston and connecting rod group. For these complex systems, it is formed an equivalent system, which has the potential and kinetic energy equal to the real system and then are calculated the frequencies, critical speeds and resonance vibration amplitudes.

1 Construction of the Equivalent System

Equivalent system for calculating crankshaft vibrations, or reduced scheme consists of five discs with mass, connected to four elastic shafts without mass (Figure 3). Construction of equivalent system consists of [3],[7]:

- Determination of reduced inertia moments of disks
- Determination of reduced rigidities of connecting elastic shaft.

For the crankshaft without counterbalances, from [5] reduced inertia moments values of discs will be:

$$I_1 = 0.101,\ I_2 = 0.101,\ I_3 = 0.101,\ I_4 = 0.101,$$
$$I_5 = 2.83\ [kg\ m^2]$$

While for the crankshaft with counterbalances (located on the opposite side of pages), inertia moment of counterbalance for given construction in Figure 2 will calculate:

$$J_{cp} = \rho b 2\alpha \int_{r_1}^{r_2} r^3\, dr = \rho b 2\alpha \frac{r_2^4 - r_1^4}{4} \qquad (1)$$

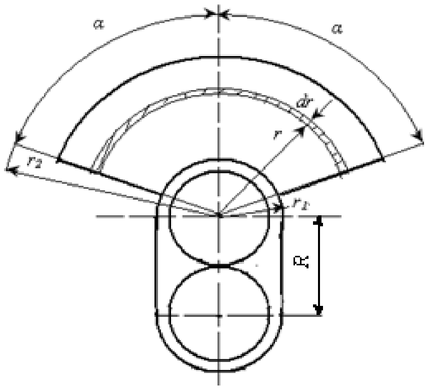

Figure 2: Schema of counterbalance.

or

$$J_{cp} = m_{cp} \frac{r_2^2 + r_1^2}{2} \qquad (2)$$

where :

$$m_{cp} = \rho\, b\, \alpha\, (r_2^2 - r_1^2) \qquad (3)$$

There mass of counterbalance, located in every crank is different and creates the add of inertia moment, $\Delta J = 10\%, 20\%, 24\%$, etc.

For counterbalance dimensions, width $b = 30\ mm$, $r_1 = 62\ mm$, $r_2 = 130\ mm$, $\alpha = 65°$, we take:

$$m_{cp} = 3.3\ kg, J_{cp} = 0.024\ kgm^2\ (\Delta J = 24\ \%),$$

So, for the crankshaft with counterbalances, reduced inertia moments values of discs will be:

$$I_1 = 0.125, I_2 = 0.125, I_3 = 0.125, I_4 = 0.125,$$
$$I_5 = 2.83\ [\ kg\ m^2]$$

The reduced rigidity of elastic shafts from [5], for crankshaft by cast iron is $C = 1592356\ [\ Nm/rad]$

In this case reduced scheme to calculate the vibration of crankshaft is shown in Figure 3 (Values in parentheses are inertia moments for the crankshaft with counterbalance mass 3.3 kg).

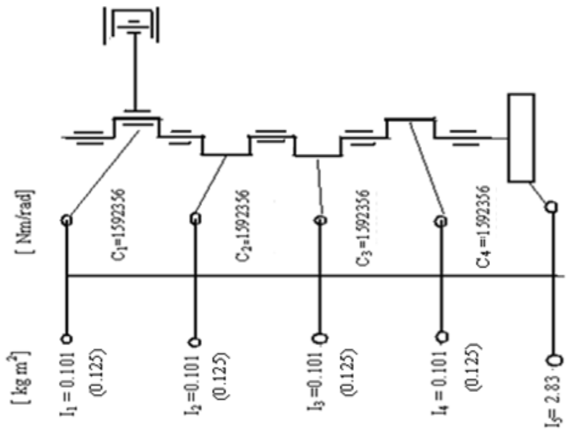

Figure 3: Reduced scheme to calculate crankshaft vibrations of counterbalance.

2 Frequencies and Free Vibration Forms of Crankshaft

To determine the frequency and free vibration forms, it is used the method without forming of differential equations of vibration. We have chosen the method Tole-Holxer [3][7], which is based on the principle of D'Alambert, that during the free vibration of system, the sum of moments of elasticity forces and inertia forces must be equal to zero. In this case the equations system of free vibration becomes a system of algebraic equations, which given [7]:

$$M(i) = M(i-1) + J(i)a(i)\omega^2$$
$$a(i) = a(i) - \frac{M(i)}{C(i)} \qquad (4)$$

where:

$M\ (i)$ - is the moment of the elasticity forces

$a\ (i)$ - is the relative amplitude of free vibrations of the disc,

ω – is the frequency of free vibrations

Under this method it is formed function :

$$R = M(n-1) + J(n)a(n)\omega^2 \qquad (5)$$

Values of ω, for which function R become to zero, are free frequencies. For this it is used the iterative method, where the cord function interrupts the abscissa axis.
This method provides a clear statement on the substance of the made calculations, directly gives the frequency and relative amplitudes of vibration and the algorithm (program) is simple.

So, the frequencies and free vibration forms were calculated according to Holxer-Tole method. Below we have analyzed only the two first frequencies, because they can operate in the area of engine speeds and the results of the calculations for various cases of disk inertia moments are shown in Table 1.

Inertia moment [kg m^2]	The first free frequency, ω_1	Relative amplitudes	The second free frequency, ω_2
J=0.091 (-10%)	1542	1; 0.865; 0.613; 0.274; -0.096	4227
J= 0.101	1471.3	1, 0.863; 0.607; 0.268; -0.098	4013,5
J=0.11 (10%)	1410	1; 0.861; 0.602; 0.260; -0.1	3829
J= 0.121 (20%)	1357	1; 0.859; 0.598; 0.253; -0.115	3653
J=0.125 (24 %)	1338.4	1; 0.859; 0.597; 0.251; -0.12	3609.5
J=0.131 (30%)	1311	1; 0858; 0.594; 0246; -0.125	3564
J=0.139 (38%)	1278	1; 0857; 0591; 0241; -0.131	3526

Table 1: Two first frequencies and relative amplitudes.

The reduction of inertia moment can be achieved by reducing the mass of the piston and connecting rod group. The change of frequency depending on the change of inertia moment is shown in figure 4.

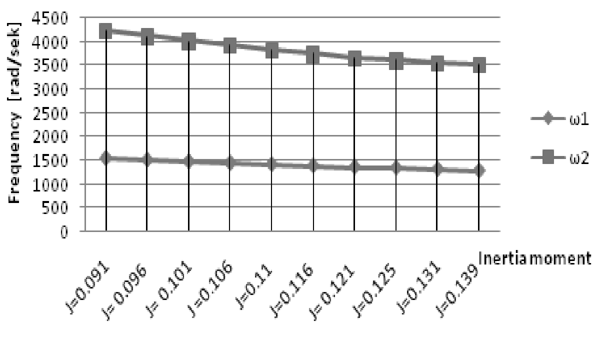

Figure 4: Change of the vibration frequency by increase of inertia moment.

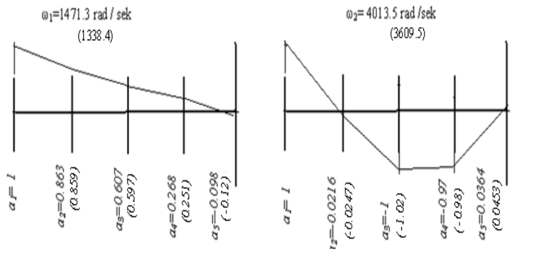

Figure 5: Two forms of free vibration of the crankshaft system.

The vibration forms for the first frequencies for crankshaft without and with counterbalance are given in Figure 5.

3 Critical Speeds of Engine

The development intensity of the torsional vibration system is determined by the value and character of the excitation moment. In this case it consists of the moment of inertia forces and gas pressure forces.
Harmonic analysis of inertia force by the approximation of Den Hartog [2] can be expressed in four harmonics:

$$T_j = m_j R\omega^2(\lambda\sin(\omega t)/4 - \sin(2\omega t)/2 - \\ -3\,\lambda\sin(3\omega t)/4 - \lambda 2\sin(4\omega t)/4\,) \qquad (6)$$

While the inertia force moment is calculated:

$$M_j = T_j R \qquad (7)$$

Harmonic analysis of the gas pressure forces moment is performed based on the principle of linear superposition, which means that vibrations caused by non harmonic moment can be seen as the sum of harmonic vibration caused by specific harmonic components of the excitation moment [7].

$$M_g(t) = M_0 + \sum_{k=1}^{n} M_{gk}\, sin(k\omega t + \alpha_k\,) \qquad (8)$$

There:

M_0 - average torque
M_{gk} - amplitude of order harmonic k
ω - angular speed of the crankshaft
$k\omega$ – frequency of order harmonic k
α_k – initial phase of order harmonic k

The amplitude of excitation moment depend on the indicatorial pressure of the engine. The average effective pressure for a speed regime of engine n_x given [9]:

$$p_{ex} = p\,(\,C_1 + C_2\frac{nx}{n} - \left(\frac{n_x}{n}\right)^2\,) \qquad (9)$$

where

$C_1 = 0.6,\ C_2 = 1.4$ (for diesel engine antechamber).

Recognizing that friction losses are constant, the pressure losses from friction are calculated:

$$P_{fx} = p_i(1-\eta) \quad (10)$$

Where η = 0.8 is the mechanical efficiency of the engine (for diesel engine without distention η = 0.7- 0.82).

Thus the ratio of the change of harmonic amplitudes of gases force, for different speed regimes will be given:

$$\frac{P_{ix}}{p_i} = 1 + \left(C_1 + C_2\frac{n_x}{n} - \left(\frac{n_x}{n}\right)^2 - 1\right) \quad (11)$$

The dangerous rate of a resonance regime depends on the work carried out by the respective harmonics of the excitation moment. Performed work by harmonic order k, for a vibration it is calculated [10]:

$$W = \pi M_k A_{k1} \sum_1^n \vec{a}\, ki \quad (12)$$

where A_{k1} is the vibration amplitude of the first disc under the action of harmonic k and $\sum_1^n \vec{a}_{ki}$ - the vector sum of the relative amplitude of free vibration, is depending on the position. To assess the most dangerous harmonics in real work, it is used relative work

$$w_r = \pi M_k \sum_1^n \vec{a}_{ki} \quad (13)$$

The resonance phenomenon occurs when one of the frequencies of the excitation harmonics becomes equal with the frequency of free vibration. Thus resonance speeds caused by i-the frequency will be determined [7]

$$n_{ik} = \frac{n_f}{k} \quad (14)$$

where $n_f = \frac{30\,\omega_i}{\pi}$, $k = 0.5, 1, 1.5, 2, ...$- excitation harmonic order for engine with four time.

K	9	9.5	10	10.5	11	11.5	12
W_r	23.99	32.66	154.73	30.97	25.43	28.67	140.37

Table 2: Relative works of harmonics.

In working speeds $n < 1500\, rpm$, free vibration of crankshaft on first frequency $\omega_1 = 1471.3\, rad/sec$ (1338.4), are excited by the harmonic of order over 9. Second frequency ω_2 is above from work speeds. Excitation power falls by increasing of the order of the harmonics, so it is used up to 24 harmonics.

Most dangerous critical speeds are determined by the relative works of these harmonics which are given in Table 2.

From Table 2 it shows that the most dangerous critical speeds are harmonics of order 10 and 12. So most dangerous critical speeds n_{10}, n_{12} depending on the change of inertia moments are given in Table 3

ΔJ	-10%	0	+10%	20%	+24%	30%
n_{10} (rpm)	1473	1405	1347	1303	1278	1253
n_{12} (rpm)	1227	1170	1127	1087	1065	1044

Table 3: Most dangerous critical speeds n_{10}, n_{12}.

Vibration amplitudes

Loading condition in torsion of the crankshaft is determined by the level of forced vibration amplitudes. The chosen method for determining of the amplitude is energy method, which is used for regimes within and outside the resonance zone [10]

According to the energy method the forced vibration amplitudes in the first disc, excited by harmonic of order k, is defined:

$$A_{k1} = A_{k0}\beta \quad (15)$$

there A_{k0} is the balance amplitude proposed by Ker Willson [10], which is calculated by:

$$A_{k0} = \frac{M_k \sum_1^n \vec{a}_{ki}}{\omega_i^2 \sum_1^n J_i a_i^2}, \quad (16)$$

and β is the dynamic factor, which is calculated by:

$$\beta = \frac{1}{1-\left(\frac{\omega_k}{\omega_i}\right)}, \quad (17)$$

ω_k - frequency of exciting harmonic of order k $(\omega_k = k\omega)$.

For resonance areas ($0.9 < k\ \omega/\omega_i < 1.1$), forced vibration form is very near with free vibration. Discrepancies grow by increasing the resistance forces [10]. The resistance coefficient is taken the same for all cylinders and is calculated:

$$\zeta = \zeta' \, Fp\, R^2 \quad (18)$$

There ζ' - specific resistance coefficient (for diesel engine given 0.04-0.05 dN sec/cm^3), Fp – the surface of piston

Finally the real vibration amplitude of the first disc caused by excitative harmonic of order k is calculated:

$$A^r_{k1} = A_{k1}/\zeta, \quad (19)$$

and other amplitudes

$$A^r_{ki} = A^r_{k1}\, a_i \quad (20)$$

The values of the resonance vibration amplitudes of the first disc calculated for the critical speeds caused by harmonics of order 10 and 12, by changing the inertia moments of discs are given in Table 4.

Change ΔJ	-10%	0	+10%	+20%	+24%	+30%
$A_{10,1}$x10^{-5} (rad)	369	390	405	425	434	444
$A_{12,1}$x10^{-5} (rad)	314	337	354	374	384	395

Table 4: Each Resonance vibration amplitudes of the first crank.

While the impact of the inertia moment on critical speeds and resonance vibration amplitudes in the first disc are shown in Figure 6.

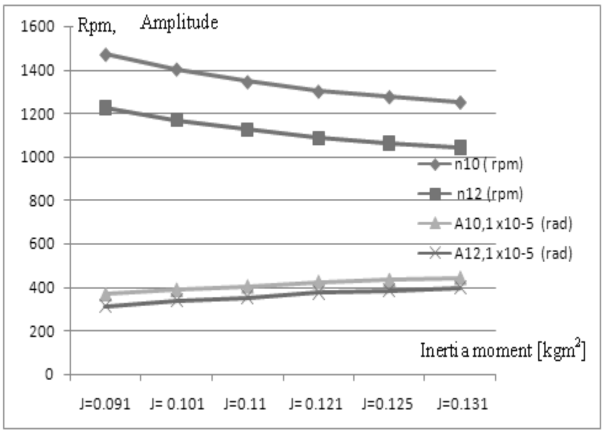

Figure 6: Critical speeds and resonance vibration amplitudes.

The vibration amplitudes of the first crank of crankshaft without and with counterbalance (ΔJ = 24%) calculated in the resonance area and outside its are given in Table 5.

4 Discussion

The construction of the equivalent system for study of crankshaft vibration has accepted a approximation of the value of the inertia moment, therefore the values of frequencies and vibration amplitudes calculated will be approximate. Holxer-Tole method is a simple method that gives accurate results in determining the frequency and relative amplitude of crankshaft vibration.

The obtained results from the calculations given in Figure 5 shows that the free vibration frequencies of crankshaft, by increasing of the inertia moments are reduced almost by linear law. While the form and relative amplitudes of vibration have very small changes.

For the crankshaft with four counterweights, the frequencies are about 10% smaller than those for the crankshaft without counterweights.

Crankshaft	without counterbalance			with counterbalance		
n (rpm)	1450	1405	1300	1300	1278	1200
$A_{10,1}$ x10^{-5} (rad)	0.248	390	0.108	0.39	434	0.13
n (rpm)	1200	1170	1100	1100	1065	1000
$A_{12,1}$ x10^{-5} (rad)	0.164	337	0.078	0.33	384	0.12

Table 5: Vibration amplitudes of the first crank of crankshaft.

The change of vibration relative amplitude of discs for the first frequency is small and the more charged sector remains the node of fifth shaft. By placing of counterweights, the change is not sensitive. The change of the relative amplitude for the second frequency is greater and the more charged sectors remain too nodes of shafts 1 and 4, but these are irrelevant, because the vibration with second frequency are only free.

Comparing the results shows that the variant with the counterweights (3.3 kg), will have a increase of inertia moments of discs up to 24%, which decreases the first frequency up to 10%.

The results given in Table 2 show that the most dangerous critical speeds are those caused by the excitations of order harmonics 10 and 12. The values of these speeds decreased with increasing of inertia moments. For the crankshaft with counterweights ($\Delta J = 24\%$) critical speeds are reduced by 10%. Increasing the inertia moment over 30% creates the risk from the impact of the order harmonic 8 of excitation moment.

The results given in Table 4 and Figure 6 shows that the resonance vibration amplitudes increase by linear law, with the increase of inertia moments. The resonance vibration amplitudes of order 12 are reduced more, than those of order 10. This shows that the dangerous torsional vibration in critical regimes decreases with increasing the order of excitation harmonics and this confirms what it is given in [10], that for vibration calculation it is enough up to 24 harmonics. Results show that for the crankshaft with counterweights, the resonance vibration amplitudes of the first crank increase 14% compared with the crankshaft without counterweights.

The results given in Table 5 shows that the vibration amplitudes of the crankshaft system away resonance regime are negligible compared with the resonance regimes, where the vibration amplitudes increase over 2000 times for order harmonic 10 and over 1500 times for order harmonic 12. This shows that during study of torsional vibration of crankshaft should be calculated only vibration amplitudes in the critical speeds, corresponding to the resonance regimes. For the crankshaft with the counterweights the vibration amplitudes away resonance regime excited by order harmonic 10, increase 2 time, and excited by order harmonic 12, increase 1.4 times. While vibration amplitudes in critical speed $n_{10} = 1278\,rpm$ created by order harmonic 10 increase 11%, and in critical speed $n_{12} = 1065\,rpm$ created by order harmonic 12, increase 14%. So, vibration amplitude in critical speed regimes are important for solidity of the crankshafts and should be performed calculation of crankshaft vibration, if their mass change [8].

The increase of the counterweight mass, that creates the add of inertia moment over 30% creates the possibility of large increase of vibration amplitude, due to the introduction of smaller excitation harmonics, which are dangerous. So, increasing of the mass over 3.3 kg leads to increasing the loading state of the crankshaft and this makes, that its solidity results uncertain.

An effective intervention that reduces the dynamic tension of torsional vibration is the introduction of flexible joints in the crankshaft system, which creates the change of frequencies and free vibration forms.

5 Conclusions

Increasing of the counterweight mass placed in crankshaft, leads in the reduction of free vibration frequencies of crankshaft, while the vibration form and the relative amplitudes remain the same level. Increase of inertia moment 24% reduce the first frequency 10%. Part more charged remains fifth shaft.

Increasing of the counterweight mass leads in reducing of engine critical speed values and can create a excitation from lower harmonics, which are more dangerous. For crankshaft with counterweight critical speeds reduce with the same rate as the first frequency.

Forced vibration amplitudes away critical speeds are not sensitive, compared with those in critical speed regimes.Vibration amplitude in critical speed regimes are important for solidity of the crankshafts.

They grow with increase of inertia moments and if their mass change it should be performed calculation of crankshaft vibration. For the crankshaft with counterweight vibration amplitudes excited by the order harmonic 10, increase 11% and those by order harmonic 12, increase 14%.

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