ISSN 1392–1207. MECHANIKA. 2014 Volume 20(3): 299–304

Design and analysis of cam lifting curve in applying to transient and heavy load

J. Yang*, J. Tan, L. Zeng***, S. Liu******
*****Central South University, 410083 Changsha, China, E-mail: yangjun8505@163.com
******Central South University, 410083 Changsha, China, E-mail: jptan@163.com
*******Central South University, 410083 Changsha, China, E-mail: zenglewish@163.com
********Central South University, 410083 Changsha, China, E-mail: 15200852420@163.com

crossref http://dx.doi.org/10.5755/j01.mech.20.3.5381

1. Introduction

Cam follower mechanism which can change continuous motion into periodic motion is widely used in industry because of its simple structure and high accuracy [1, 2]. Since the cam profile curve directly determine the kinematics and dynamics characteristics of the mechanism, designing appropriate cam profile curve caused the wide attention of scholars. Traditional profile curves are harmonic ladder cycloid arc curves and improved ones based on these curves [3, 4]. With the development of CAD technology and modern processing technology, further research are taken place by using high order polynomial curve bezier curve B-spline curve etc. as cam profile curve. The applications of these complex curve profile make the system drive more flexible and less impact [4-10].

Although the Cam profile curve mentioned above can meet most requirements in industry, such as in high speed and light load occasion or in low speed and heavy load occasion, little study was being done in cam profile curve applied for transient and heavy load occasion. Using traditional curves as cam profile directly cannot match the characteristics of specific load, while transients and heavy load is a normal case in industry [11-13].

Aiming at transient and heavy load in large hydraulic press operating system, this paper put forward a new kind of cam lifting curve which can match up with the transient and heavy load. A composite curve was designed as the lifting curve of the cam profile. The curve matched up with the load characteristics using Involute to achieve smaller pressure angle in heavy load area and using quadratic curve to achieve faster opening speed in light load area. This cam curve used in large hydraulic press operating system improved the force condition and prolonged service life of the system. In particular, part 2 describes the operating system of large hydraulic press and the load characteristics. Part 3 puts forward the design method aiming at the characteristics of transients and heavy load. In order to give further illustration of the method, part 4 gives a specific example about how to design the curve in the operating system in 300 MN hydraulic press. The field application was given in part 5 and followed by concluding remarks.

2. Characteristics of the operating system

The operating system of large hydraulic press are made up of four parts which are hydraulic power part gear and rack transmission cam follower and Other accessory parts. As shown in Fig. 1. The rack drives the gear to rotate. The rotating gear shaft drives the cam shaft and the cam at the same shaft to rotate. At last, the rotating cam pushes the follower rise, then opens or closes the water valve Slowly.

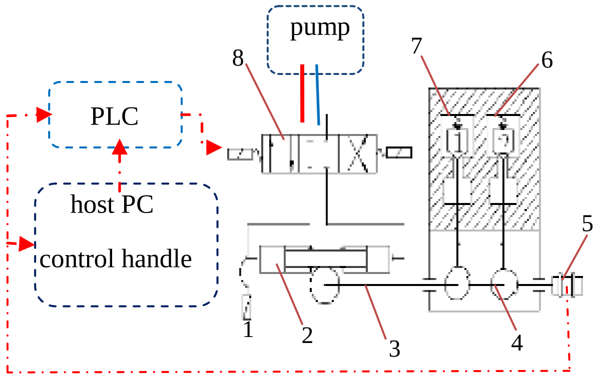

Fig. 1 Schematic diagram of operation system: *1* - pressure sensor; *2* - hydraulic cylinder; *3* - gear and rack *4* - cam follower; *5* - encoder; *6* - inlet valve; *7* - outlet valve; *8* - proportional valve

During the working process, the water pressure of supply is higher than 30 MPa generally. So the valve usually uses two levels of structure which are pilot pressure relief valve and main valve. The process of opening water valve is divided into two stages. The first step is opening the pressure relief valve, the opening force is small. The second step is opening the main valve after relieving the pressure. Even with two-stage valve structure, the main valve opening force is still large. In literature [14], authors deeply researched on the rule in water valve opening process. As shown in Fig. 2, the opening force is small in the early stage. Along with the rotation of the cam, the opening force instantly reached at about 50 kN during 0.2 ~ 0.3 s. After a short time for lasting on heavy load, the opening force reduced rapidly. By the analysis of the load, it found the characteristic of load is transient and heavy.

Fig. 3 shows the force diagram for the cam follower mechanism, where G is the opening force of water valve (including the opening force of valve and the weight of follower, etc.), r is the radius of the involute base circle, α is the pressure angle, F is the force between cam with roller, φ_1 is the friction angle, M is the driving torque for cam. Guide sleeve to guide rod on both sides of the reaction force are F_1, F_2 respectively, and the

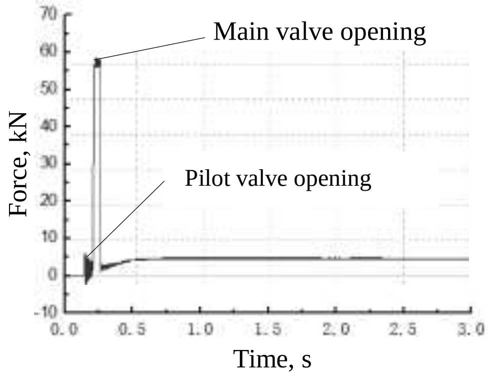

Fig. 2 Opening force of valve

friction angle is φ_2. The length of the guide sleeve is L_1, the distance between guide sleeve and the roller is L_2, the eccentricity is e and the base circle of cam is R.

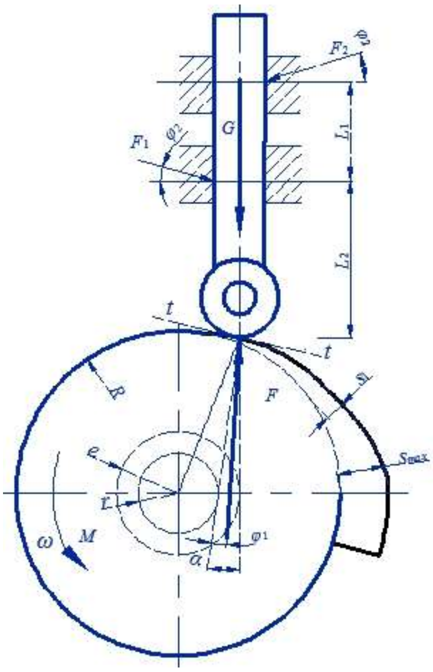

Fig. 3 Force analysis of cam follower

According to equilibrium of force and torque, F_1, F_2 and M are shown as follow:

$$\left.\begin{aligned} F_1 &= \frac{L_1+L_2}{L_2\cos\varphi_2}\times\frac{G\tan(\alpha+\varphi_1)}{1-L\tan(\alpha+\varphi_1)\tan\varphi_2},\\ F_2 &= \frac{L_1}{L_2\cos\varphi_2}\times\frac{G\tan(\alpha+\varphi_1)}{1-L\tan(\alpha+\varphi_1)\tan\varphi_2},\end{aligned}\right\} \qquad (1)$$

$$M=\frac{G\,e+G\left(\sqrt{R^2-e^2}+h\right)\tan(\alpha+\varphi_1)}{1-L\tan(\alpha+\varphi_1)\tan\varphi_2}, \qquad (2)$$

where $L=\dfrac{2L_1+L_2}{L_2}$.

According to Eqs. (1) and (2), F_1 and F_2 are related with pressure angle and friction angle. Reduction of the pressure angle and friction angle can reduce F_1 and F_2. M is not only related to the pressure angle and friction angle but also related to the eccentricity, Reduction of the eccentricity within a certain range can reduce driving torque. During the main valve open stage, the opening force is large. If the pressure angle of cam is big, F_1 and F_2 are great, at the same time, guide sleeve force also increases accordingly. The result can lead to serious wear and tear on cam and roller and guide sleeve.

So in view of opening process of the water valve, the research of a kind of curve that the pressure angle of cam can match the load characteristics is very important. By designing a reasonable lifting curve of cam, it made smaller pressure angle in open initial stage and have a faster lifting in the subsequent stage. In this way it can both meet the rapid opening of the valve and effectively improve force condition of the Cam follower system, eventually improve the service life of the device and operation safety.

3. Design lifting curve

Lifting curve is made up by involute in heavy load area and quadratic curve in light load area. When the radius of involutes base circle and the offset circle of cam are equal, the cam profile can lead zero pressure angle with the roller reducing the stress on guide sleeve. Quadratic curve can guarantee a faster velocity and constant acceleration. As shown in Fig. 4, total lifting is s_{max}, total rotation angle of cam is θ_{max}, the lifting for the heavy load area is s_1, the cam rotation angle in overload area is θ_1, so cam rotation angle in light load condition is $\theta_{max}-\theta_1$ and the lifting is $s_{max}-s_1$.

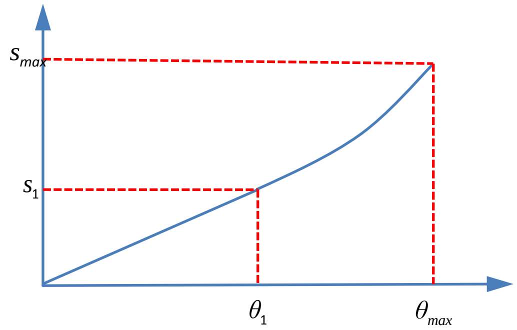

Fig. 4 Composite lifting with rotation angle

3. 1. Involute profile

For the involute profile, lifting has a linear relation with the rotation angle. Involute profile lifting conforms to

$$h=r\theta,\ \left(0\le h\le s_1\right). \qquad (3)$$

For involute profile, the radius of the involute r is a very important parameter. From Eq. (3), we know when r is smaller, the Cam rotation Angle θ_1 is greater to complete same lifting s_1 of heavy load area. But if the radius is too small, the parameter θ_1 is close to θ_{max}, leading to smaller angle in light load area, and influencing the dynamic characteristics of the joint point.

3.2. Quadratic curve profile

It uses the quadratic curve to complete all the light load area. The design on the one hand ensures the fast opening, on the other hand, reduces the shock on the joint point. Two conditions must be meet: 1. Displacement and speed function is continuous on the join point; 2. The requirements of the lifting based on the two conditions. Set a quadratic curve which meets the conditions:

$$h = a\theta^2 + b\theta + c \quad \left(s_1 \le h \le s_{max}\right),$$

where a , b , c satisfy the Eq.(4):

$$\left.\begin{aligned} &h = a\theta_1^2 + b\theta_1 + c = s_1, \\ &\frac{dh}{d\theta}\Big|_{(\theta=\theta_1)} = 2a\theta_1 + b = r, \\ &h = a\theta_{max}^2 + b\theta_{max} + c = s_{max}. \end{aligned}\right\} \tag{4}$$

According to Eq. (4) a , b and c are expressed as:

$$\left.\begin{aligned} &a = \left(\frac{s_{max} - s_1}{\theta_{max} - \theta_1} - r\right) \Big/ \left(\theta_{max} - \theta_1\right), \\ &b = \frac{\theta_{max} + \theta_1}{\theta_{max} - \theta_1} r - \frac{2\theta_1}{\left(\theta_{max} - \theta_1\right)^2}\left(s_{max} - s_1\right), \\ &c = \frac{\theta_1^2}{\left(\theta_{max} - \theta_1\right)^2}\left(s_{max} - s_1\right) - \frac{\theta_1}{\theta_{max} - \theta_1} s_1. \end{aligned}\right\} \tag{5}$$

If r is known, for the given lifting s_1 in heavy load area, then θ_1 is known. According to the Eqs. (3), (4), (5) the cam lifting curve is uniquely identified. The r determines the location of the joint point and the dynamic characteristics of cam follower mechanism.

3.3. Discussion the proper values of r

Considering the requirement of lifting, when s_{max} and θ_{max} are constant, with r smaller, the parameter θ_1 is larger and $\left(\theta_{max} - \theta_1\right)$ is smaller, namely, it is required for the cam to rotate smaller angle to complete the lifting of $\left(s_{max} - s_1\right)$. So smaller r leads greater impact on cam profile and poor dynamic characteristics on joint point.

Set λ as ratio of the average speed of two curves stage, namely:

$$\frac{s_{max} - s_1}{\theta_{max} - \theta_1} = \lambda \frac{s_1}{\theta_1}. \tag{6}$$

When the cam guide rod complete the lifting of s_1, the speed is v_1, when complete the lifting of $\left(s_{max} - s_1\right)$, the speed of the guide rod reaches maximum value v_{max}, according to Eq. (4) and Eq. (6), the relation between v_1 and v_{max} is as following:

$$\frac{v_{max}}{v_1} = 2\lambda - 1. \tag{7}$$

As $\lambda = 1$, the whole segment of cam profile is involute without joint point, but r is very big . To ensure the lifting requirements and little impact of joint point, it is very important to choose the appropriate values of λ .

From Eqs. (3), (6) and (7) r can be calculated as following:

$$r = \frac{\left(s_1 + \dfrac{s_{max} - s_1}{\lambda}\right)}{\theta_{max}}. \tag{8}$$

3.4. Modification of cam profile

To establish the mathematical model of pressure angle and the eccentricity, the angle can be expressed:

$$\alpha = arctan \frac{r\left(h_0 + r\theta\right) - e\sqrt{\left(h_0 + r\theta\right)^2 + r^2 - e^2}}{\left(h_0 + r\theta\right)^2 - e^2}, \tag{9}$$

where $h_0 = \sqrt{R^2 - r^2}$ according to the Eq. (9), with the eccentricity increasing, the angle turns from positive to negative. When the eccentricity $e > r$, it can produce a negative pressure angle between the cam and the roller.

The relationship between the pressure angle and radius of the involute base circle is shown below:

$$\left.\begin{aligned} &\alpha < 0 \qquad e > r, \\ &\alpha = 0 \qquad e = r, \\ &\alpha > 0 \qquad e < r. \end{aligned}\right\} \tag{10}$$

As in the actual situation, there is friction on the contact surface of the cam and the roller, so the zero pressure angle is not the optimal choice. When the pressure angle and the friction coefficient is equal, the negative pressure angle and friction angle balances and the force between guide sleeve and rod is zero. So according to the friction coefficient between roller and cam, choose the best eccentricity e to make system optimal.

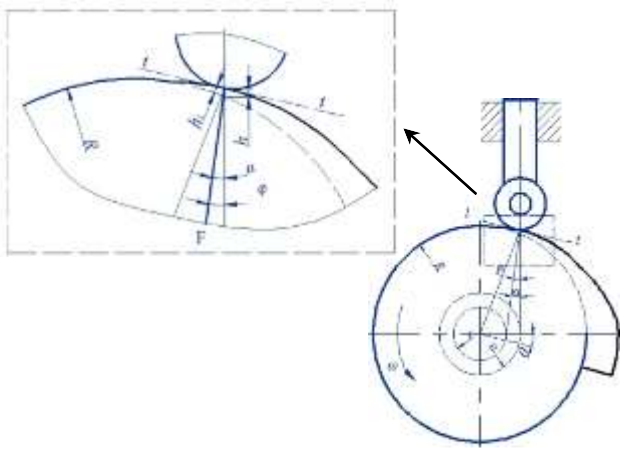

Fig. 5 Modification of cam lifting

As the existence of friction force, eccentricity and radius of involute base circle aren't equal, so the lifting have a small changes. As shown in Fig. 5. According to the geometry relationship, the relationship between the design lifting h_1 and the actual lifting h is

$$h_1 = \sqrt{\left(h + \sqrt{R^2 - e^2}\right)^2 + e^2 - r^2} - \sqrt{R^2 - r^2} \qquad (11)$$

According to Eq. (11) to revise profile curve, it can make the cam meet the requirements of lifting.

4 Example and application

4.1. Specific design example

The example below explains cam lifting design method for further instructions. Use the related parameters of 300MN hydraulic press as example to instruct the design method. The related parameters are: the valve opening force $G = 50$ kN, the total angle $\theta_{max} = 80°$, coefficient of friction between cam and roller $\mu_1 = 0.1$, coefficient of friction between the guide sleeve and rod $\mu_2 = 0.05$, cam lifting in heavy load area is 12 mm, the total lifting is 30 mm.

Step 1: The determination of cam base circle radius. According to the test and analysis of opening force in the field of large hydraulic press valves, the base circle radius is $R = 100$ mm.

Step 2: Determine the involute base circle radius. For different λ the cam lifting relations are shown in Fig. 6. When $\lambda = 1$ the whole segment of cam profile is involute. The base circle radius of involute is $r = 21$ mm. As $\lambda \to \infty$ the base circle radius of involute is $r = 8.6$ mm. The cam rotates 80°, but the lifting is only 12 mm.

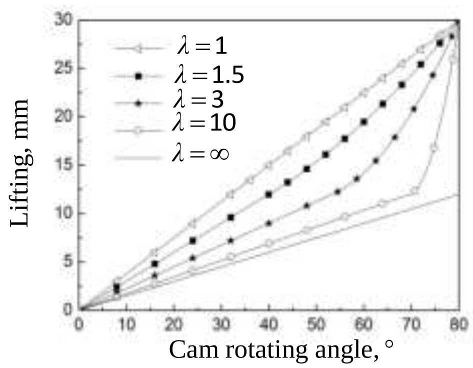

Fig. 6 Lifting with different λ

In order to make sure the cam profile has small impact and the joint point is flexible, the better range of λ is 1.5 to 3 from comprehensive analysis. According to Eq. (8), it generates that the range of the involute base circle is $12.89 \text{ mm} \le r \le 17.19$ mm. Select the middle value $\lambda = 2$ to calculate $r = 15$ mm,

Step 3: The determination of eccentricity e. According to the Eq. (9), set different eccentricity equal to 10, 15, 20, 25 and 30 mm respectively, the pressure angle with the change of different eccentricity is shown in Fig. 7. The pressure angle α change a little with the change of lifting when e is sure. To simplify this analysis, this paper ignore the fluctuation of α.

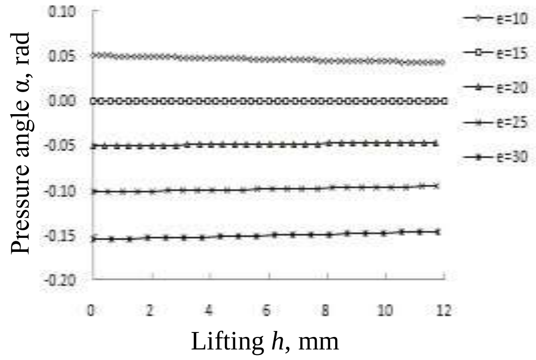

Fig. 7 The relation of α and lifting h

When $r = e = 15$ mm, the pressure angle $\alpha = 0$; when $e \ge 15$ mm, the pressure angle $\alpha \le 0$.

The Fig. 8 shows F_1 and M as the eccentricity changes. It shows the torque range is $1230 \sim 1300$ Nm. F_1 changes very apparently ranging from 0 to 25 kN. So the value of eccentricity influence on guide sleeve is very obvious.

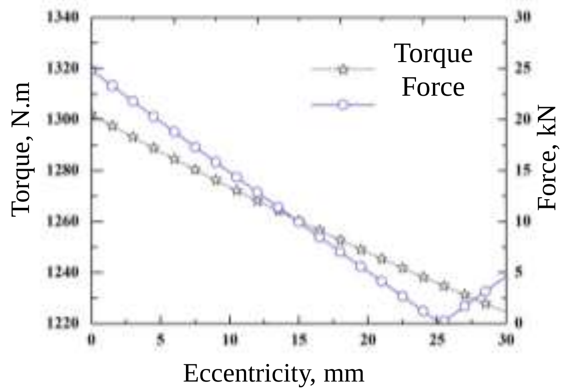

Fig. 8 Torque and force with different eccentricity

Due to the friction coefficient between the cam and roller is 0.01, F_1 is not zero when eccentricity is 15 mm, though the pressure angle is zero. When the eccentricity $e = 25$ mm, the friction angle balances the pressure angle and F_1 is approximately equal to zero.

When $e = 25$ mm torque and force change as lifting increasing in heavy load zone as shown in Fig. 9. It shows the force is very small in the overload zone. The greatest force is 0.45 kN, the maximum torque is 1245 Nm.

Step 4: When the eccentricity is not equal to the radius of base circle and the cam lifting has little change. The cam lifting modification according to Eq. (9). When $r = 15$ mm and $e = 25$ mm, while the actual lifting $h = 12$ mm, the design lifting $h_1 = 11.78$ mm, while the actual lifting $h = 30$ mm, the design lifting

$h_1 = 29.52$ mm. So, when $r = 15$ mm and $e = 25$ mm, take the design lifting $s_1 = 11.78$ mm to ensure the actually lifting 12 mm; take the design lifting $s_{max} = 29.52$ mm to ensure the actually lifting 30 mm.

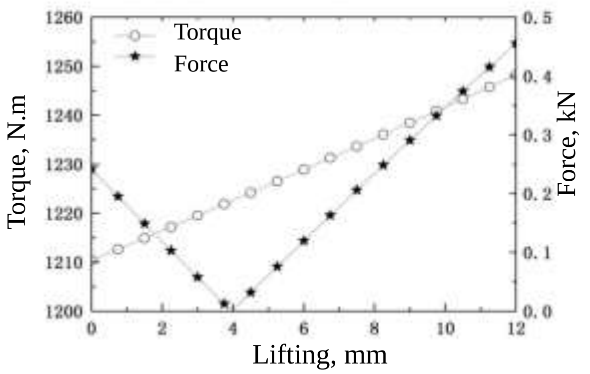

Fig. 9 The force and torque with different lifting when e = 25 mm

4.2. Application situation

The above composite curve lifting cam is applied in industrial field of 300 MN hydraulic press as shown in Fig. 10. Field environment of the original cam is basically identical and the equivalent load of forging work piece has the same statistical rules.

Fig. 10 Cam field application picture

The statistics are based on the replacement frequency of roller and the guide sleeve. The rod bending and guide sleeve fracture happens once a month, while the original parts failure happens three to eight times a month.

Through the statistics data of failure frequency, it find that cam follower with composite curve lifting effectively improved the cam follower mechanism force condition, reduced the fault rate and increased the service life. The analysis illustrates that the method to design the cam lifting for transient and heavy load has certain adaptability.

5. Conclusion

According to the transient and heavy load characteristics of the operation system in the large hydraulic press, this paper proposed a method to reduce force between guide sleeve and bar by using cam follower mechanism with eccentricity. The cam lifting curve with the involute curve and the quadratic curve matches the load characteristic. It uses involute to realize the little pressure angle in heavy load zone, and uses quadratic curve to implement the joint point smooth and the high lifting in low load zone. The paper analyses the selection method of eccentricity considering the existence of friction. The proposed method provides a theoretical guidance for designing cam profile suitable for heavy and transient load. At the same time it improves the reliability of 300 MN hydraulic press system and reduces the failure frequency of the guide sleeve, roller, etc.

Acknowledgments

This work was supported by the national high-tech research and development program of China. The authors greatly appreciate the comments and suggestions by the reviewers.

Reference

1. **Flocker, F.W.** 2012. A versatile cam profile for controlling interface force in multiple dwell cam follower systems, Journal of Mechanical Design 134(9): 1-6. http://dx.doi.org/10.1115/1.4007146.
2. **He, W.; Cao, J.J.; Yang, H.L.** 2005. Review of research of cam mechanisms in China, Chinese Journal of Mechanical Engineering 41(6): 1-6 (in Chinese). http://dx.doi.org/10.3901/JME.2005.06.001.
3. **Hsieh, J.F**. 2010. Design and analysis of cams with three circular-arc profiles, Mech Mach Theory 45(6): 955-965. http://dx.doi.org/10.1016/j.mechmachtheory.2010.02.001.
4. **Gatti, G; Mundo, D.** 2010. On the direct control of follower vibrations in cam follower mechanisms, Mech Mach Theory 45(1): 23-35. http://dx.doi.org/10.1016/j.mechmachtheory.2009.07.010.
5. **Hua, Q.A; Lin, C.J.; Li, Z.Y.** 2005. A universal optimal approach to cam curve design and its applications, Mech. Mach. Theory 40(6): 669-692. http://dx.doi.org/10.1016/j.mechmachtheory.2004.12.005.
6. **Zayas, E.E.; Cardona, S.; Jordi, L.; et al**. 2013. Synthesis of displacement functions by Bezier curves in constant-breadth cams with parallel flat-faced double translating and oscillating followers, Mech. Mach. Theory 62(1): 51-62.
7. **Xuan, G.T.; Shao, Y.Y.; Lv, Z.Q; et al.** 2012. Profile reconstruction of spatial indexing cam and simulation based on NURBS, Transactions of the Chinese Society for Agricultural Machinery 43(6): 227-234.
8. **Hsieh, J.F.; Lin, P.D.** 2007. Application of homogenous transformation matrix to measurement of cam profiles on coordinate measuring machines, Int J Mach Tool Manu 47(10): 1593-1606. http://dx.doi.org/10.1016/j.ijmachtools.2006.11.001
9. **Lampinen, J.** 2003. Cam shape optimisation by genetic algorithm, Computer Aided Design 35(8): 727-737. http://dx.doi.org/10.1016/S0010-4485(03)00004-6.

10. **Pridgen, B.** 2012. Comparison of Polynomial Cam Profiles and Input Shaping for Driving Flexible Systems, Journal of Mechanical Design 134(12): 1-7. http://dx.doi.org/10.1115/1.4007797.
11. **Biswas, A.; Stevens, M.; Kinzel, G.L.** 2004. A comparison of approximate methods for the analytical determination of profiles for disk cams with roller followers, Mech Mach Theory 39(6): 645-656. http://dx.doi.org/10.1016/j.mechmachtheory.2003.10.001.
12. **Andresen, U.; Singhose, W.** 2004. A simple procedure for modifying high speed cam profiles for vibration reduction, Journal of Mechanical Design1 26(6): 1105-1108.
13. **Garziera, R.; Carra,S.; Pellegrini, M.** 2004. Synthesis of cams with negative radius follower and evaluation of the pressure angle, Mech Mach Theory 39(10): 1017-1032. http://dx.doi.org/10.1016/j.mechmachtheory.2004.05.001.
14. **Chen, H.; Tan, J.P.** 2012. Opening force characteristic of water valve with high pressure and large flow capacity in large die forging hydraulic press, Journal of Central South University 43(8): 2993-2999 (in Chinese).

J. Yang, J. Tan, L. Zeng, S. Liu

PEREINAMOMS IR DIDELĖMS APKROVOMS SKIRTO KUMŠTELIO PAKILIMO KREIVĖS PROJEKTAVIMAS IR ANALIZĖ

R e z i u m ė

Šiame straipsnyje pateikiamas didelio hidraulinio preso kumštelio pereinamoms ir didelėms apkrovoms perduoti pakilimo kreivės projektavimo metodas. Pakilimo kreivė sudaroma iš dviejų dalių pradiniam segmentui panaudojant involiutę ir užpakaliniam segmentui panaudojant kvadratinę kreivę. Pakilimo kreivė gali atitikti apkrovos charakteristikoms. Ji gali pasiekti mažesnį slėgio kampą sunkioms apkrovoms naudojant involiutę ir didesnį atidarymo greitį lengvoms apkrovoms naudojant kvadratinę kreivę. Tikslu detaliai aprašyti metodą, suprojektuota kumštelio pakilimo kreivė 300 MN kalimo hidraulinio preso operacinei sistemai. Taikymai parodė, kad kumštelis su tokia pakilimo kreive gali efektyviai sumažinti žalingą valdančiosios movos ir kumštelio sekiklio dažnį. Pateiktas efektyvaus kumštelio pakilimo kreivės pereinamoms ir sunkioms apkrovoms projektavimo metodas.

J. Yang, J. Tan, L. Zeng, S. Liu

DESIGN AND ANALYSIS OF CAM LIFTING CURVE IN APPLYING TO TRANSIENT AND HEAVY LOAD

S u m m a r y

This paper presents a method for designing cam lifting curve applying to transients and heavy load in the operating system of large hydraulic press. The lifting curve is made up of two parts with the frontal segmental using involute and the posterior segment using quadratic curve. The lifting curve can match up with the characteristics of load. It can achieve smaller pressure angle in heavy load area using involute, and faster opening speed in light load area using quadratic curve. In order to give a detailed description of the method, it designed the cam lifting cure used in the operating system of 300MN forging hydraulic press. Field applications showed that the cam with this lifting curve can effectively reduce the fault frequency of the guide sleeve and the cam follower. It provides an effective method for designing the cam lifting curve with transient and heavy load.

Keywords: hydraulic press, cam lifting curve, involute, quadratic curve, heavy and transient load.

Received October 11, 2013
Accepted May 30, 2014