ISSN 2181-8622

Manufacturing technology problems

Scientific and Technical Journal Namangan Institute of Engineering and Technology

Volume 10
Issue 1
2025

A STUDY OF CRITICAL SPEED OF LINTER SHAFT ROTATION AND RESONANCE PHENOMENON

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Abstract: Insufficient smoothness of the saw cylinder shaft leads to impermissible misalignments in the gap between the columns, which has a very negative effect on the grinding process. In this case, using the method of equivalent diameters developed by B.P. Jemochkin, we determine the bending and turning angles of the saw cylinder shaft.

Keywords: shaft, singularity, bending vibrations, saw cylinder, slack, technological slot, saw, colossian.

Introduction. Resonance event is used periodic strength frequency occurred when it equals or is close to the natural frequency of the affected systemof amplitudede scribes the phenomenon of increase. When an oscillating force is applied at the resonant frequency of a machine dynamic system, the system is higher than when the same force is applied at other non-resonant frequencies.amplitude vibrates. All parameters of the system are unchanged, and if the period of the external force of this system is sinusoidal, that is, if the system's specific oscillation frequency corresponds to the frequency of the external force, resonance occurs.

From our research, it is known that this external force is not random, but it is often considered to be a periodic influence on the movement of the car. Under the influence of such a periodic excitation force, the oscillations are forced rather than free. At the same time, it is absolutely irrelevant whether this force is actually an external force, whether it is considered a driving force or a resistance force; it can be friction force or inertia force, it can be a force that occurs in the movement of the machine and is caused by the action of rotating parts; only the periodic character of this force is important in the operation of the machine, and not the magnitude of the force, but its periodic frequency is important.

At the same time, in some conditions, the phenomenon of resonance occurs, that is, the period of the driving force coincides with the period of the specific vibrations of the elastic body. In this case, the amplitude of oscillations can gradually increase and reach an infinite value, and during resonance, the elastic link must collapse.

Materials and methods. Resonance is used periodic strength frequency which occurs when it is equal to or close to the natural frequency of the system it affectsof amplitudedes cribes the phenomenon of increase. When an oscillating force is applied to a dynamic system at its resonant frequency, the system is higher than when the same force is applied at other non-resonant frequencies.amplitude vibrates.

The speed of the machine at which the phenomenon of resonance begins is called the critical speed. If we denote the centrifugal inertial force of mass m by P:

$$P = m \cdot j_n = m \cdot \omega^2 \cdot R \tag{1}$$

in which it is transmitted to its vertical member acting vertically up and down and causing their forced vibrations. The phenomenon of resonance can occur in similar conditions and in cases of torsional vibrations.

In production, that is, the shafts may not reach or exceed the critical speed in the technological process, but in this case, it is necessary to limit the amplitude of vibrations with the help of elastic supports and prevent the decay of the elastic body.

We will think about the number of critical revolutions of machines, which is of great importance in modern engineering.

There is a shaft AB loaded with a disk D from the center by a weight G (Fig. 1), and under the influence of this weight G, the shaft bends to an amount a. It can also be said that when the shaft is stationary, the center of gravity of the disk S is located below the geometric axis of the shaft at a distance a.

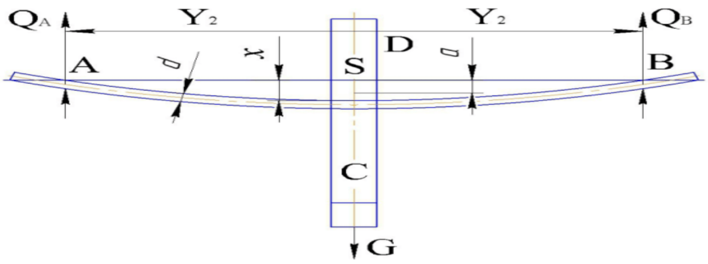


Figure 1. Bending of the shaft when the phenomenon of resonance occurs

When the disk rotates with an angular speed ω, the centrifugal inertial force increases and bends the shaft, increasing the initial eccentricity a to the magnitude x, at which a balance is established between this centrifugal force C:

$$C = m \cdot \omega^2 \cdot x = \frac{G}{g}\omega^2 \cdot x \tag{2}$$

The elastic resistance tolerance depends on the dimensions of the shaft and the location of the supports (bearings) and the load, and is directly proportional to the linear deformation of the shaft. If at the given dimensions of the shaft and the given location of the supports, the force Q will produce a deflection of 1 cm if we accept, then the elastic resistance of a shaft bent in rotation (xa) centimeters will be Q (xa) kilograms, and according to the statements:

$$Q \cdot (x - a) = C = \frac{G}{g}\omega^2 \cdot x \tag{3}$$

because the elastic resistance of the shaft at the angular velocity ω and (xa) caused the deflection balance is established between centrifugal force; so

Q, kg/cm.

From here, the final eccentricity of the shaft rotating at speed ω is x

$$x = \frac{a}{1-\frac{G}{g}\cdot\frac{\omega^2}{Q}} \tag{4}$$

So,

$$i > \frac{G}{g}\cdot\frac{\omega^2}{Q}$$

when the eccentricity x increases with an increase in the angular velocity ω of rotation, when the expression is equal to zero:

$$1 - \frac{G}{g}\cdot\frac{\omega^2}{Q} = 0 \tag{5}$$

or

$$\omega_k = \sqrt{\frac{Q\cdot g}{G}} \tag{6}$$

in equality, the eccentricity x is equal to infinity, that is, if the infinite increase in deflection is not affected by other parts of the machine, the shaft will wear out.

At the further increase of the angular velocity ω, the eccentricity x takes a negative value, that is, the shaft begins to tilt in the opposite direction, and at the value takes a value, which means that a new stable equilibrium of the rotating shaft is formed. It can be said that at infinitely high speed, the elastic shaft is fixed by itself, and its axis of rotation is a free axis, that is, the loaded elastic shaft exhibits the property of automatic centering, based on the expression (2.22), the critical number of rotations is equal to:$\omega = \infty x = 0$

$$n_k = \frac{30\omega_k}{\pi} = \frac{30}{\pi}\sqrt{\frac{Q\cdot g}{G}} \approx 300\sqrt{\frac{Q}{G}} \tag{7}$$

because the acceleration of gravity$g = 9{,}81\ m/s^2$.

*Q*from determining the force, the bending arc for it f equal to:

$$f = 1 = c\frac{Q\cdot l^3}{E\cdot I} \tag{8}$$

from here

$$Q = \frac{E\cdot I}{c\cdot l^3} \tag{9}$$

and the critical speed of rotation at which resonance begins:

$$\omega_k = \sqrt{\frac{Q\cdot g}{G}} = \sqrt{\frac{E\cdot I\cdot g}{G\cdot c\cdot l^3}} = \sqrt{\frac{g}{a}} \tag{10}$$

if we denote the deformation of the shaft under the influence of a given load by a:G

$$a = c\frac{Q\cdot l^3}{E\cdot I} \tag{11}$$

PGeneral expression for the bending arc of a shaft loaded with force:

$$f = c\frac{P\cdot l^3}{E\cdot I} \tag{12}$$

coefficient c depending on the type of load and support.

Substituting from (2.21) to (2.22) according to the expressions, we get:$G/Q\cdot g\ 1/(\omega_k)^2$

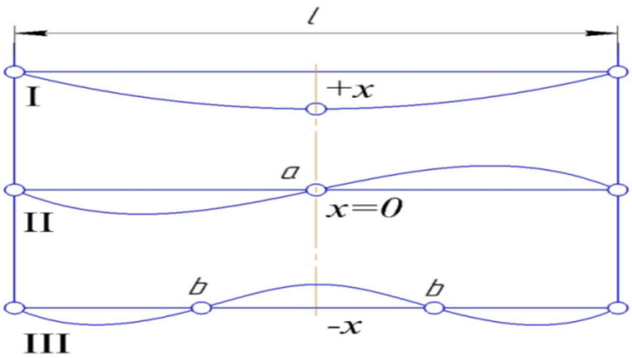
$$x = \frac{a}{1-\left(\frac{\omega}{\omega_k}\right)^2} \tag{13}$$

It can be seen from here that when , that is, when the angular velocity of rotation is equal to the critical angular velocity expressed in the formula (2.25), resonance occurs.$\omega = \omega_k x = \infty \omega \omega_k$

3.Results and discussion

In general, the larger the ratio of the angular velocity of rotation to the critical angular velocity, the smaller the dynamic deformation, or amplitude of vibration.$\omega \omega_k x$

This is an incomprehensible phenomenon at first glance - a change in the sign of deformation with a decrease in its absolute magnitude when the speed is increased - a high-speed rotating shaft respectively, it can be explained by the properties of taking different forms with different numbers of waves at different vibration frequencies (Fig. 2). it is known that when the speed is increased, the wavelength decreases and nodal points appear (a, b, b) where the amplitude of oscillations is zero.



In I-deformation, in II-deformation, and in III, the shape of the elastic body in the same place.$+x x = 0 - x$

Figure 2. Three different shapes of high-speed shaft are shown

Of course, not only the shaft, but also all other elastic bodies have these properties, the deformation at the critical speed does not pass through zero, but theoretically through infinity, increases to plus infinity when the speed increases up to the critical speed, and decreases to minus infinity when the speed exceeds the critical speed. Basically, the phenomenon under consideration is essentially the phenomenon of oscillations under consideration during rotation, in a period equal to one revolution of the shaft, of equal magnitude, but directed at an angle of 90° to each other, two simultaneously acting elastic Looks like a combination of 'ndalang vibrations. Thus, during one rotation of the shaft, the tension of its fibers changes direction, and the critical speed is independent of the influence of gravity, that is, it depends on the location of the shaft in space. coincides with the frequency of specific elastic transverse vibrations of the shaft.

The corresponding angular speeds of shaft rotation are called the first critical (Fig. 2, I), the second critical (Fig. 2, II), and the third critical (Fig. 2, III).

In order to determine the period of specific vibrations of an elastic body T in a simple way, we see that the frequency of changes in the force of gravity relative to the same surface of this shaft, which causes forced vibrations of the shaft in the considered case, is exactly equal to the number of its revolutions: k = n. Therefore, the period T of this driving force in one second is equal to:

$$T = \frac{60}{k} = \frac{60}{n} \tag{14}$$

From this;

$$T = \frac{2\pi}{\omega} \quad (15)$$

But when there is a resonance, it means that the period of the driving force coincides with the period of the specific vibrations of the elastic body, as mentioned above.$\omega = \omega_k T T_s$

From here, based on the formula (12):$\omega = \omega_k$

$$T_s = T = 2\pi\sqrt{\frac{a}{g}} \quad (16)$$

or using formula (2.23) for this case, we find:

$$T_s = \frac{2\pi}{\omega_k} \quad (16)$$

It is known that the external force, namely the force of gravity, is always directed downwards, and its effect on the shaft, causing forced vibrations - driving it, occurs due to the rotation of the shaft around its inclined axis, in which, at each revolution of the shaft, under the constant force of the disk weight G, the same fiber is sometimes on the convex side, and sometimes on the concave side. Therefore, the shaft remains stationary, and the force of gravity changes its direction up and down with each revolution of the shaft, which means that we are observing a case of forced oscillations here.

But the reaction force in the opposite direction (elastic force) does not depend on one or another position of the shaft in space. Therefore, the period of free oscillations of the disc shaft, and therefore the critical number of revolutions, remains constant for the same shaft, which is horizontal, inclined or vertical, as shown above.

It follows from the above that we can propose a very strong, heavy shaft acting on centrifugal inertial forces or, on the contrary, a hollow shaft that adopts a new form of freely flexible and elastic balance under the influence of inertial forces.

Linter if the saws attached to the shaft of the machine are unevenly located or random resistances on the raw material shaft cause forced oscillations of the saw cylinder shaft, while the period of this driving force is equal to the time of one revolution of the shaft. If the number of rotations of the shaft of such a machine is equal to the number of special vibrations of the system consisting of the machine, then resonance occurs with all the harmful consequences.

In order to avoid this, the number of revolutions of the machine shaft should be different from the number of critical revolutions corresponding to the resonance, it should not be equal to it in a simple ratio, i.e., etc., because resonances of the second, third, etc. ladi But the latter, which is safer than the first-order resonance, is not always straightforward.$n_k n = n_k/2, n = n_k/3 \;\; n = n_k n_k$

The number of rotations of the 5LP linter machine shaft in the scope of our study is usually not reached to the critical state, that is, the maximum number of revolutions is not reached, at the same time, it is enough to bring the difference to 10-15% to significantly reduce the vibrations, and the critical speed is 35-70% higher the one-time speed for is thus obtained.

It is best to avoid resonance at the beginning of the design. To do this, it is necessary to calculate the period of specific oscillations of the elastic system or the critical speed of the machine shaft by calculation, and then, if necessary, make appropriate structural changes in the dimensions or location of individual parts. Also, in the process of assembly of the saw cylinder, a certain amount of displacement of the saw and inter-saw gaskets attached to the shaft, or if they are not properly positioned, helps to cause the phenomenon of resonance.

If the machine is designed and it is necessary to reckon with the phenomena that exist with resonance, then the aforementioned, not only in the design, but also in the construction of the details that are designed and used, except for the measures that can sometimes be used, i.e. It is very important to carefully balance the inertial forces of the moving masses both during design and when balancing them at the factory.

After constructing the coolness curve of the shaft, its specific lower frequency, that is, the critical number of revolutions, is found as follows: the product of the weight of some sections of the shaft by the coolness of this section and the square of the coolness values of the weights of some sections It is calculated by multiplying the weight of the plot and the weights of some plots by the square of the coolness values. Then the angular critical speed of the shaft is found from the following formula.

$$\omega_k = \sqrt{\frac{g \cdot \sum G \cdot y}{\sum G \cdot y^2}},$$

critical number of revolutions: $n_k = \dfrac{\omega_k}{2\pi}$

where: G is the weight of some sections of the shaft, in kg;

y –the coolness ordinate for the respective sections.

It is known that saw discs are installed on the shaft along with seals along the entire longitudinal bar. At one end there is a half coupling. How the load is distributed is shown in the diagram below the shaft (Fig. 3). The direction of the loads at one end of the shaft and the direction of the forces acting on the surface of the shaft through the series of dialed saws are inversely proportional.

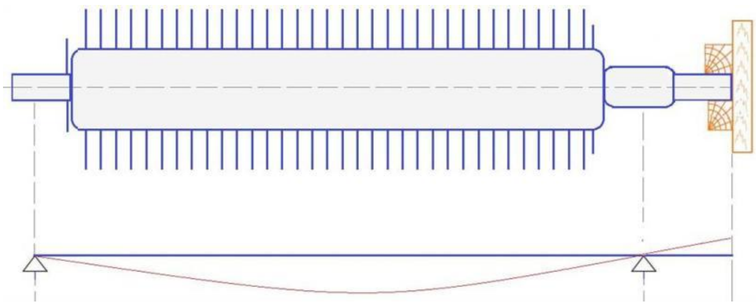


Figure 3. The first transverse oscillations of the shaft axis bent shape under the influence of frequency

This case corresponds to the greatest shaft coolness and the shape of the bent shaft axis at the time of the first or fundamental frequencies of the characteristic vibrations. Taking into account the above, we construct the ordinates of the bent shaft axis curve (in Figure 3) by the pen graphic method. We will write down the analysis of the obtained results in order to find the stiffness in the sections of the shaft corresponding to the points of some loadings (tables 1 and 2). In order to find the actual amount of torques, the ordinates of the torques obtained from the diagram y are determined by the critical value of the rotation frequency at which the resonance phenomenon begins in the shafts. The structure of each shaftand depending on the working conditions, the critical value of the rotation frequency will be different. When the actual rotation frequency of the shafts reaches the critical value, the frequency of the external forces will correspond to the specific vibration frequency. In such cases, the vibration amplitude increases dramatically, and as a result, the shaft breaks. Therefore, in order for resonance not to occur, the value of the rotation speed of the shafts in this condition should not be equal to the critical value of the rotation frequency. It should also be assumed that the shafts with the rotation frequency reaching a critical value do not suddenly break.For this reason, the actual rotation frequency of the shafts in the required points may be greater than the critical value.

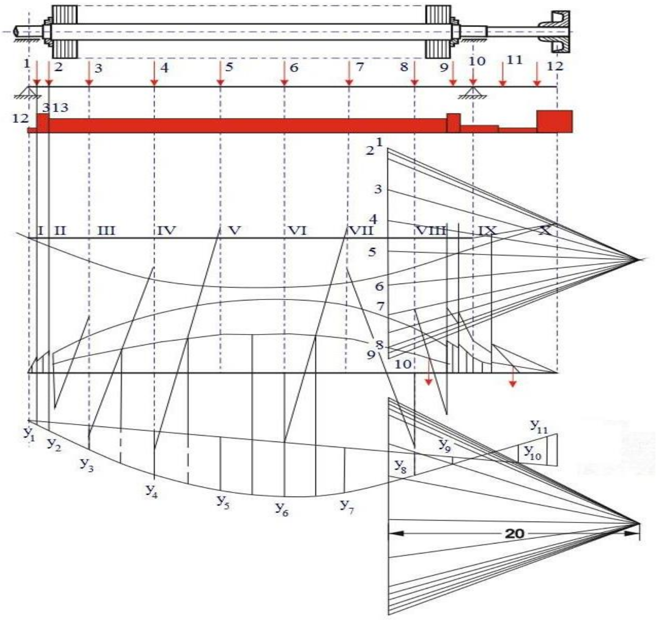


Figure 4.5LP lintergiven graph for determining the critical speed of rotation of the machine saw shaft

Table 1

No	Moment of inertia, J	Bending moment, M	M/J	A dummy loading plot	Fake upload coming down plot face
	cm4	kg/cm	kg/cm3		kg/cm2
I	44.8 55.7	340	7.6 6.1	A	22.8
II	55.7 146	600	10.7 4.1	B C	37.8 81
III	146	1250	8.55	D	302
IV	146	2200	15	E	430
V	146	2700	18.5	F	475
VI	146	2700	18.5	K	445
VII	146	2350	16.1	L	330
VIII	146	1400	9.6	M	97
IX	146 55.7	810	5.5 4.3	N O	56 60.2
X	55.7 44.8	605	10.7 13.4	P R	27.7 61.6
XI	44.8	300	6.7		
XII	44.8 30.7	251	4.45 6.5		

Table 2

No	Coolness ordinate y (cm)		Weight of shaft section (kg)	Multiplication	
	According to the drawing (cm)	to scale (cm)		$G \cdot y$	$G \cdot y^2$
1.	**2.**	**3.**	**4.**	**5.**	**6.**
1.	0.4	$18{,}6 \cdot 10^{-4}$	1.2	22.3	$415 \cdot 10^{-8}$
2.	0.9	$42 \cdot 10^{-4}$	3.15	$132 \cdot 10^{-4}$	$5550 \cdot 10^{-8}$
3.	2.5	$116 \cdot 10^{-4}$	18	$2090 \cdot 10^{-4}$	
4.	4.7	$218 \cdot 10^{-4}$	18	$3920 \cdot 10^{-4}$	$4985000 \cdot 10^{-8}$
5.	5.9	$274 \cdot 10^{-4}$	18	$4940 \cdot 10^{-4}$	
6.	6.0	$280 \cdot 10^{-4}$	18	$5040 \cdot 10^{-4}$	
7.	4.7	$218 \cdot 10^{-4}$	18	$3920 \cdot 10^{-4}$	
8.	2.6	$120 \cdot 10^{-4}$	18	$2160 \cdot 10^{-4}$	
9.	1.0	$46{,}5 \cdot 10^{-4}$	3.15	$146 \cdot 10^{-4}$	$6780 \cdot 10^{-8}$
10.	0	$0 \cdot 10^{-4}$	2.04	$0 \cdot 10^{-4}$	$8900 \cdot 10^{-8}$
11.	1.5	$69{,}5 \cdot 10^{-4}$	1.85	$128 \cdot 10^{-4}$	$8900 \cdot 10^{-8}$
12.	3.2	$149 \cdot 10^{-4}$	14.0	$2082 \cdot 10^{-4}$	$311000 \cdot 10^{-8}$

In such cases, it is necessary to ensure a quick transition from the operating state with the critical rotation frequency. In addition,resonance phenomenon is criticalrepeats at multiples of the rotation frequency, which should not be forgotten.

The resonance of the shafts was investigated in theoretical and experimental studies. It was found that the resonance phenomenon did not occur in the existing construction. The resonance phenomenon was not observed in the proposed construction. There was no overlapping of the amplitude of the natural and forced oscillations.

Conclusion In order to reduce the bending of the shafts of the long, heavy and fast-rotating parts of the technological machines, the longitudinal groove structure of the lightened shafts was developed, including for the saw cylinder, on the basis of which the technological gap between the saw cylinder and the fiber separation brush is sufficiently maintained, the fiber output is increased, pollution and power consumption are reduced.

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CONTENTS

CHEMICAL SCIENCES

TECHNICAL SCIENCES: MECHANICS AND MECHANICAL ENGINEERING

ADVANCED PEDAGOGICAL TECHNOLOGIES IN EDUCATION