Balancing fast-rotating parts of hand-held machine drive

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Abstract. The article considers the issues related to the balancing of fast rotating parts of the hand-held machine drive including a wave transmission with intermediate rolling elements, which is constructed on the basis of the single-phase collector motor with a useful power of 1 kW and a nominal rotation frequency of 15000 rpm. The forms of balancers and their location are chosen. The method of balancing is described. The scheme for determining of residual unbalance in two correction planes is presented. Measurement results are given in tables.

1. Introduction

The drive for a hand-held electric machine is constructed on the basis of a collector motor and a wave reducer with intermediate rolling elements [1]. It has a small mass and dimensions in comparison with drives, in designs of which the other types of motors and mechanical transmissions are used. The wave generator of a gear with intermediate rolling elements can be designed both structurally balanced [2], and unbalanced. That can be seen in Figure 1.

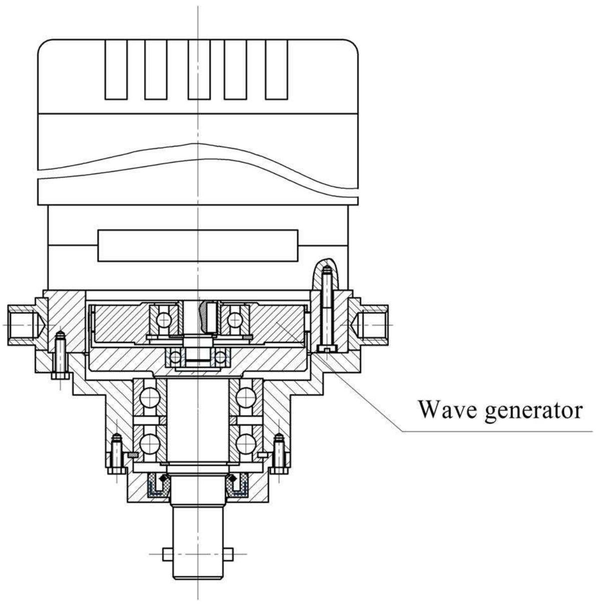

Figure 1. Drive of the hand-held machine

In the latter case, axial sizes of the drive decrease and its weight decreases too, but there is a necessity of balancing of fast-rotating parts: armature of the electric motor with the disk of the wave generator installed on it with ecscentricity. At the first stage of work, it is necessary to balance the armature of the motor and the wave generator independently of each other, and then to balance their assembly. The peculiarity of the being solved problem is due to the non-constant angular position of the wave generator disk in relation to the eccentric mounted on the motor armature. Therefore, the stability of measuring results of residual imbalance is not high and it is in a certain range of values. For this assembly unit, it is necessary to perform the dynamic balancing in accordance with ISO 1940-1 [3]. The dynamic balancing is effective for most rotors working between the first and the second own frequencies and it allows one to reduce vibration in bearings significantly [4].

To do this, the location of correction planes has been chosen. Figure 2 shows that first plane A is located between the collector and the bearing face, and second plane B is located on the fan.

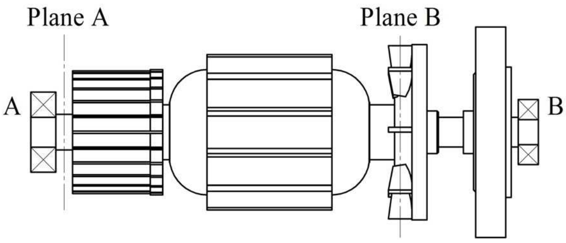

Figure 2. Location of the correction planes

The design of a balancer for the first correction plane is shown in Figure 3a. As shown in Figure 3b, the balancing weights are located on the fan in the second correction plane.

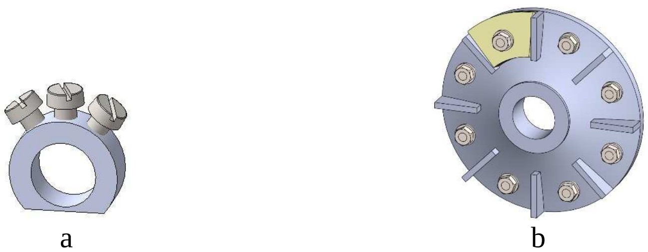

Figure 3. Balancers for: a – the correction plane A; b – the correction plane B

The designs of balancers make it possible to adjust their mass and location in a certain range of values, which is sufficient to achieve the target goal.

2. Results and discussion

The unbalance of the generator disk, located eccentrically in relation to the axis of rotation of the motor armature, is determined from the dependence [5]:

$$D = m \cdot e, \tag{1}$$

where m – mass of the generator disk, g; e – mass center displacement of the generator disk in relation to the axis of motor armature rotation, mm.

This imbalance can be eliminated by the installation of loads, as shown in Figures 3a,b in correction planes A and B. The location of the correction planes is determined by the design peculiarities of the device and, as a rule, they are located at the largest possible distance from each other. It should also be noted that the influence coefficients determined for each correction plane must be different, otherwise the balancer machine software will perceive both correction planes as one, and it will not be possible to reduce the dynamic imbalance of the device. In the case under consideration, the balancing weights are located at the largest possible radius in the most remote planes, which ensures the fulfillment of the specified requirements.

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To determine the values of the influence coefficients, it is necessary to perform three starts of the balancing machine [7]. At the first start-up, the initial values of vibration parameters $\bar{A}_0$ and $\bar{B}_0$ in both bearing supports are determined. During the second start-up, test load P_1 is set in plane A and the values of vibration parameters $\bar{A}_1$ and $\bar{B}_1$ in both bearings are determined due to the test load. After that, the changes in the values of the vibration parameters caused by the installation of test load P_1 are calculated:

$$\begin{aligned} \Delta\bar{A}_1 &= \bar{A}_1 - \bar{A}_0, \\ \Delta\bar{B}_1 &= \bar{B}_1 - \bar{B}_0. \end{aligned} \qquad (2)$$

The values of the influence coefficients of plane A ($\bar{\alpha}_1$) on the vibration level in bearing supports A and B are determined from the dependences:

$$\begin{aligned} \bar{\alpha}_{1A} &= \frac{\Delta\bar{A}_1}{\bar{P}_1}, \\ \bar{\alpha}_{1B} &= \frac{\Delta\bar{B}_1}{\bar{P}_1}. \end{aligned} \qquad (3)$$

Then test load P_1 is removed from plane A and installed in plane B. The load, set in the second correction plane, is designated as P_2. The third start is necessary to determine the influence coefficients for correction plane B. Then the changes in the values of the vibration parameters, caused by the installation of test load P_2 are calculated:

$$\begin{aligned} \Delta\bar{A}_2 &= \bar{A}_2 - \bar{A}_0, \\ \Delta\bar{B}_2 &= \bar{B}_2 - \bar{B}_0. \end{aligned} \qquad (4)$$

The value of the influence coefficients of plane B ($\bar{\alpha}_2$) on the vibration level in bearing supports A and B is determined from the dependences:

$$\begin{aligned} \bar{\alpha}_{2A} &= \frac{\Delta\bar{A}_2}{\bar{P}_2}, \\ \bar{\alpha}_{2B} &= \frac{\Delta\bar{B}_2}{\bar{P}_2}. \end{aligned} \qquad (5)$$

Test load P_2 is removed from plane B.

Thus, all four influence coefficients are determined, and after that with the help of them it is necessary to select balancing masses M_1, M_2 for the both correction planes. The equation which determines the vibration in bearing support A is:

$$\bar{\alpha}_{1A}\bar{M}_1 + \bar{\alpha}_{2A}\bar{M}_2 + \bar{A}_0 = \varepsilon_A, \qquad (6)$$

where ε_A is the residual value of the vibration parameter in bearing support A.

The equation which determines the vibration in bearing support B is:

$$\bar{\alpha}_{2B}\bar{M}_2 + \bar{\alpha}_{2B}\bar{M}_2 + \bar{B}_0 = \varepsilon_B, \qquad (7)$$

where ε_B is the residual value of the vibration parameter in bearing support B.

The solution of the system of two vector equations gives the required balancing masses for each correction plane:

$$\begin{cases} \bar{\alpha}_{1A}\bar{M}_1 + \bar{\alpha}_{2A}\bar{M}_2 + \bar{A}_0 = 0, \\ \bar{\alpha}_{1B}\bar{M}_1 + \bar{\alpha}_{2B}\bar{M}_2 + \bar{B}_0 = 0. \end{cases}$$

(8)

Then the balancing weights in the appropriate correction planes are set, and the test run is carried out. The values of vibration parameters $\bar{A}_3$ and $\bar{B}_3$ obtained during the control start are used to calculate the additional balancing masses using the system of equations (8). The values obtained at control start $\bar{A}_3$ and $\bar{B}_3$ are substituted in the left part of equations instead of initial vibration parameters $\bar{A}_0$ and $\bar{B}_0$. Additional balancing weights are set in the corresponding correction planes and the next test run is carried out. Control starts with the calculation and setting of additional weights have to be repeated until the residual imbalance is reduced to the values set in the standard for the adopted class of balancing.

Balancing of the fast rotating parts of the drive was performed on a BM-010 balancing machine [6]. All calculations are automated, and they are performed by the software of the balancing machine in a few seconds. The results of the measurements are given in Tables 1 and 2.

Table 1. Results of measurements of residual imbalance of the wave generator disk

№	1	2	3	4	5	6	7
Residual specific imbalance, g·mm/kg	3.3	4.7	4.3	3.6	5.2	3.8	5.3
Balancing mass, g	0.08	0.12	0.11	0.09	0.13	0.10	0.13
Installation phase balancing mass, degrees	209	188	198	198	193	205	191

Table 2. Results of measurements of residual imbalance of the fast-rotating parts

№	1	2	3	4	5	6	7
				Plain A			
Residual specific imbalance, g·mm/kg	6.2	5.1	7.5	6.6	7.3	2.8	5.9
Balancing mass, g	0.58	0.49	0.72	0.64	0.70	0.27	0.57
Installation phase balancing mass, degrees	353	343	328	296	320	357	339
				Plain B			
Residual specific imbalance, g·mm/kg	5.3	4.2	9.7	3.2	5.0	9.0	2.7
Balancing mass, g	0.27	0.22	0.51	0.17	0.26	0.47	0.14
Installation phase balancing mass, degrees	108	151	323	70	253	335	153

Table 1 shows seven non-grouped measurement results of the residual imbalance of the wave generator disk, the arithmetic mean of which is $\bar{X}$ =4.31 g·mm/kg. Estimate S of the standard deviation σ for these values is 0.79. With a confidence probability of 0.99, it can be said that the following measurements of the residual imbalance of the wave generator disk will be in the range of 4.31±1.11 g·m/kg, which corresponds to the G6.3 balancing class. Mass of the wave generator disk is 574 g.

Table 2 shows seven non-grouped measurement results of the residual imbalance of the fast rotating parts of the drive for correction plane A, the arithmetic mean of which is $\bar{X}$ =5.91 g·mm/kg. Estimate S of the standard deviation σ for these values is 1.6. With a confidence probability of 0.99, it can be said that the following measurements of the residual imbalance of the wave generator disk will be in the range of 5.91±2.24 g·mm/kg, which corresponds to the G16 balancing class. Mass of the fast rotating parts of the drive is 1835 g.

Seven non-grouped measurement results of the residual imbalance of the fast rotating parts of the drive for correction plane B are also presented in Table 2, the arithmetic mean of which is $\bar{X}$ =5.58 g·mm/kg. Estimate S of the standard deviation σ for these values is 2.73. With a confidence probability of 0.99, it can be said that the following measurements of the residual imbalance of the wave generator disk will be in the range of 5.58±3.82 g·mm/kg, which corresponds to the G16 balancing class [3, 8].

The manufacturer guarantees the level of residual unbalance corresponding to G2.5 for the motor armature.

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3. Conclusion

Thus, due to the work done the balancing of the fast rotating parts of the drive of the hand-operated electric drill has been carried out in accordance with the class of balancing G16. The achieved level of residual vibration is acceptable for machines of this type. Further reduction of the magnitude of the residual imbalance in this construction is not possible because the value of the phase of the installation of an additional load during repeated tests is not stable. The range of values is significant 40±140 degrees. It is planned to reduce the amount of residual unbalance of fast rotating parts of the drive by reducing the mass of the disk of the wave generator at the next stage of work, for example by drilling holes in it, etc.

References

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