General mechanisms balancing rational schemes

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Abstract. The worldwide trend of the modern machine building industry is the produced machines power density increase resulting in increasing the operation velocity while the machines mechanical systems joints and rings inertial loading grows and is comparable to the level of the main power flow transformed by the mechanical system. Inertial loading characterizes machines vibration activity. In this regard, machines balancing problem is of particular relevance. The paper provides the mechanical systems balancing technique by installing the counterweight directly in the inertial excitation generation areas.

Key-words: general mechanical system, inertial loading, balancing, counterweight, mass centre.

1. Introduction
Balancing is the machine mechanics dynamic synthesis problem in order to minimize the machine moving parts reactions to the shaft by the rational mass distribution of the moving rings and added counterweight. The general mechanisms include the cycle mechanisms implementing the movement conversion with the cycle variable transfer function.

The main source of generating the variable and most significant module reactions are the forces and inertia forces moments of the rings exhibiting non inertial motion. Inertial loading of rings is not useful, is perceived by the joints, is transmitted to the shaft (casing link) and then into the frame and various systems supporting the machine operation as well as into the control and measuring instruments, automatic control and machine operator system.

Machines power density increase is the objective development trend of the modern machine building [1] and therefore resulting in the machines efficiency and operations velocities increase that in turn leads to the increased inertial loading of the rings and joints. Accordingly, the mechanical systems balancing [2] as part of the high technology machines developing objective is relevant.

According to paper [3], the conditions, under which the perfect balancing is achieved, are known and reduced to zero equality of the mechanical system inertia forces main vector $R_{in}$ and moment $M_{in}$ i.e.

$$\begin{align*}
P_{in} &= 0 \\
M_{in} &= 0
\end{align*}$$

(1)

In general actual systems the given conditions fulfillment only by means of the basic mechanism design is impossible, therefore the counterweight addition is the forced measure of the balancing objectives realizing.

As shown in [4], the complete satisfaction of the conditions (1) leads to the machines mechanisms irrational engineering solutions, therefore the static balancing techniques, when the conditions (1) are partially performed, are the most widely used ones. An example of full static balancing is shown in figure 1. According to the above scheme, crank-slider mechanism, extending the connecting rod 3, place the corrective mass $m_{c1}$ at the point C.
The center of mass of the modified connecting rod moves to point A. Next, we constructively develop crank 2 and place the correction mass $m_{c2}$ at point D. Accordingly, its value will be as follows

$$m_{c2} = \frac{(m_4 + m_3 + m_{c1})l_{OA} + m_1l_{OS_2}}{l_{OD}}$$

(3)

Figure 1. Static balancing crank-slider mechanism.

Under conditions (3), the center of mass of the system of masses of all links moves to point O, therefore the main vector of inertia forces at any angle of rotation of the leading link 2 will be absent. But it is worth noting that the arrangement of corrective masses in the presented mechanical system in a similar manner leads to a significant increase in the working space of the operating mechanism, and to an increase in the mass of the body part, an increase in the reaction in the intermediate links, and also multiply the value of the moving part of the machine. [5].

In any case, the balancing scheme choice is estimated by meeting the conflicting quality criteria of the engineering solution.

2. Problem statement

Reduction of vibration activity of machine units in addition to static balancing is achieved by several well-known techniques: the creation of resistances on the propagation paths of vibrations, dynamic damping using additional devices - absorbers, as well as a decrease in inertial loading source in the point of its origin. We propose to calculate the values and counterweights location directly in the scheme nodes with non-inertial motion of the rings to maintain the specific node mass centre unchanged position, therefore to eliminate the variable inertial loading source in the point of its origin.

3. Theory

Figure 2 shows the slider node with mass of $m_1$ with possible referring the connecting rod mass portion 2 to $m_1$ and to the point B, and kinematic coupling of the slider movements and correction counterweight is $m_{c1}$. Kinematic coupling is implemented by using the chain gearing with gear ratio of $U=1.0$ (figure 2a). Other solutions are also possible, for example, by providing the slider and counterweight with gear racks, reverse-phase movement of which will be provided by the gear wheel engaged with the rack and mounted on the body ring axis, while kinematic coupling will be equal to $U=1.0$. 

$$m_{c1}l_{AC} = m_4l_{AB} + m_3l_{AS_2}$$

$$m_{c1} = \frac{m_4l_{AB} + m_3l_{AS_2}}{l_{AC}}$$

(2)
Figure 2. Slider inertial loading compensation schemes: a) kinematic coupling through the chain gearing; b) through the gear coupling; c) through the multiplicative coupling. Kinematic gear coupling is presented in figure 2b while the multiplicative one is shown in figure 2c. In both cases transfer coupling function is $U\neq 1.0$.

According to the diagrams in figures 2b and 2c it is possible to control the counterweight value. Thus for the gear coupling the counterweight is equal to $m_c=m_1/U$ and for the multiplicative scheme it is $m_c=m_1/U$. Furthermore, slider mass $S_1$ and counterweight $S_c$ centres (figure 3) should be placed according to these masses static moment zero equality condition referred to the point 0, i.e. the distances should be referred as follows:

$$\frac{0S_1}{0S_c} = U$$

(4)

Mass centre position of this node in motion is always in the fixed point 0 (figure 3).
Figure 3. Balanced nod mass centre position calculation.

Figure 3 presents the kinematic coupling gear variant showing that the gear cluster angular acceleration $\varepsilon$ is defined by the slider linear acceleration $a^T_B$, i.e., $\varepsilon = a^T_B/r_1$, and the counterweight linear acceleration is calculated by the following formula:

$$a_{S_{ct}} = \varepsilon \cdot r_c = a^T_B \cdot \frac{r_c}{r_1} = \frac{a^T_B}{U}$$

(5)

The ratio is the same for the linear movements of $m_1$ and $m_c$ and is defined by the equation:

$$\frac{S_1B_1}{S_cS_{ct}} = U$$

(6)

Taking into account the calculated values of $m_c$ the mechanical system mass centre occupies the unchanged position at the point 0.

In analysing the proposed scheme of the nodal balancing, it should be noted that such approach is rational one as compared to the known balancing variant by means of «connecting rod-slider» system mass centre shifting to the crank pin resulting in the joints reactions increase and other negative phenomena mentioned above.

The dynamics study of the machine provided with the proposed balancing scheme should take into consideration the counterweight energy and kinematic coupling with the slider motion when calculating the reduction link inertial characteristic.

4. Results discussion

It is necessary to mention that the proposed scheme of the general mechanical system nodal balancing appears to be expedient as it does not lead to many fold increase in weight dimension characteristics of the system balanced according to the static balancing well-known scheme and does not result in the dynamic unbalance degradation due to the connecting rod console large balancing mass.

The proposed balancing scheme engineering implementation does not require significant design complexity of the mechanical system.

We considered the option of the general mechanical system nodal balancing is considered, when slider 1 is driven by connecting rod 2 (Fig. 2), but the greatest effect will be given by a node-wise balancing scheme in a mechanical system with dynamic coupling, for example, in percussion instruments or automatic weapons mechanical system to compensate for the inertial force from the reciprocating motion of the shutter during automatic reloading.
5. Conclusions

- The general mechanical system nodal balancing rational scheme making possible to achieve one of the machine dynamics relative objective that is reduction of vibration activity and inertial loading level of rings and joints of the mechanical system in motion, was proposed.
- Kinematic coupling using of the movements of the balanced object and counterweight allows to control the value of the latter.

The technical solution of nodal balancing is constructively achievable and promising for the implementation in mechanical systems of machines for various purposes.

6. References

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