ISSN 2181-8622

Manufacturing technology problems

Scientific and Technical Journal Namangan Institute of Engineering and Technology

Volume 10
Issue 3
2025

STUDY OF THE DYNAMICS OF THE DRIVE MECHANISM OF MOVING NEEDLES

MURADOV AKRAMJON

Professor, Namangan State Technical University, Namangan, Uzbekistan
Phone.: (0599) 519-8053, E-mail.: muradovakramjontmj@gmail.com

Abstract: This article examines design issues and calculations that take into account the dynamic nature of loads acting on machines. As is well known, all bodies with mass and elasticity are capable of oscillatory motion. Unwanted vibrations in a mechanism can reduce the strength of the connection between parts, disrupt the machine's technological process, or ultimately lead to failure of the mechanism or machine. The goal of vibration analysis is to determine the parameters of an oscillatory system that will ensure optimal operating conditions.

Keywords: dynamics, vibrations, displacement, kinematic disturbance, needle bed, knitting and stitching machine, drive, mechanism, loop formation, frequency.

Introduction. Modern industry is characterized by high-speed process equipment used to produce both consumer and industrial goods. Therefore, calculations that take into account the dynamic nature of the loads acting on machines play an increasingly important role in the design of new equipment [1]. In particular, calculations of machines, mechanisms, and individual components for vibrations caused by process and inertial loads arising during machine operation are of great importance.

The overall performance of a knitting-stitching machine is determined by the perfection of the loop-forming system and the drive mechanisms of the working elements. The loop formation process on knitting-stitching machines is carried out simultaneously, in which all the machine needles move simultaneously, as in the production of knitwear on warp knitting machines [2, 3]. One of the features of the technological process carried out on knitting-stitching machines is the need to pierce the fibrous web with moving needles, and the technological load arising from piercing the fibrous web makes a significant contribution to the workload on the moving needle drive.[4].An important characteristic of the newly designed drive of the sliding needles is the natural frequency of the beams, knowledge of which allows one to avoid possible resonances of the structure at operating speed.The sliders and rods of the sliding needles of the VP-II, VP-180M, and Maliwatt knitting and stitching machines have a rectilinear reciprocating motion [5]. A significant drawback of these mechanisms is the presence of a kinematic pair of sliders and guides. This pair requires continuous lubrication during operation. To eliminate this drawback, a new mechanism for driving the sliding needles andcontains an elastic link, while the driven link of the mechanism is a connecting rod that performs a flat movement.[6].

Supported system. In these mechanisms, the kinematic disturbance is applied not to the mass itself, but to the support or base of the system. In this case, both the absolute and relative motion of the mass and base are of interest.

Let us first consider the absolute motion $x_2(t)$ of mass m (Fig. 1).

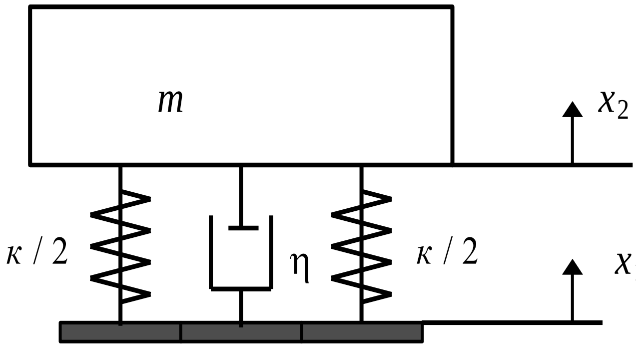

Fig. 1. System on substructure supports

If the base of the system moves according to a given law x1(t). According to Newton's second law, we will have

$$m\ddot{x}_2 = \sum Xi = -k(x_2 - x_1) - \eta(\dot{x}_2 - \dot{x}_1)$$

or

$$m + \eta\, \dot{x}_2 + \text{to } x2 = \text{to } x1 + \eta.\ (1.)$$

Let x1 (t) = a sinωt. Then (using the abbreviation for shorthand notation $\frac{\omega}{p}$ = c) ,

m $\ddot{x}_2$+η $\dot{x}_2$+ to x2 = to a (sinωt+$\frac{\eta\omega}{\kappa}$ cosωt) = to a $\sqrt{1+(2\zeta c)^2}$ sin(ωt+γ),

Where γ= arctg (2ζc).

If for a given frequency ω coefficient of sin(ωt+γ) is a constant value, then, denoting it by Fэ, we can write

$m\ddot{x}_2$+η $\dot{x}_2$+ to x2 = Fэ sin(ωt+γ). (2)

We know a particular solution to this equation:

$$x_{2\ \text{hours}} = \frac{F_э}{\kappa}\chi\sin(\omega t+\gamma-\varphi),$$

Where$\chi = \dfrac{1}{\sqrt{(1-c^2)^2+(2\zeta c)^2}}$; φ= arctg $\dfrac{2\zeta c}{1-c^2}$.

Substituting the expression for Fэ into the equation, we get

$x_{2\ \text{hours}}$= a $\sqrt{1+(2\zeta c)^2}$ χsin(ωt+γ-φ). (3)

From formula (3) it is clear that asω/p the amplitude value x2h will change in the same way as the dynamic gain factorχ, i.e. x2h→0 at ω/p→ ∞.

If we accept that to→ ∞(the machine is installed without intermediate elements on a movable base), then p→ ∞, ω/p→0 and x2h max→a , i.e. the mass will move together with the base.

In many cases, this situation is undesirable. For example, suppose a precision metal-cutting machine is located on the second floor of a production facility (a rare, but possible, occurrence).

Due to vibrations in the ceiling, the machine may not be able to provide the required accuracy of processing the part.

Then it is installed on springs, the rigidity of which is selected so that x2h does not exceed the specified value.

Let's look at an example. It is necessary to select the rigidity of the elastic suspension of the machine so that when ζ= 0.1 the maximum amplitude x2чmax of the machine oscillations was equal to (or less than) 0.1a, i.e.

$$\text{x2h max} = 0.1a = \frac{a\sqrt{1+0{,}04c^2}}{\sqrt{(1-c^2)^2+0{,}04c^2}}.$$

Having solved this equality, we obtain $c^2 = (\omega/p)2 = 13.37$.

Since p2 = k / m, the suspension stiffness is determined by the formula

$$To \le \frac{m\omega^2}{13{,}37}.$$

Of great interest is the relative displacement of the mass and the base.

The relative displacement is defined as the difference between the absolute displacement x2 of the mass and the given or known movement x1 of the base, i.e.

$$x = x2 - x1.$$

From here

$$x_2 = x + x1.$$

Substituting x2 into equation (1.35), we obtain

$$m\ddot{x}+\eta\dot{x} + \text{to } x = -m\ddot{x}_1 = m\omega^2 a\sin\omega t. \quad (4)$$

The particular solution will be written as follows:

$$x_h = \frac{m\omega^2 a}{\kappa}\chi\sin(\omega t - \varphi) = c2\chi a\sin(\omega t - \varphi); \quad \varphi = \mathrm{arctg}\frac{2\zeta c}{1-c^2}. \quad (5)$$

Factor $c^2\chi$ is an indicator of the deviation of the amplitude of relative motion from the given kinematic amplitude a.

Let us plot a graph of the change in this multiplier (Fig. 2).

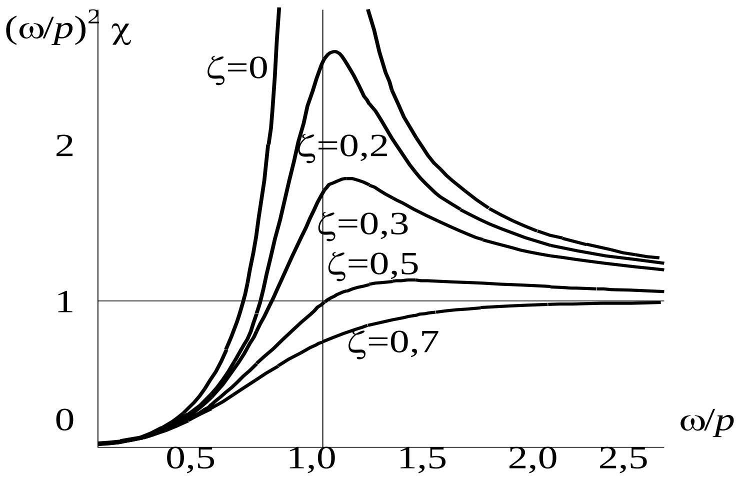

Fig. 2. Graph of changes

As can be seen from the graph, in the high frequency region the value $c^2\chi$ approaches 1 . In fact, when $\omega/p \to \infty$ magnitude $c^2\chi = \to 1$

$$\lim_{c\to\infty} \frac{c^2}{\sqrt{(1-c^2)^2+(2\zeta c)^2}} = \lim_{c\to\infty} \frac{1}{\sqrt{(1/c^2-1)^2+(2\zeta/c)^2}} = 1. \quad (6)$$

It should be noted that the phase angle of the relative displacement x in this case does not coincide with the phase of the base movement by almost 1800.

Indeed, when $\omega/p \to \infty$ (see equation (5)

$$\text{tg}\varphi = \frac{\sin 180^0}{\cos 180^0} = -\,0.$$

Let us use the above and consider how to obtain the equations of motion of the system with unbalanced rotational and reciprocating movements.

In knitting and stitching machines, the needle bar develops the greatest inertial force compared to other mechanisms.

The diagram of the machine is shown in Fig. 3. The reciprocating mass m n consists of the mass of the needle bed, the mass of the rods and part of the mass of the connecting rods.

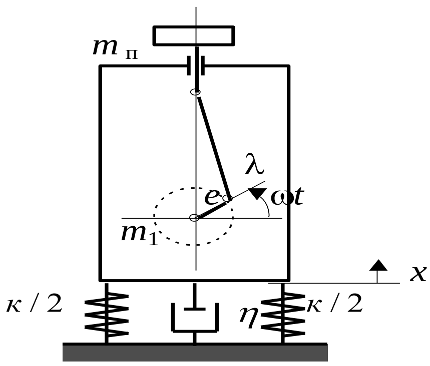

Fig. 3. Model of the mechanism

The inertial force of the reciprocating motion of the mass m n is approximately equal to

$$P_{And} = \text{m p}\,\omega^2 e(\sin\omega t + \text{e/l} \sin 2\omega t).$$

If e/l is small, which is practically the case in warp knitting (knitting-stitching) machines, then the second term can be neglected and then

$$F_{eh} = \text{m p}\,\omega^2 e;\ F(t) = F_Э \sin\omega t.$$

Conclusion. Thus, harmonic excitation occurs in a wide range of textile and light industry machines. The resulting relationships can be used to analyze the dynamics of these machines.

References

1. Antsiporovich P.P.Dynamics of Machines and Mechanisms in Steady-State Motion: A Tutorial. BNTU, 2004. – 42 p.

2. Muradov AA, Rahimberdiyev DO Tirqishli ignalar sterjenlarining brusidagidagi texnologik yuklamalarni eksperimental tadqiq qilish. // Mexanika va Texnologiya ilmiy jurnali 5-jild, 2-son, 2024.

3. Rahimberdiyev DO, Muradov AA Designing resource-saving mechanisms for stable operation of the loop-forming system of stitch-knit machines // AIP Conf. Proc. 2969, 060002 (2024)

4. Muradov AA Designing a loop-forming system of knitting and stitching machines "Vestnik" Tash No. 4, 2016.

CONTENTS

CHEMICAL SCIENCES

TECHNICAL SCIENCES: MECHANICS AND MECHANICAL ENGINEERING

ADVANCED PEDAGOGICAL TECHNOLOGIES IN EDUCATION

ECONOMICAL SCIENCES